Book of presentations of the 2nd Symposium on High-Temperature Heat Pumps

9 September 2019
Copenhagen
Denmark

Editors:

Benjamin Zühlsdorf, Danish Technological Institute
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SINTEF

DANISH TECHNOLOGICAL INSTITUTE

DTU

Foreword

Heat pumps operating at higher temperatures enable the supply of energy efficient and emission free process heat. High temperature heat pumps are the "hidden champions" when it comes to de-carbonizing the industry in order to meet the climate targets of the Paris agreement. Utilizing this potential of industrial heat pumps is highly attractive since it allows the industry not only to reduce emissions but also their primary energy consumption.
However, there are challenges connected with implementing heat pump technology, especially in high temperature applications like industrial processes and district heating. There is a need for technical innovations to achieve lower specific investment costs and increased energy efficiency while maintaining technical feasibility and stable operation.
The 2^{nd} Conference on High-Temperature Heat Pumps was organized in collaboration of SINTEF Energi, the Technical University of Denmark (DTU) and the Danish Technological Institute (DTI). It was held on the 9^{th} of September in Copenhagen, Denmark.
The day comprised 15 oral presentations and 12 poster presentations with speakers from in total 11 different countries. The presentations were organized in three sessions with oral presentations, and the day was concluded by a poster session. The poster session created the possibility for fruitful discussions of the posters as well as the oral presentations. The presentations were organized in three sessions with a focus on:

- Potential and demand for high-temperature heat pumps
- Industrial cases and examples of successful integration of heat pumps
- Current developments and trends for high-temperature heat pumps

There was a wide consent among the presenters and the participants about the large potential of high-temperature heat pumps (HTHP). A broad variety of potential applications was presented and the considerable potential that HTHPs imply with respect to reducing GHG emissions by electrifying the industrial heat supply becomes apparent. Thomas Nowak, European Heat Pump Association, underlined in his keynote speech, that this potential may only be exploited, if the CO_2 emissions are internalized, if the tax burden on electricity and fossil fuels for heating is reviewed and if the subsidies for fossil fuel-based technologies are stopped.
The presentations about the technical developments revealed that there are different systems under development, which are (close to) becoming commercially available for supply temperatures of up to 150 °C in different capacity ranges. The beneficial impact of HTHPs was presented for different case studies. It was found to be highest, if the integration process comprised a simultaneous optimization of both the process and the heat pump system.
The presentations did however also reveal the requirement and the potential of further developments. The required developments are covering a broad range

and aim among others on improved performances, decreased investment costs and simplified and improved integration processes. The conference presentations indicated the following developments to be promising contributions for accelerating the deployment of high-temperature heat pumps:

- Optimization of cycle layout and component performances
- Improved integration procedures considering a re-evaluation of supply temperatures, buffer tanks and possibilities to access cheap electricity
- Compressors capable of high supply temperatures and lubrication systems if required
- Reduction of investment cost

Considering the rapid development of R&D activities that we experienced since the organization of the previous event in 2017, we are looking forward to following up with the ongoing developments and especially with the new developments that may be expected in the next two years.
As the organizing committee, we want to thank all participants for their attendance and in particular the speakers for interesting and well-prepared presentations. In the following, a compilation of all presentations and posters, supplemented with an extended abstract, can be found.

Benjamin Zühlsdorf, Danish Technological Institute
Michael Bantle, SINTEF
Brian Elmegaard, Technical University of Denmark

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1 Potential and demand for high-temperature heat pumps

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1.2 Analysis of technologies and potentials for heat pump-based process heat supply above 150 °C, Benjamin Zühlsdorf, DTI

1.3 High Temperature heat pump market assessment in industry sectors, Jean-Marie Fourmigué, EDF

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1.6 Industrial heat pumps in the Netherlands - developments and demonstrations, Robert de Boer, TNO

How can high temperature heat pumps contribute to reach Europe's climate targets?

Thomas Nowak[1]

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Thomas.nowak@ehpa.org

Keywords:

High temperature heat pump, European energy and climate policy

Abstract

Heat pump technologies are perfectly suited to become the hub of a European decarbonised energy system. They integrate renewable and waste heat in a highly efficient manner thus reducing CO2 emissions from heating and cooling, potentially close to zero.

The technology is one of the options in the quest to limit global warming to below 1,5°C by 2050 – unfortunately, progress is not fast enough, even though both technology recognition with policy makers and annual sales have increased.

At the end of 2018 11.8 million residential and light commercial heat pumps were installed in Europe, contributing 128 TWh of renewable energy to Europe's energy system and saving 33 Mt of CO_2. At the same time, the stock of heat pumps has reduced import dependency and secured local jobs. However it is mainly residential heat pumps that have received political recognition while a significant lack of understanding exist for the contribution potential of large / industrial heat pumps. A simple truth prevails: Heat pumps, large or not, cannot contribute to any target if their potential is not recognized.

Clearly, our sector has more work to do make the benefits of the technology known. Heat pumps contribute to the renewable energy target (32%), the energy efficiency target (32.5%) and the CO2 emission reduction target (40%). Since reaching the targets by 2030 is more than uncertain, available solutions should be very welcome to the responsible policy makers.

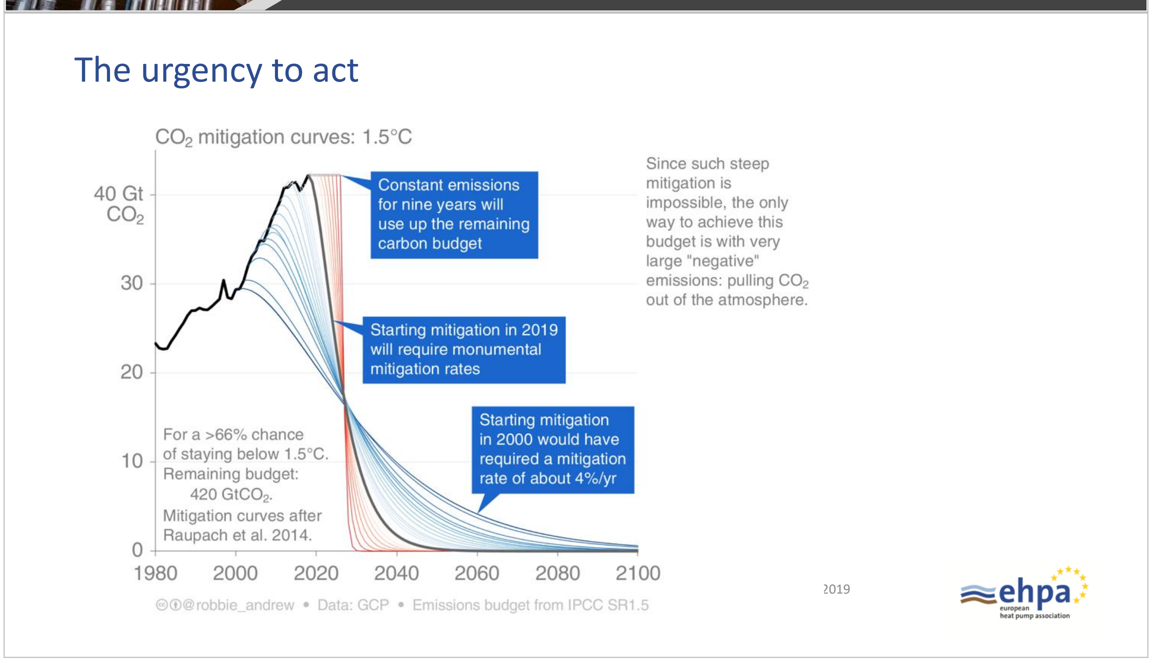
Not only must the technology be recognized, but its deployment must be accelerated, eg. by creating a market framework that allows for a successful competition of heat pumps with the fossil incumbents. In order to achieve that, three major steps have to be taken

1. the external cost of polluting the environment with CO2 must be internalized via a CO2 price,
2. the burden put on electricity via taxes and levies has to be review in order to reduce the electricity price for all sectors,
3. the support of fossil technologies via subsides must be stopped immediately.

Supporting action can be an improved energy statistics that distinguishes between energy sources used per industrial sector and temperature level. Additional positive effects are expected from sector integration, which would facilitate a larger share of renewable electricity generation and a more stable electric grid.

In conclusion, heat pumping technologies are ready to contribute to the greatest challenge of our times – a decarbonised energy system. It is now the responsibility of policy makers to shape a market framework that turns potential into reality – fast.


How can high temperature heat pumps contribute to reach Europe's climate targets?
Thomas Nowak | Secretary General
European Heat Pump Association,
Brussels, Belgium
ehpa
european heat pump association


The urgency to act
CO2 mitigation curves: 1.5°C
40 Gt CO2
30
20
10
0
1980
2000
2020
2040
2060
2080
2100
Constant emissions for nine years will use up the remaining carbon budget
Starting mitigation in 2019 will require monumental mitigation rates
Starting mitigation in 2000 would have required a mitigation rate of about 4%/yr
For a >66% chance of staying below 1.5°C. Remaining budget: 420 GtCO2. Mitigation curves after Raupach et al. 2014.
@robbie_andrew • Data: GCP • Emissions budget from IPCC SR1.5
Since such steep mitigation is impossible, the only way to achieve this budget is with very large "negative" emissions: pulling CO2 out of the atmosphere.
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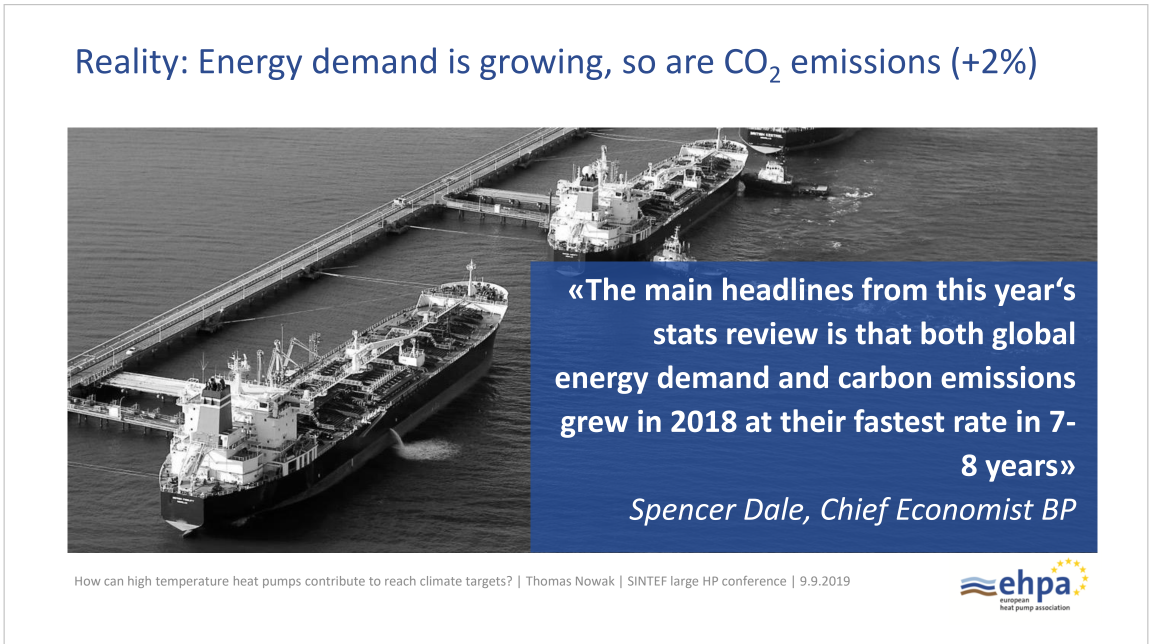
Reality: Energy demand is growing, so are CO_2 emissions (+2%)
«The main headlines from this year's stats review is that both global energy demand and carbon emissions grew in 2018 at their fastest rate in 7-8 years»
Spencer Dale, Chief Economist BP
How can high temperature heat pumps contribute to reach climate targets? | Thomas Nowak | SINTEF large HP conference | 9.9.2019
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"We normally think of heat pumps in the context of cooling and heating of residential areas, where the technology is most commonly employed. It is therefore interesting to learn about your assessment, showing that up to 10% of industrial process heat can be covered by heat pumps."
High level representative of the European Commission in a letter to EHPA
ehpa
european heat pump association


The challenge
»Heat pumps, large or not, can not contribute to any target if their potential is not recognized«
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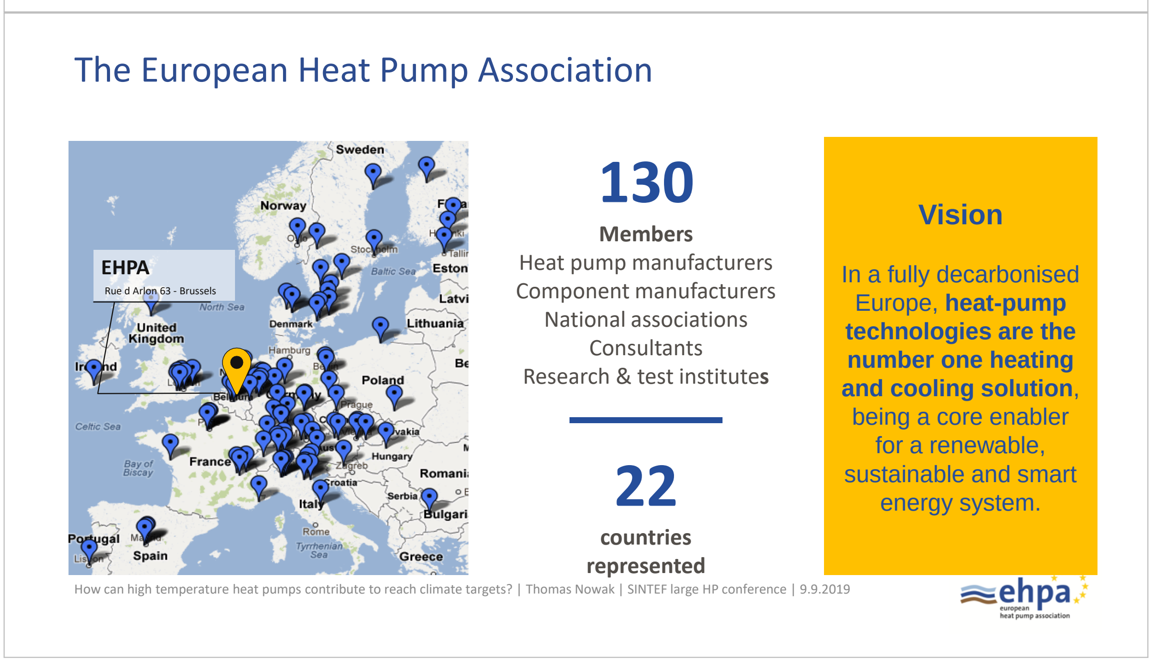
The European Heat Pump Association
EHPA
Rue d Arlon 63 - Brussels
Sweden
Norway
United Kingdom
Ireland
Denmark
Lithuania
Poland
Belgium
France
Hungary
Romania
Serbia
Croatia
Italy
Bulgaria
Greece
Portugal
Spain
130
Members
Heat pump manufacturers
Component manufacturers
National associations
Consultants
Research & test institutes
22
countries represented
Vision
In a fully decarbonised Europe, heat-pump technologies are the number one heating and cooling solution, being a core enabler for a renewable, sustainable and smart energy system.
How can high temperature heat pumps contribute to reach climate targets? | Thomas Nowak | SINTEF large HP conference | 9.9.2019
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Heat Pumps provide heating, cooling and hot water for...
Residential Applications
Commercial Applications
Industrial Applications
District Heating
Big buildings
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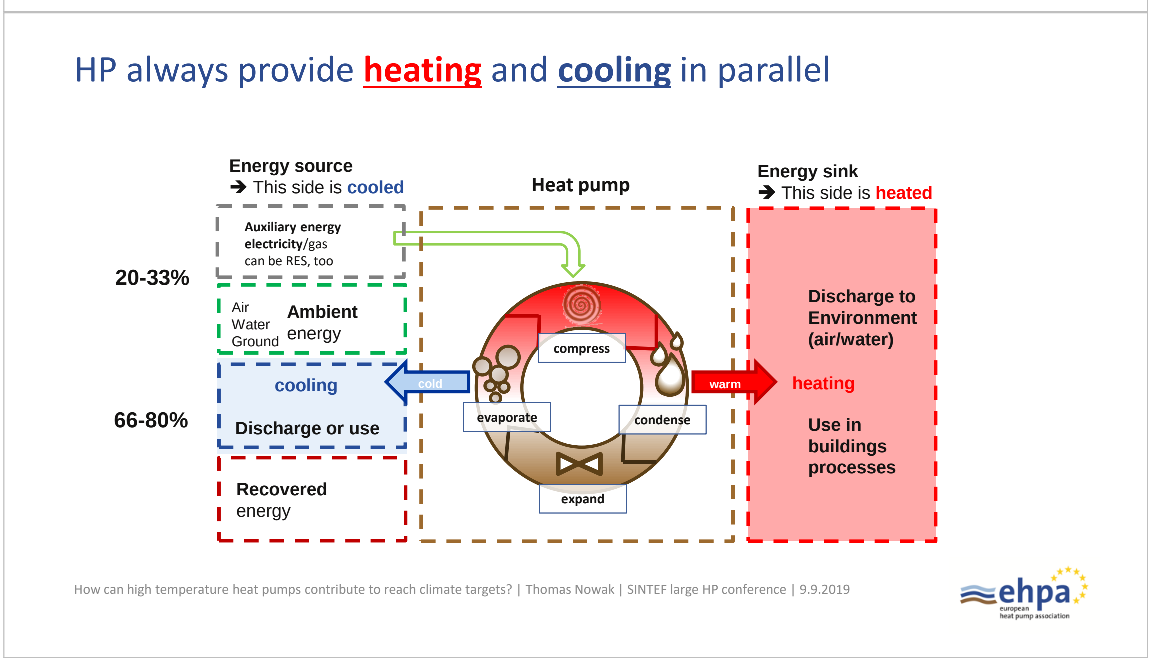
HP always provide heating and cooling in parallel
Energy source
➔ This side is cooled
Heat pump
Energy sink
➔ This side is heated
Auxiliary energy electricity/gas can be RES, too
20-33%
Air
Water
Ground
Ambient energy
cooling
cold
Discharge or use
66-80%
Recovered energy
compress
evaporate
condense
expand
warm
Discharge to Environment (air/water)
heating
Use in buildings processes
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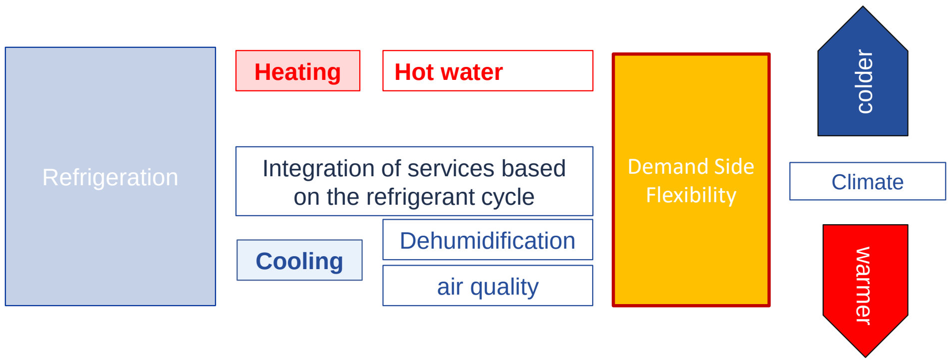
What is a heat pump?



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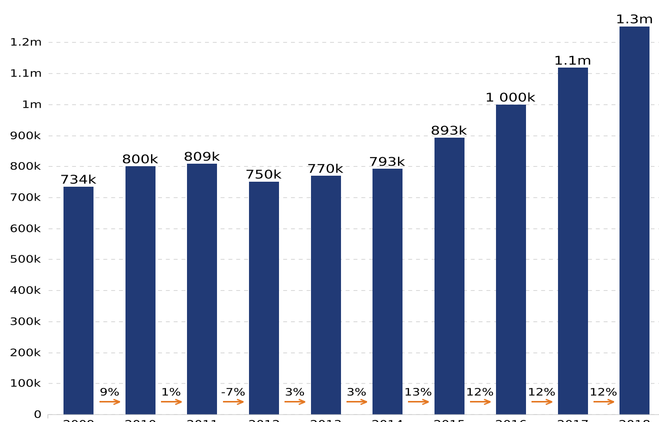
Market growth '05 - '18



12%

Growth in 2018

11.8 m

Heat Pump installed

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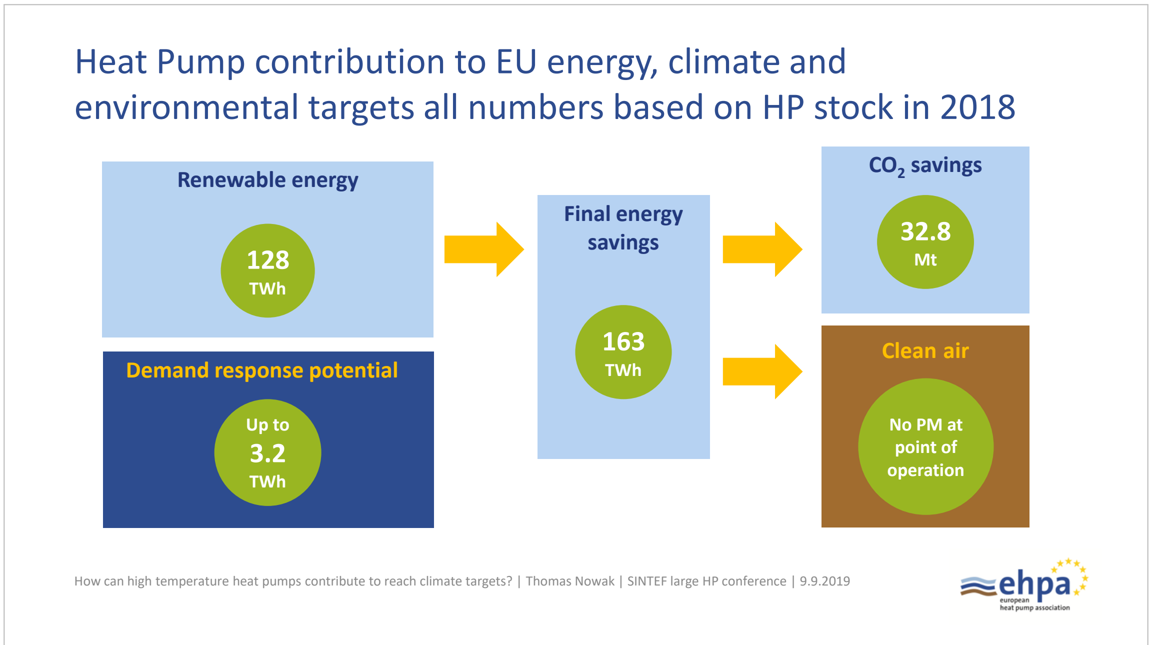
Heat Pump contribution to EU energy, climate and environmental targets all numbers based on HP stock in 2018
Renewable energy
128 TWh
Demand response potential
Up to 3.2 TWh
Final energy savings
163 TWh
CO_2 savings
32.8 Mt
Clean air
No PM at point of operation
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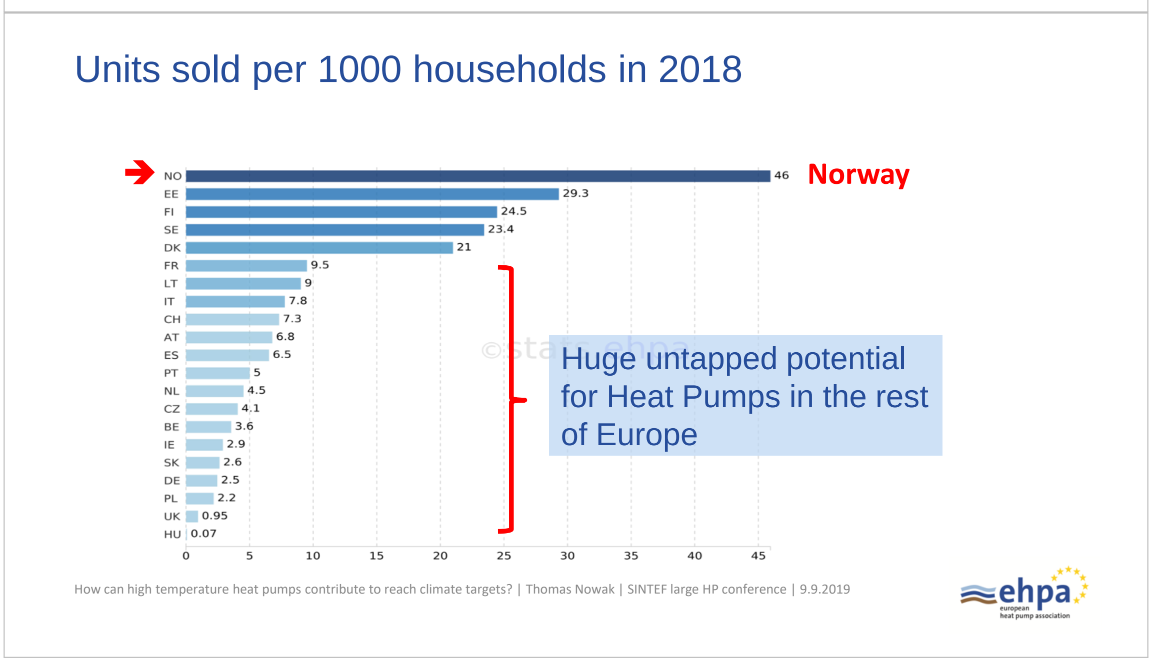
Units sold per 1000 households in 2018
NO 46
EE 29.3
FI 24.5
SE 23.4
DK 21
FR 9.5
LT 9
IT 7.8
CH 7.3
AT 6.8
ES 6.5
PT 5
NL 4.5
CZ 4.1
BE 3.6
IE 2.9
SK 2.6
DE 2.5
PL 2.2
UK 0.95
HU 0.07
0 5 10 15 20 25 30 35 40 45
Norway
Huge untapped potential for Heat Pumps in the rest of Europe
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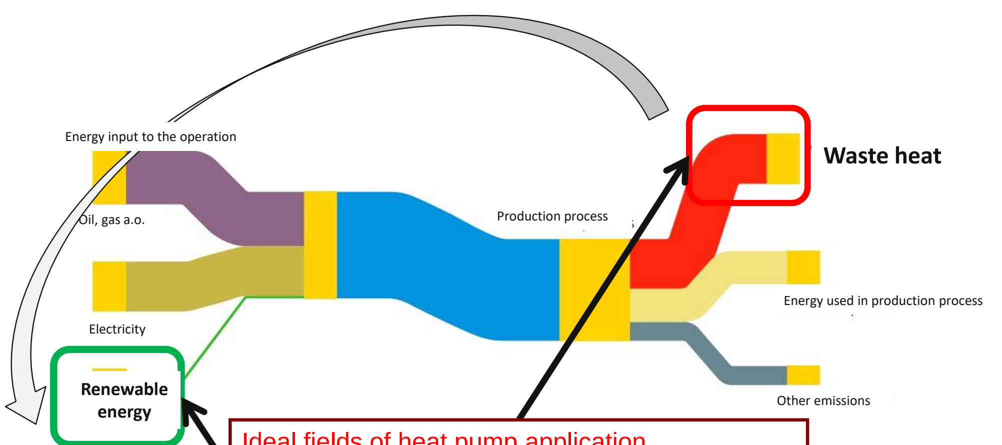
The potential of large heat pumps - from a policy makers/society perspective

Heat pumps in industrial applications (& district systems)



Ideal fields of heat pump application
- Include **renewable energy**
- Make use of waste heat **(energy efficiency)**

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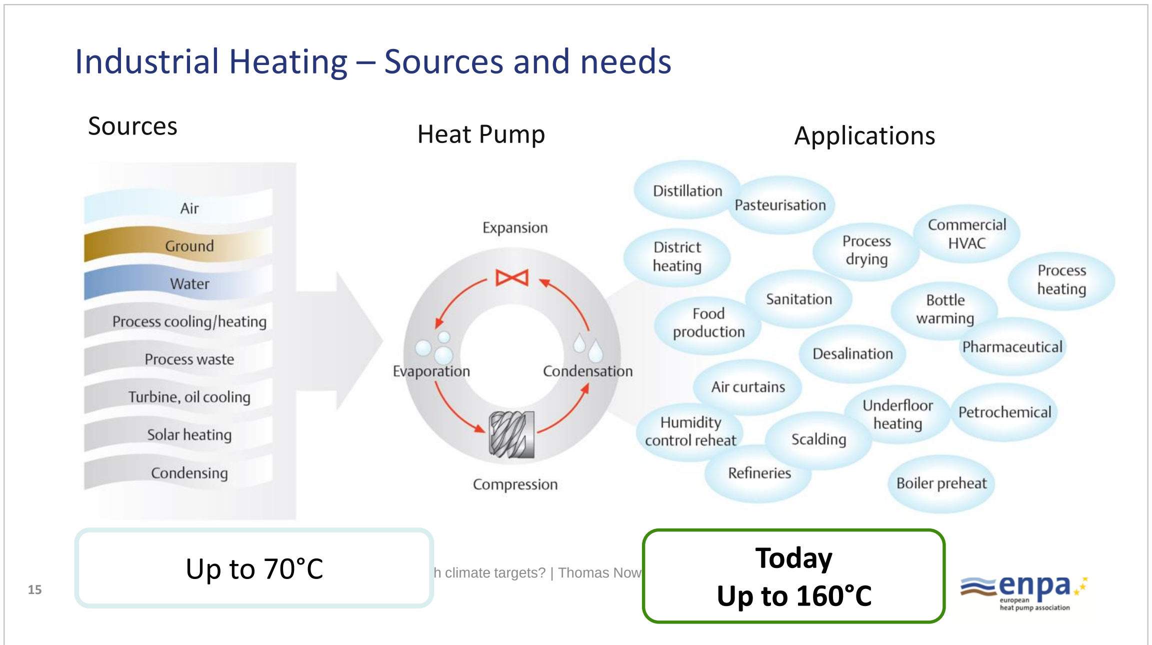
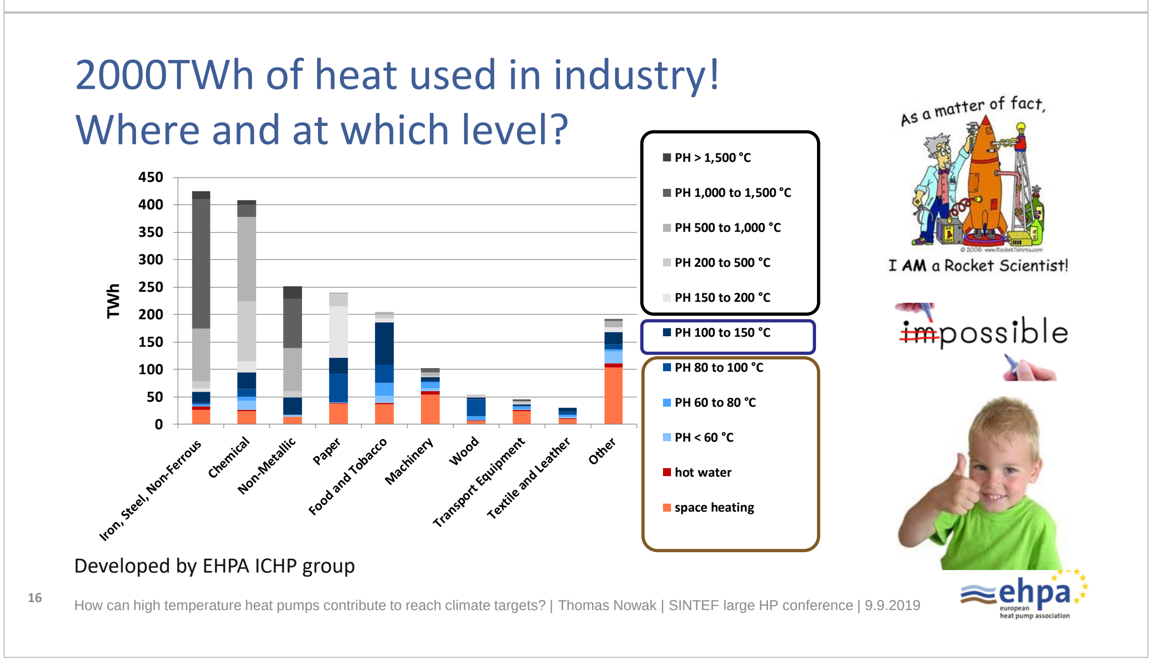

Industrial Heating – Sources and needs
Sources
Heat Pump
Applications
Air
Ground
Water
Process cooling/heating
Process waste
Turbine, oil cooling
Solar heating
Condensing
Expansion
Evaporation
Condensation
Compression
Distillation
Pasteurisation
District heating
Process drying
Commercial HVAC
Process heating
Sanitation
Bottle warming
Food production
Desalination
Pharmaceutical
Air curtains
Underfloor heating
Petrochemical
Humidity control reheat
Scalding
Refineries
Boiler preheat
Up to 70°C
h climate targets? | Thomas Now
Today
Up to 160°C
15
enpa
european heat pump association


2000TWh of heat used in industry!
Where and at which level?
TWh
450
400
350
300
250
200
150
100
50
0
Iron, Steel, Non-Ferrous
Chemical
Non-Metallic
Paper
Food and Tobacco
Machinery
Wood
Transport Equipment
Textile and Leather
Other
PH > 1,500 °C
PH 1,000 to 1,500 °C
PH 500 to 1,000 °C
PH 200 to 500 °C
PH 150 to 200 °C
PH 100 to 150 °C
PH 80 to 100 °C
PH 60 to 80 °C
PH < 60 °C
hot water
space heating
As a matter of fact,
I AM a Rocket Scientist!
impossible
Developed by EHPA ICHP group
16
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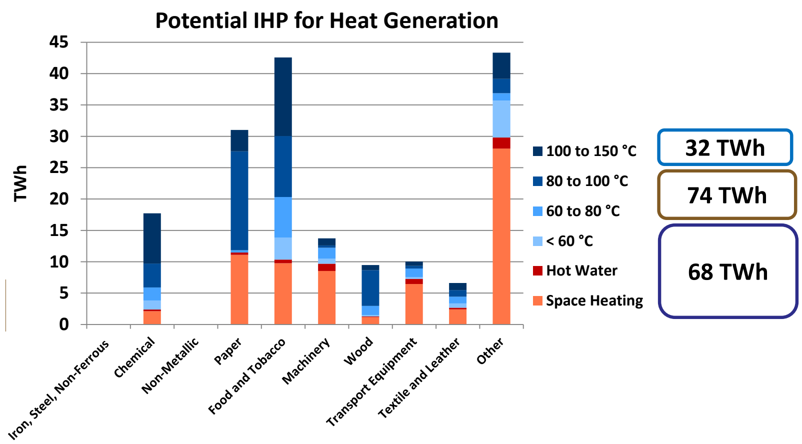
Which heating needs can be covered by heat pumps?



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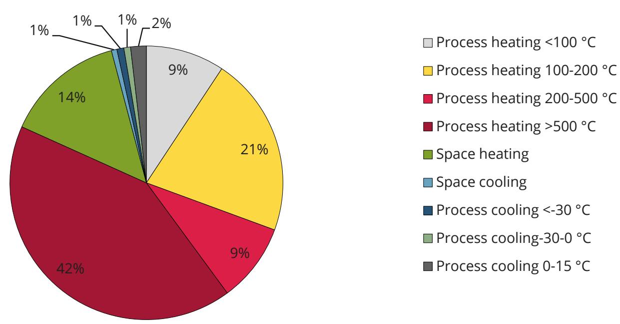
Heatroadmap Europe (2015)

www.heatroadmapeurope.eu

Process heating <100 °C	222,5
Process heating 100-200 °C	508,1
Process heating 200-500 °C	222,5
Process heating >500 °C	998,6
Space heating	338,0
Space cooling	15,6
Process cooling <-30 °C	19,9
Process cooling-30-0 °C	18,8
Process cooling 0-15 °C	44,3



659 TWh in temperature ranges below 100°C | 1167 TWh if 100°-200° is included

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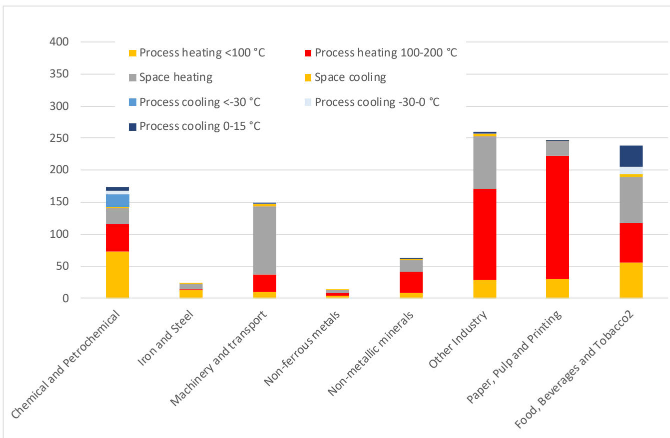
Final energy demand in TWh | 2015

www.heatroadmapeurope.eu



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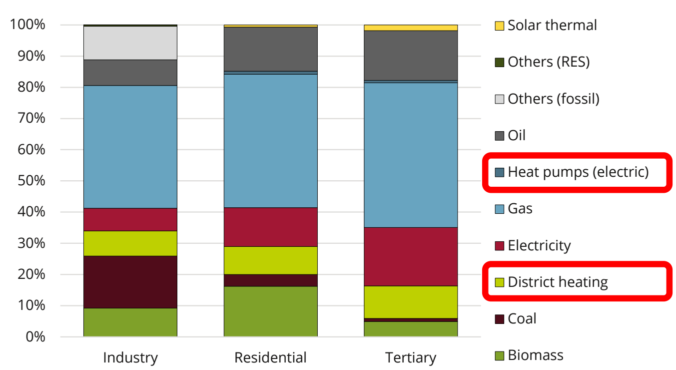
Energy carriers used to cover demand 2015

www.heatroadmapeurope.eu



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EHPA action

- Shaping a „incustrial and commercial heat pumps working group
- Webinars
- Workshops with policy makers
- Special session at annual conference hp-forum.eu/
- Projects
- Application brochures

+ action of other stakeholders:
IEA Annex 48, national programs

How can high temperature heat pumps contribute to reach climate targets? | Thomas Nowak | SINTEF large

Where do large heat pumps fit in the EU legislative landscape?

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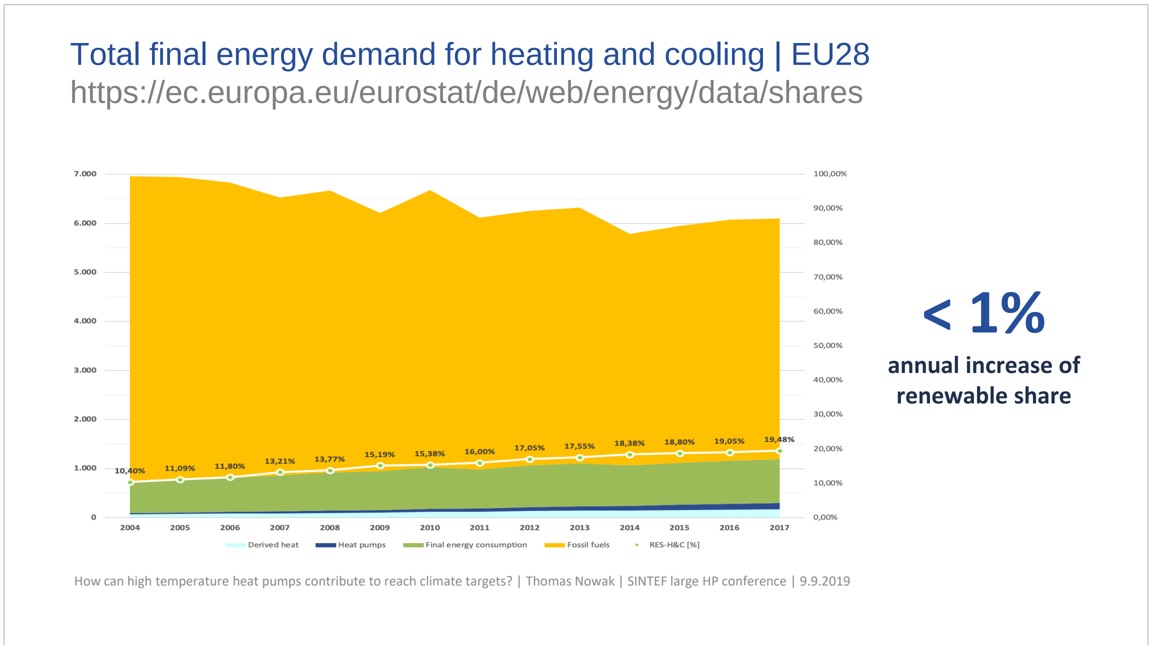
Total final energy demand for heating and cooling | EU28
https://ec.europa.eu/eurostat/de/web/energy/data/shares
7.000
6.000
5.000
4.000
3.000
2.000
1.000
0
100,00%
90,00%
80,00%
70,00%
60,00%
50,00%
40,00%
30,00%
20,00%
10,00%
0,00%
10,40%
11,09%
11,80%
13,21%
13,77%
15,19%
15,38%
16,00%
17,05%
17,55%
18,38%
18,80%
19,05%
19,48%
2004
2005
2006
2007
2008
2009
2010
2011
2012
2013
2014
2015
2016
2017
Derived heat
Heat pumps
Final energy consumption
Fossil fuels
RES-H&C [%]
< 1%
annual increase of renewable share
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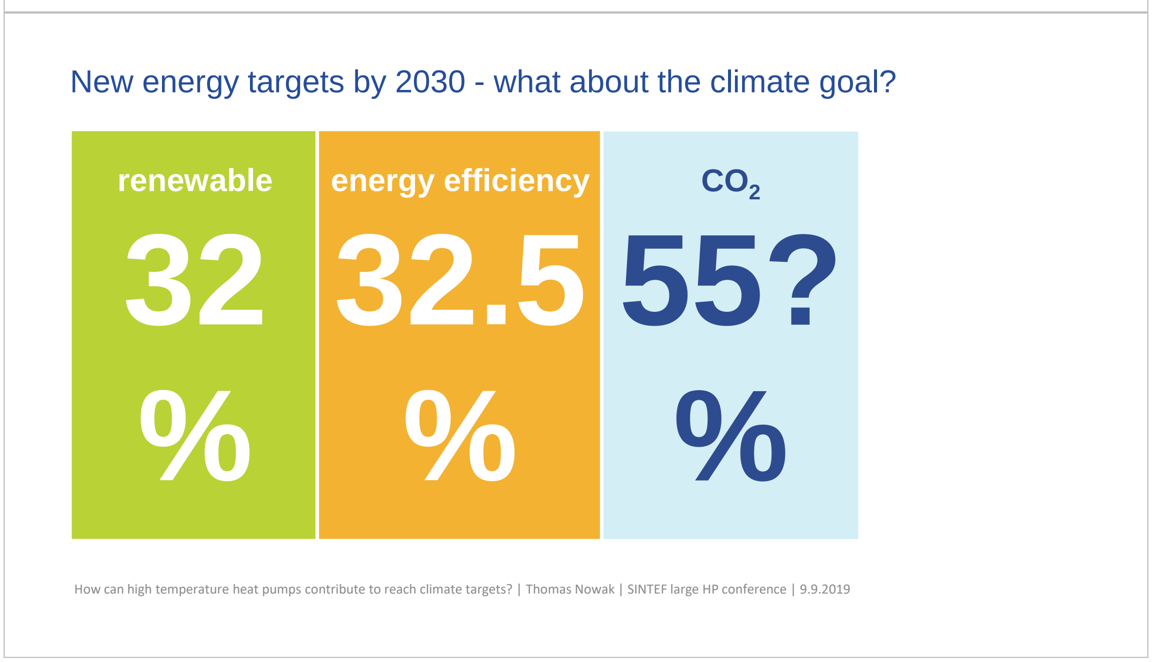
New energy targets by 2030 - what about the climate goal?
renewable
32 %
energy efficiency
32.5 %
CO2
55? %
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Renewable Energy Directive

Heat pumps use renewable energy from air / water / soil, waste energy treated similarly

renewable

32 %

- Focus on RE heating/cooling, acknowledge waste heat
- Annual increase of RE heating/cooling (Art. 23) **+1.3pp**
- More power to prosumers (heat & electricity)
- Upward revision **2023**
- Part of national energy &climate plans NECP

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Energy efficiency directive (2012/27/EU, amended by 2018/2002/EU)

Heat pumps are the most efficient technology for heating and cooling

energy efficiency

32.5 %

- Obligation to reduce annual final energy demand (Art. 7)
  - 1.5% annually until 2020
  - 0.8% annually until 2030
- **New primary energy factor: 2.1**

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Reduction of CO2 emissions

Heat pumps are one of the lowest-emission technologies

CO_2

55?

%

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EU low-carbon Energy roadmap: 100% emission reduction by 2050

"Meeting the Paris goal of keeping climate change well below 2°C – and aiming for no more than 1.5°C – requires bold action, including reaching climate neutrality this century" - *Miguel Arias Cañete*

- Carbon neutrality to become 2050 target
- Commissions work will include "in-depth analysis of the economic, social and environmental transformations needed to inform the political debate in the context of the development of the mid-century strategies"
- Public consultation ongoing
- Only 2 scenarios inline with max. 1.5°C warming

How can high temperature heat pumps contribute to reach climate targets? | Thomas Nowak | SINTEF large HP conference | 9.9.2019

Heat Pumps in the EU energy system 2018

5.6 million boilers — 1.1 mill. heating heat pumps (19.6%)

7.8 million sanitary hot water — 161 000 dedicated hot water heat pumps (2%)

Industrial heating and cooling — ?

- Use of fossil energy for heating dominates in Europe!
- To decarbonise heating and cooling heat pumps are a perfect option

How can high temperature heat pumps contribute to reach climate targets? | Thomas Nowak | SINTEF large HP conference | 9.9.2019

100% renewable heating is possible – with heat pumps

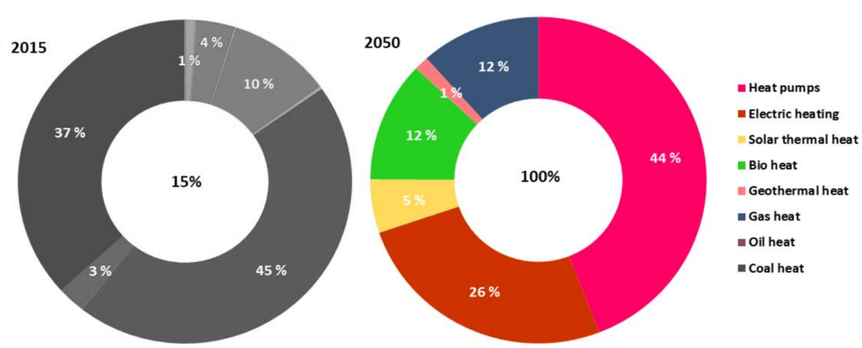

Figure ES-4: Shares of heat supply in 2015 and 2050.

Source: Global Energy System based on 100% Renewable Energy – Energy Transition in Europe Across Power, **Heat,** Transport and Desalination Sectors

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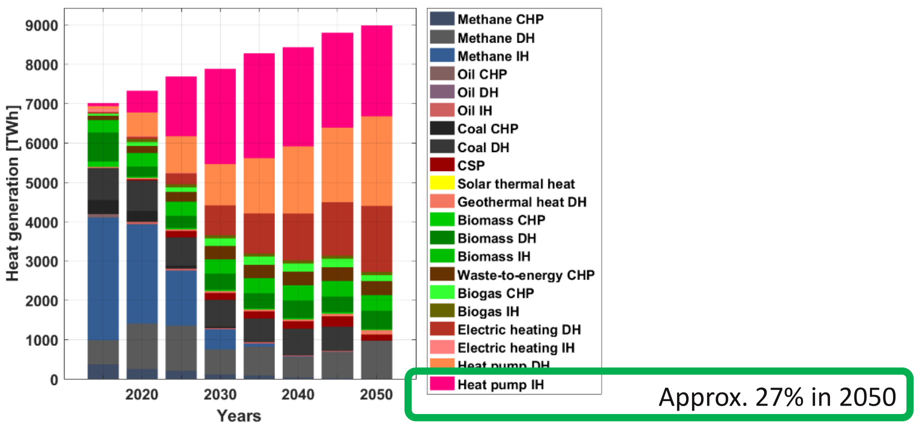
The electrification of industrial heat



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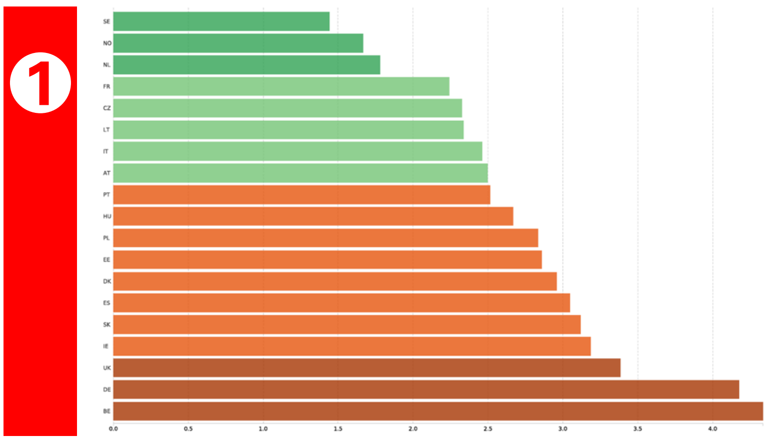
The glass ceiling

How can high temperature heat pumps contribute to reach climate targets? | Thomas Nowak | SINTEF large HP conference | 9.9.2019

Relative prices: electric power vs. gas in Europa



High factors
=
High operative costs for heat pumps

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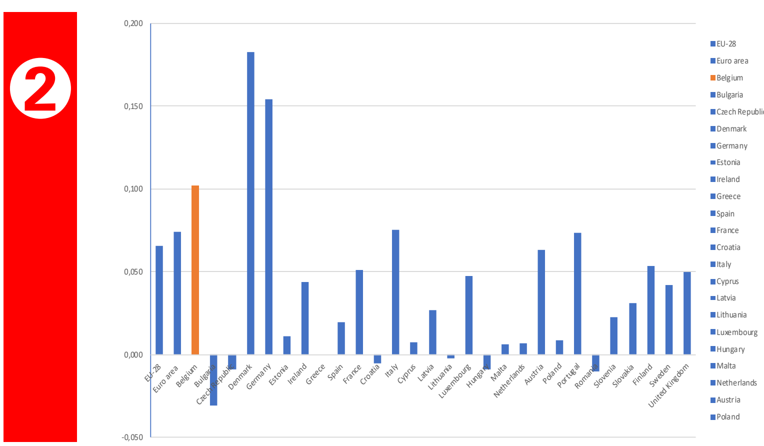
European energy taxation favours fossil gas



Postive values
=
Tax burden on electricity is higher

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Impact of a CO_2 price signal

3

Consequence for standard building

- 6t CO_2-emission/year
- Annual pollution cost: 690 €

Consequence for an industrial process subject to ETS

- Current price: 25€
- Emission cost increase x 4.6

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Subsidies continue to be paid for fossil energy in Europe

4

- CAN Europe2017: 122 billion € / year (in 14 EU countries)
- IEA2013: fossil fuel consumption subsidies USD 548 billion
- OECD2013 estimates total fossil fuel support
  - at USD 160-200 billion/year for OECD and BRIICS states
  - 39 billion for EU member states

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ehpa european heat pump association

Conclusions

- Large heat pumps are increasingly often recognized
- Possibilities are more and more understood
- Need for action (Fridays for future etc) is stressed by the public
- Heat pump technology is ready
- Positive market framework still missing to accelerate market deployment

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Analysis of technologies and potentials for heat pump-based process heat supply above 150 °C

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Keywords:
Cascade multi-stage steam compression, Decarbonization, High-temperature heat pump, Process heat, Reversed Brayton cycle, R718, R744.

Introduction

The ambitions to reduce greenhouse gas emissions do inevitably require sustainable alternatives to fossil fuel-based combustions for supply of process heat to industrial processes. Electricity-driven heat pumps imply the general potential to operate emission free and do thereby represent a sustainable long-term solution for emission free process heat supply.

Currently available heat pump technologies are however limited to supply temperatures of 100 °C to 150 °C, while electric boilers and biomass boilers are often mentioned as alternatives in energy transition strategies. The overall feasibility for heat pump systems in such applications is among others limited by technical component constraints as well as limited thermodynamic performances, resulting in limited operating performances.

Zühlsdorf et al. [1] have therefore analyzed the possibilities for heat pump-based process heat supply at large capacities and temperatures above 150 °C. They evaluated the technical and economic feasibility of two heat pump systems for two case studies. The main results from [1] are summarized by this extended abstract. The article focused on large-scale applications and considered components as known from oil- and gas applications, as these are capable of operating in more challenging conditions and enable exceeding the limitations known from available refrigeration equipment [2]. In addition, the focus was on applications, in which the plant owners have access to electricity at low costs or the possibility to invest in own renewable electricity generators, such as wind farms and photovoltaics, as these are ensuring low levelized cost of electricity [3].

Methods

The study considered two different heat pump systems, namely a cascade multi-stage steam compression system and a reversed Brayton cycle. The cascade multi-stage steam compression system is shown in Figure 1 and consists of bottom cycles that are recovering the heat from the heat sources while providing heat to the evaporator of the top cycle, in which the steam from the evaporator is compressed in several stages. The steam is cooled by liquid injection after each compression stage. The system can supply steam at every pressure level to the system, ensuring an optimal integration into the process and thereby maximum performances.

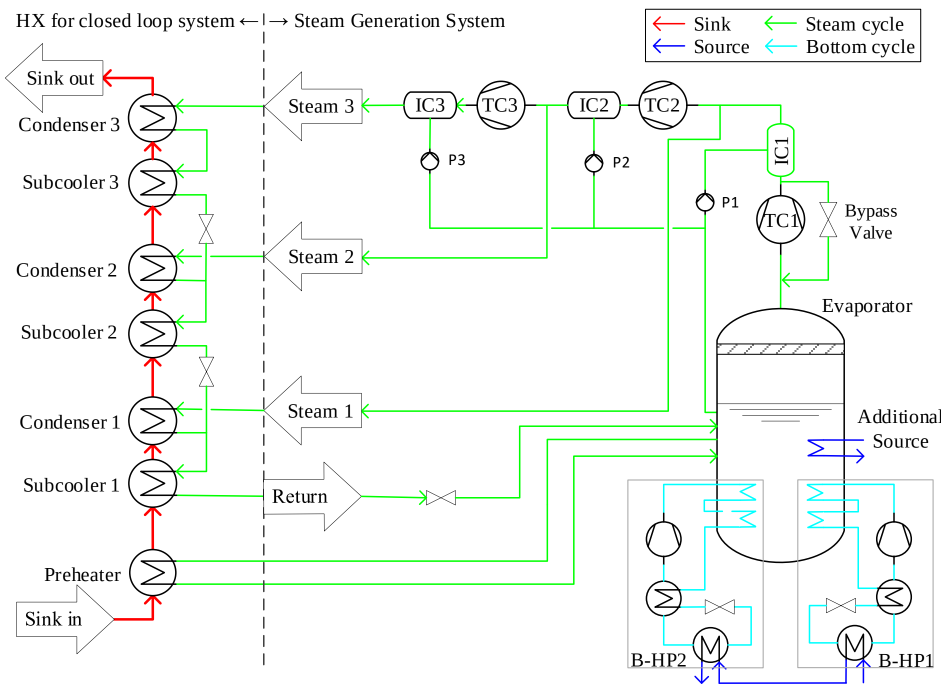

Figure 1: Flow sheet of a cascade heat pump with a multi-stage R-718 cycle for steam generation or closed loop heat supply at different temperature levels (B-HP = Bottom heat pump, IC = Intercooler, P = Pump, TC = Turbocompressor), [1]

The less complex layout of the reversed Brayton cycle is shown in Figure 2. The cycle consists of three heat exchangers, as well as a turbocompressor and a turboexpander, which are mounted on the same shaft. The cycle uses CO_2 as working fluid and operates completely in the gas phase.

The cycles were modelled with energy and mass balances. Design variables, such as pinch points in the heat exchangers or pressure levels were defined or optimized under consideration of common limitations. The investment cost of the equipment was estimated with cost correlations and validated with estimations obtained from manufacturers.

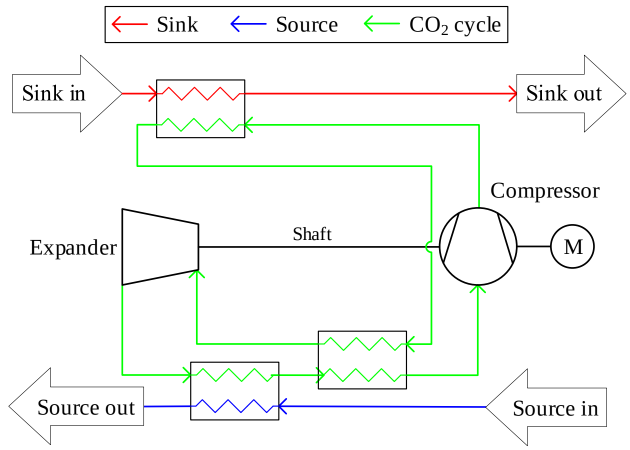

Figure 2: Flow sheet of reversed Brayton cycle, [1]

Both cycles were evaluated for two case studies. The first case study was alumina production in which 50 MW were supplied to heat thermal oil from 140 °C to 280 °C, while heat was recovered between 110 °C and 60 °C. The second case study was a spray dryer for milk powder production in which an air stream was heated up from 64 °C to 210 °C with a capacity of 8.2 MW, while a heat source at 50 °C was recovered.

Both technologies were evaluated in both cases for a set of economic boundary conditions. Three economic scenarios were considered that corresponded to the fuel cost in Norway, Germany and Denmark in 2020 and one scenario was considered corresponding to the acquisition and operation of own renewables.

Results

The heat pump systems were designed and optimized for both case studies. Table 1 shows the COP and the total capital investment TCI for both cases and both technologies. It may be seen that the COP for the cascade system was estimated to be 1.9 in both cases, while it was 1.7 for the reversed Brayton cycle in the alumina production and 1.6 in the spray dryer case. The investment cost were relatively similar for the two technologies, while the economy of scale yielded considerably lower specific investment cost for the alumina production.

Table 1: COP and Total capital investment TCI for both cases and cycles [1]

	Alumina production		Spray dryer	
	Cascade multi-stage system	Reversed Brayton cycle	Cascade multi-stage system	Reversed Brayton cycle
Coefficient of performance COP, -	1.92	1.72	1.92	1.61
Total capital investment TCI, Mio. €	47.3	48.3	16.4	15.4
Specific total capital investment TCI_{spec}, €/kW	946	966	1,997	1,868

Figure 3 shows the levelized cost of heat for both technologies and both case studies for all economic scenarios and compares them to the alternative heat supply technologies. The levelized cost of heat is divided into the contributions accounting for the investment, the fuel cost and an exemplifying CO_2 tax of 50 €/ton to indicate the impact of a potential tax. In the case of the alumina production, the levelized cost of heat reaches as low as 31 €/MWh to 33 €/MWh under consideration of own renewable electricity facilities, while it is between 44 €/MWh and 46 €/MWh for the spray dryer case. In the spray dryer case, the heat pump-based solutions are competitive with a biomass boiler and a natural gas boiler under consideration of the assumed CO_2 tax. In the alumina production case, the lowest levelized cost of heat are obtained for the heat pump systems.

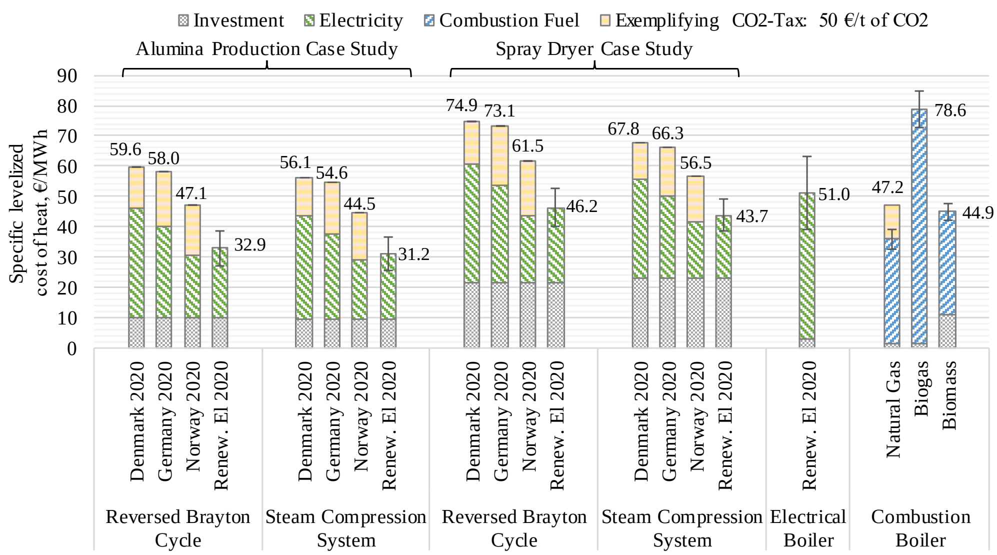

Figure 3: Specific levelized cost of heat c_h for both case studies including the reversed Brayton cycle, the multi-stage steam compression cycle, an electrical boiler and combustion-based boiler using natural gas, biogas and biomass. The cost scenarios are as defined in [1] *while the ranges for the cost for electricity from renewables, natural gas, biogas and biomass are indicated by the black bars* [1]

Conclusions

The study analyzed a reversed Brayton cycle and a cascade multi-stage steam compression for large-scale process heat supply at temperatures above 150 °C. It was pointed out that these temperatures might be reached by components from oil- and gas industries and that low electricity prices, as typically accessible for energy intensive industries or obtainable from acquiring and operating own renewable facilities, may improve the economic performance considerably. The levelized cost of heat for the heat pump-based systems were competitive to the biomass boilers and natural gas boilers for the spray dryer case study and outperformed both for the alumina production case study. This study has accordingly demonstrated, that heat pump systems are a viable alternative for process heat supply in industrial processes at temperatures of up to 280 °C.

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Analysis of technologies and potentials for heat pump-based process heat supply above 150 °C
09.09.2019 – Copenhagen, Denmark
2nd conference on high-temperature heat pumps
Benjamin Zühlsdorf, F. Bühler, M. Bantle, B. Elmegaard
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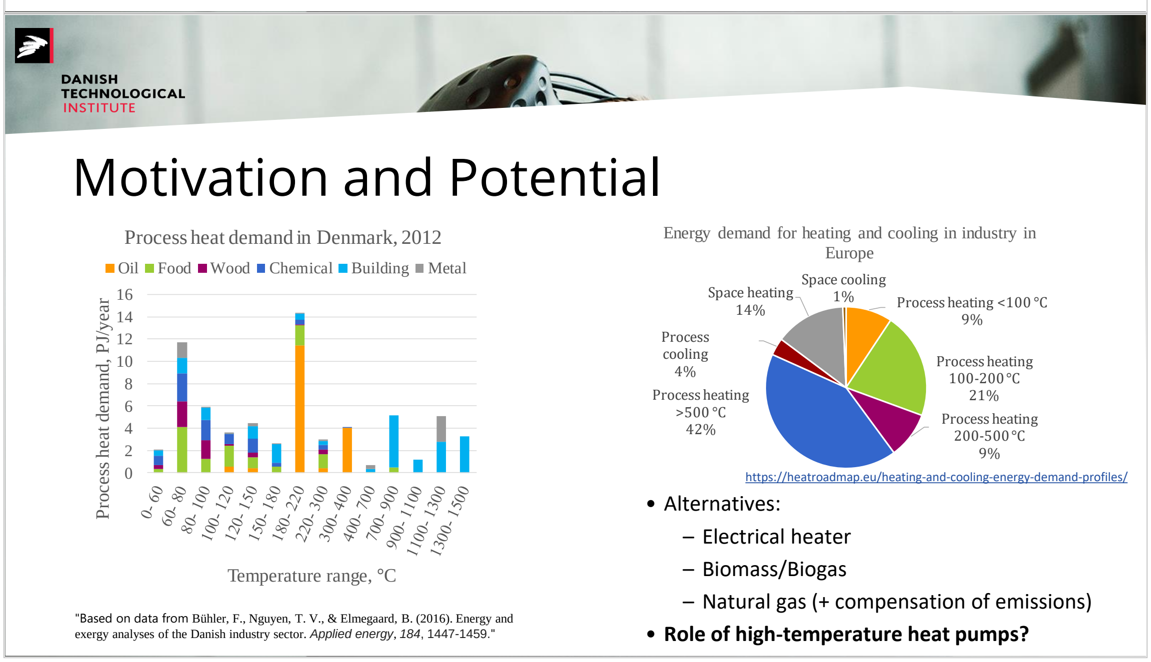
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Motivation and Potential
Process heat demand in Denmark, 2012
Oil Food Wood Chemical Building Metal
Process heat demand, PJ/year
16
14
12
10
8
6
4
2
0
0- 60
60- 80
80- 100
100- 120
120- 150
150- 180
180- 220
220- 300
300- 400
400- 700
700- 900
900- 1100
1100- 1300
1300- 1500
Temperature range, °C
"Based on data from Bühler, F., Nguyen, T. V., & Elmegaard, B. (2016). Energy and exergy analyses of the Danish industry sector. Applied energy, 184, 1447-1459."
Energy demand for heating and cooling in industry in Europe
Space cooling 1%
Space heating 14%
Process heating <100 °C 9%
Process cooling 4%
Process heating 100-200 °C 21%
Process heating >500 °C 42%
Process heating 200-500 °C 9%
https://heatroadmap.eu/heating-and-cooling-energy-demand-profiles/
• Alternatives:
– Electrical heater
– Biomass/Biogas
– Natural gas (+ compensation of emissions)
• Role of high-temperature heat pumps?

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Motivation and Potential

Challenges for HTHPs
- Limited performance (COP_{Lor})
- High investment cost → Economic performance
- Component constraints

Motivation for electrification: Changing boundary conditions
- Decreasing LCOE from renewables
- Cost of emission increasing
- Limitations of biomass/biogas
- Political/industrial strategies to become carbon neutral

Possibilities
- Components from e.g., oil & gas industries operate in more challenging conditions (up to >400 °C)
- Combination of heat pumps and own renewable electricity utilities

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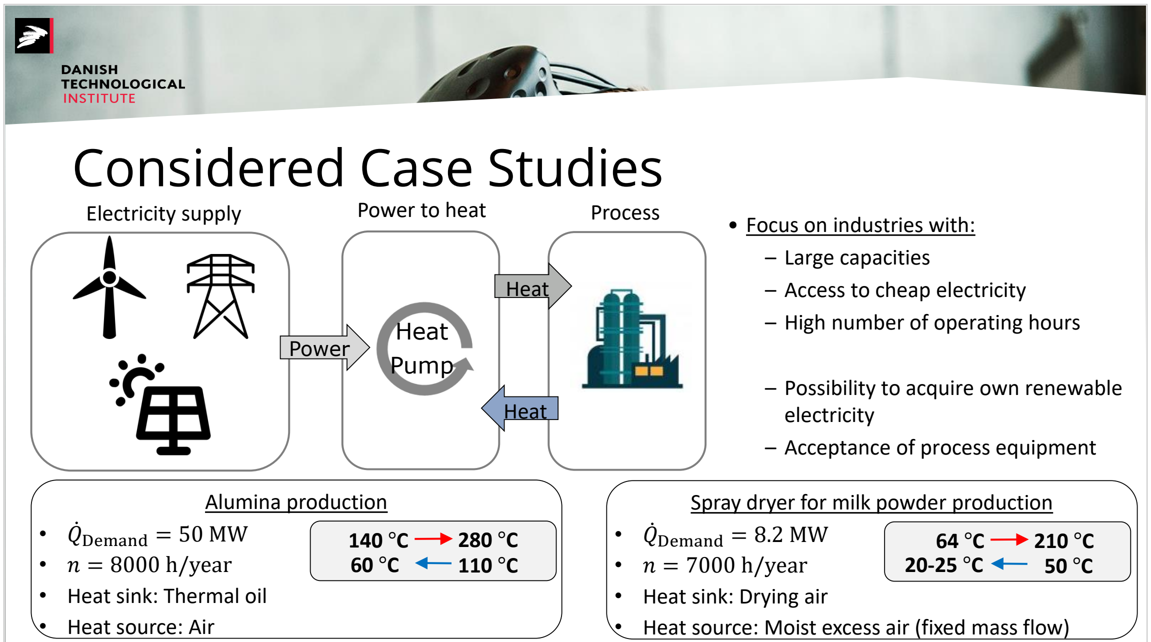
Agenda

- Considered case studies
  - Alumina production case study
  - Spray dryer case study
- Technical concepts
  - Cascade multi-stage compression cycle (R718)
  - Reversed Brayton cycle (R744)
- Economic Analysis
- Summary and outlook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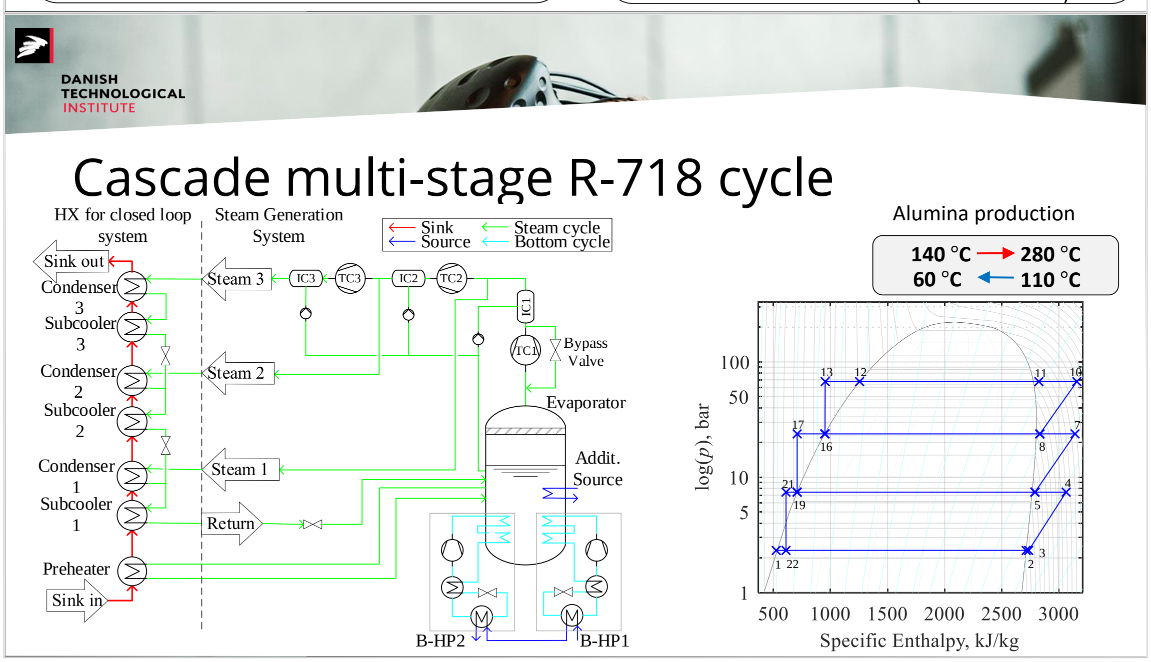
Considered Case Studies
Electricity supply
Power to heat
Process
Power
Heat
Pump
Heat
Heat
• Focus on industries with:
– Large capacities
– Access to cheap electricity
– High number of operating hours
– Possibility to acquire own renewable electricity
– Acceptance of process equipment
Alumina production
• $\dot{Q}_{\text{Demand}} = 50$ MW
• $n = 8000$ h/year
• Heat sink: Thermal oil
• Heat source: Air
140 °C → 280 °C
60 °C ← 110 °C
Spray dryer for milk powder production
• $\dot{Q}_{\text{Demand}} = 8.2$ MW
• $n = 7000$ h/year
• Heat sink: Drying air
• Heat source: Moist excess air (fixed mass flow)
64 °C → 210 °C
20-25 °C ← 50 °C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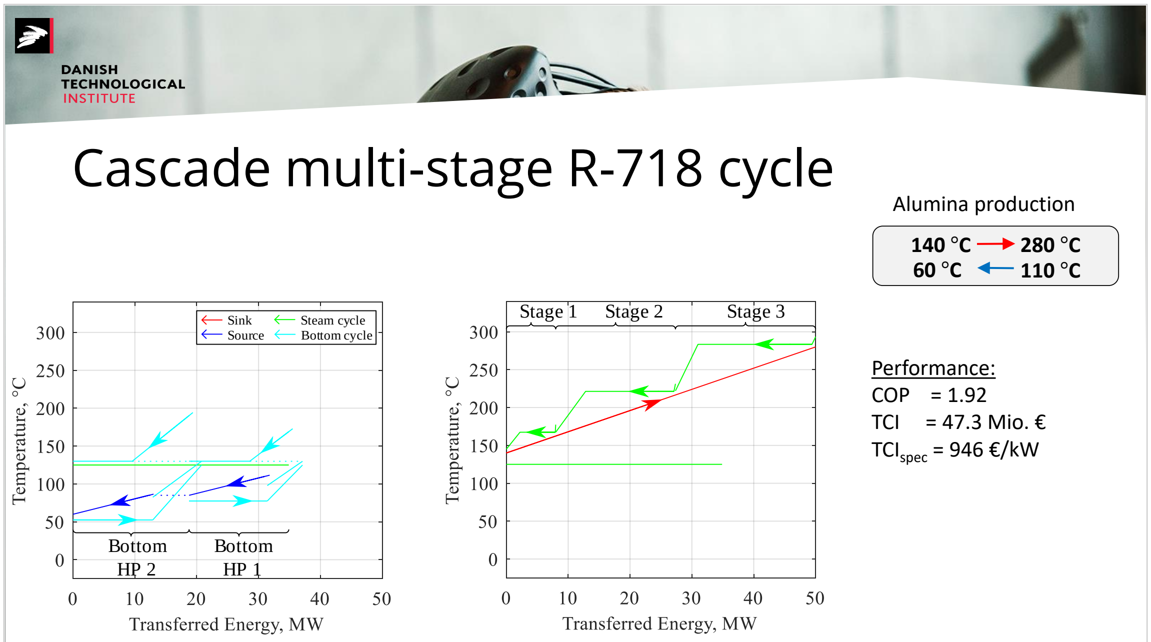
Cascade multi-stage R-718 cycle
HX for closed loop system
Steam Generation System
Sink
Source
Steam cycle
Bottom cycle
Sink out
Condenser 3
Subcooler 3
Condenser 2
Subcooler 2
Condenser 1
Subcooler 1
Preheater
Sink in
Steam 3
Steam 2
Steam 1
Return
IC3
TC3
IC2
TC2
IC1
TC1
Bypass Valve
Evaporator
Addit. Source
B-HP2
B-HP1
Alumina production
140 °C → 280 °C
60 °C ← 110 °C
log(p), bar
Specific Enthalpy, kJ/kg
1
5
10
50
100
500
1000
1500
2000
2500
3000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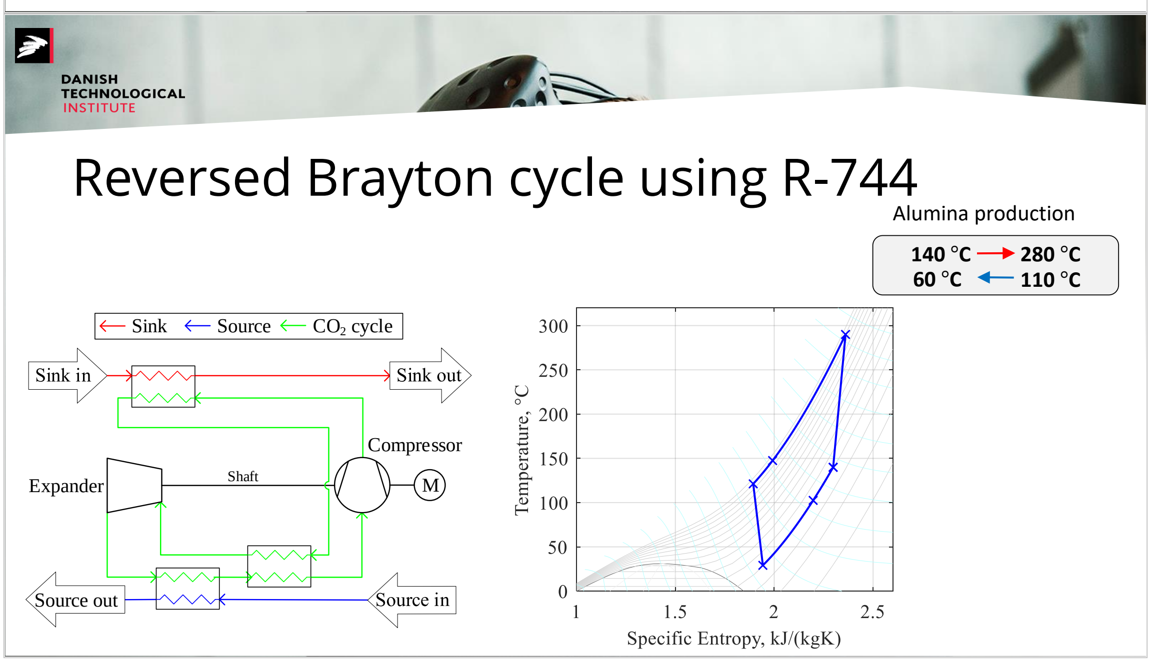
Cascade multi-stage R-718 cycle
Alumina production
140 °C → 280 °C
60 °C ← 110 °C
Sink
Source
Steam cycle
Bottom cycle
Bottom HP 2
Bottom HP 1
Stage 1
Stage 2
Stage 3
Temperature, °C
Transferred Energy, MW
Performance:
COP = 1.92
TCI = 47.3 Mio. €
TCI$_{spec}$ = 946 €/kW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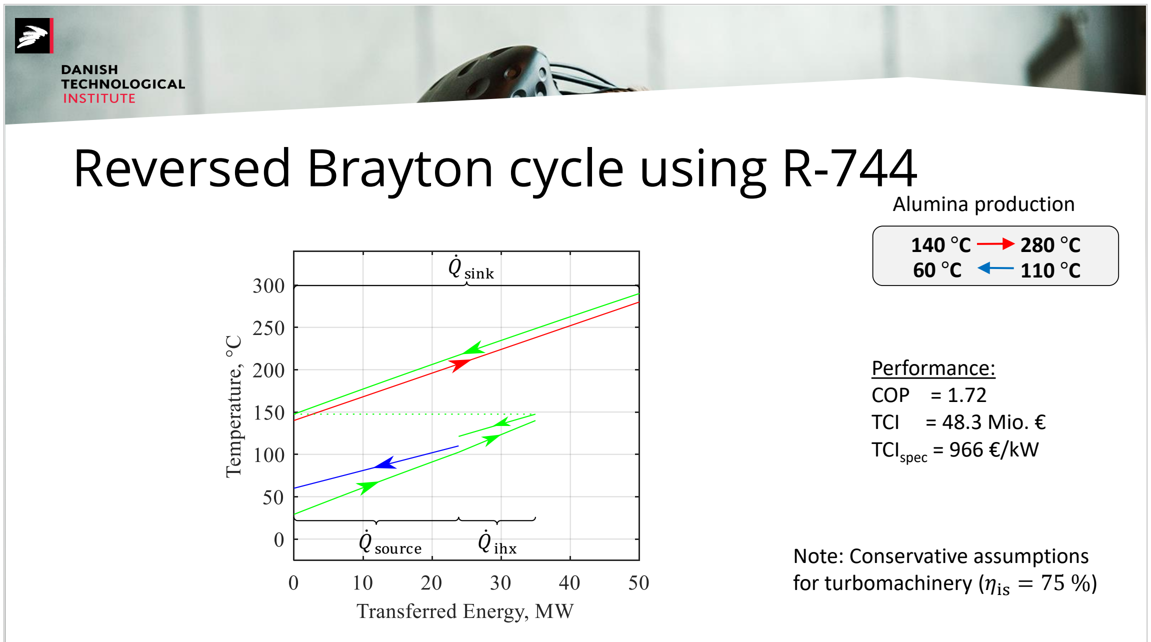
Reversed Brayton cycle using R-744
Alumina production
140 °C → 280 °C
60 °C ← 110 °C
Sink
Source
CO_2 cycle
Sink in
Sink out
Compressor
Expander
Shaft
M
Source out
Source in
Temperature, °C
Specific Entropy, kJ/(kgK)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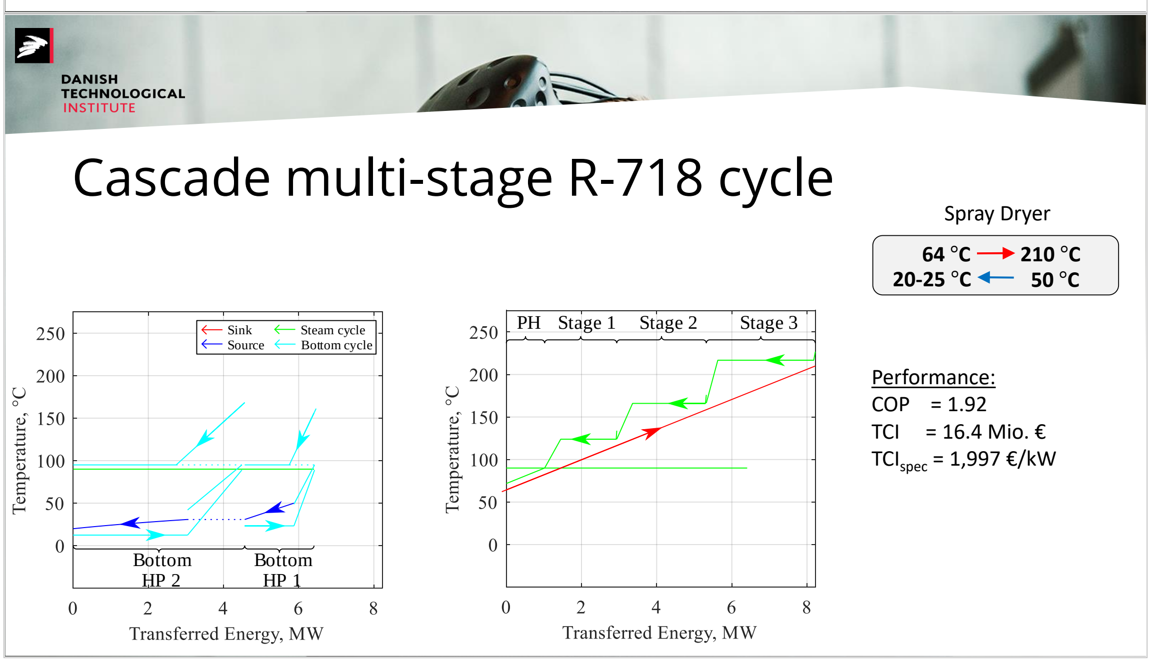
Reversed Brayton cycle using R-744
Alumina production
140 °C → 280 °C
60 °C ← 110 °C
$\dot{Q}_{sink}$
$\dot{Q}_{source}$
$\dot{Q}_{ihx}$
Temperature, °C
Transferred Energy, MW
0 10 20 30 40 50
0 50 100 150 200 250 300
Performance:
COP = 1.72
TCI = 48.3 Mio. €
TCI_{spec} = 966 €/kW
Note: Conservative assumptions
for turbomachinery (η_{is} = 75 %)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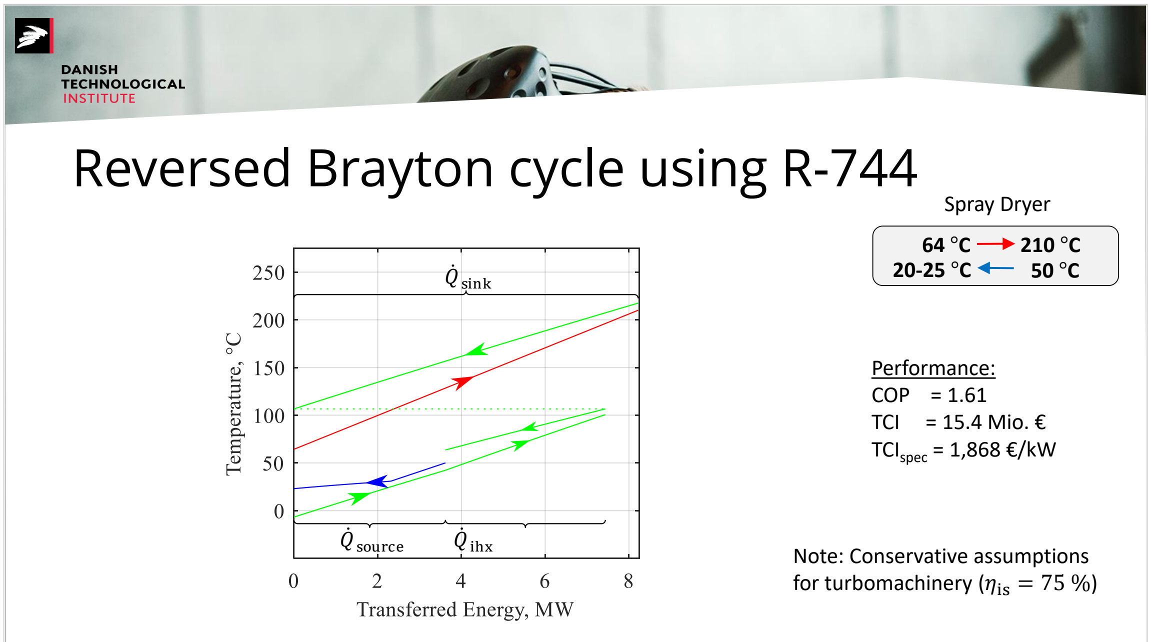
Cascade multi-stage R-718 cycle
Spray Dryer
64 °C → 210 °C
20-25 °C ← 50 °C
Sink
Source
Steam cycle
Bottom cycle
Bottom
HP 2
Bottom
HP 1
PH
Stage 1
Stage 2
Stage 3
Temperature, °C
Transferred Energy, MW
0 2 4 6 8
0 50 100 150 200 250
Performance:
COP = 1.92
TCI = 16.4 Mio. €
TCI_{spec} = 1,997 €/kW


DANISH
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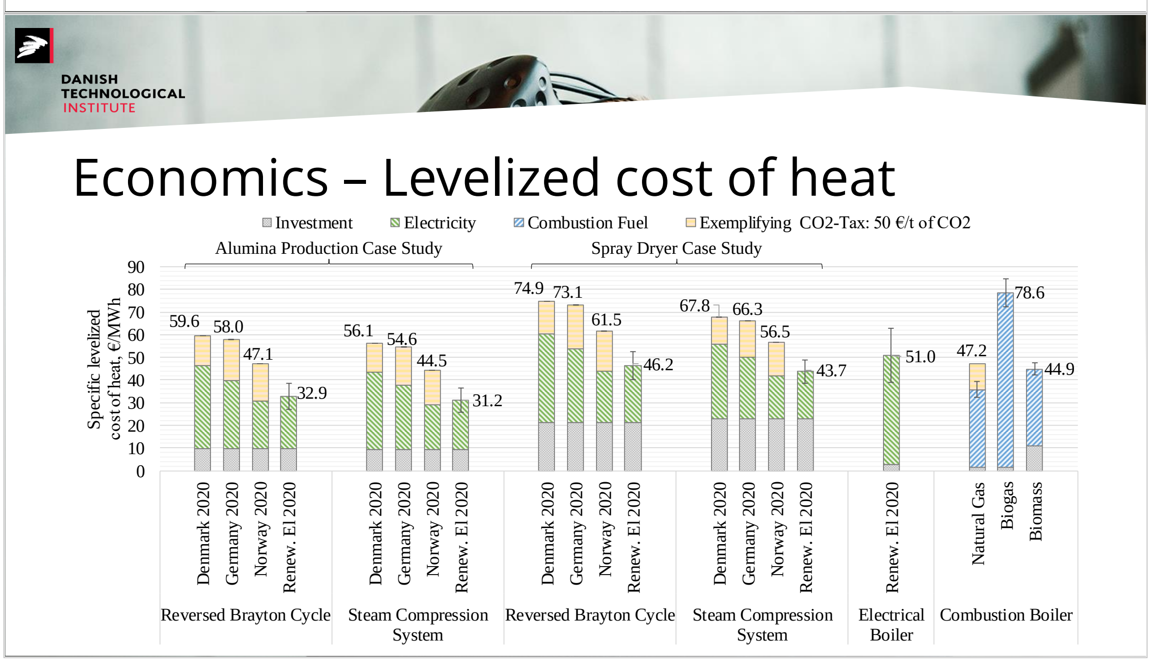
Reversed Brayton cycle using R-744
Spray Dryer
64 °C → 210 °C
20-25 °C ← 50 °C
Performance:
COP = 1.61
TCI = 15.4 Mio. €
TCI$_{spec}$ = 1,868 €/kW
Note: Conservative assumptions for turbomachinery (η_{is} = 75 %)
Temperature, °C
Transferred Energy, MW
$\dot{Q}_{sink}$
$\dot{Q}_{source}$
$\dot{Q}_{ihx}$
0
50
100
150
200
250
0
2
4
6
8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Economics – Levelized cost of heat
Investment
Electricity
Combustion Fuel
Exemplifying CO2-Tax: 50 €/t of CO2
Alumina Production Case Study
Spray Dryer Case Study
Specific levelized cost of heat, €/MWh
0
10
20
30
40
50
60
70
80
90
59.6
58.0
47.1
32.9
56.1
54.6
44.5
31.2
74.9
73.1
61.5
46.2
67.8
66.3
56.5
43.7
51.0
47.2
78.6
44.9
Denmark 2020
Germany 2020
Norway 2020
Renew. El 2020
Natural Gas
Biogas
Biomass
Reversed Brayton Cycle
Steam Compression System
Reversed Brayton Cycle
Steam Compression System
Electrical Boiler
Combustion Boiler

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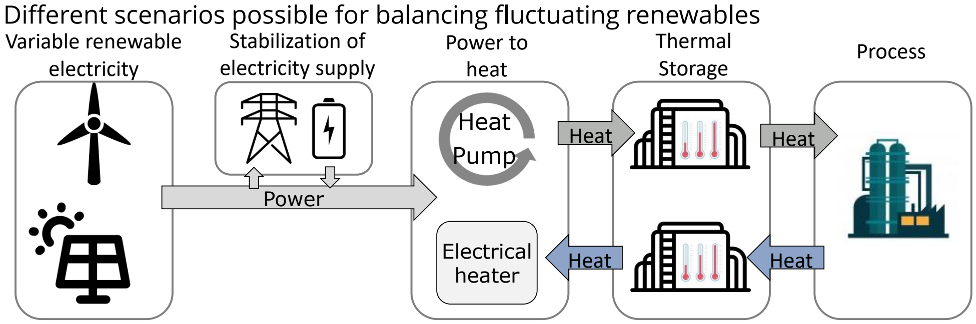
Discussion

- Different scenarios possible for balancing fluctuating renewables



- Large investments feasible?

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Conclusions

- Technical feasibility:
  - Supply temperatures of up to 280 °C analyzed
  - COPs between 1.6 to 1.9
- Economic feasibility:
  - Economy of scale → lower specific investment at increasing capacities
  - Levelized cost of heat strongly dependent on electricity cost
  - **Heat pump solutions competitive to natural gas and biomass boilers**
- Future work and potentials:
  - Component optimization
  - Optimization of investment cost
  - Demonstration

Assessing High Temperature Heat Pumps Market

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Keywords:
High temperature heat pump, industry, waste heat recovery, market assessment

Abstract

Introduction

As reducing the environmental impacts of human activities has become a global priority, greenhouse gas emissions come as one of the top target to mitigate climate change. EDF R&D wants to play a key role in improving the overall EDF Group's performance regarding these objectives by promoting and working on disruptive technologies such as high temperature heat pumps to prepare the future of energy.

Therefore, technologies such as heat pumps have been subjects to many studies [1-4] because they allow for both energy waste reduction and renewable energy production.

This presentation proposes a method to evaluate the potential of heat recovery with heat pumps (HP) in France, for each industrial sector and for three level of temperature of the heat provided by the heat pumps.

Methods

This method to assess the HP market in France combines thermal equipments in operation database with a process energy needs characterization database. An expertise of industrial sectors and processes from EDF completes it. The CEREN (Centre d'Etudes et de Recherches Economiques sur l'Energie) conducts studies on the energy consumption of the French industry. Data is structured by 130 industrial sectors (steel, food, chemical...), by energy types (electricity, natural gas, fuel...) and by energy uses (furnaces, boilers, dryers ...). The database regarding industry is built from about thousand investigations on the largest industrial sites. It is then completed with over six thousand annual phone surveys on the small and medium-size enterprises. About seventy specific questionnaires have been filled to evaluate the energy contained in the fumes of the furnaces and sixty others to evaluate energy contained in the steam of dryers.

The waste heat is characterized by its energy value. This energy value is given by type of warm effluent, by temperature level, by availability of heat and by industrial sector. Energy data are analysed on the basis of these parameters.

The Process needs are characterized by their consumption, number and installed power. These values are given by type of equipment, by industrial function, by technology, by power level, by age, by temperature level, by running time, and by industrial sector. Data are analysed on the basis of these parameters.

Three types of heat pumps HP 1, 2 and 3 are considered, respectively providing heat up to 70°C, up to 100°C and up to 150°C. This three types of HP allow to connect a waste heat in which energy is recovered to a heat need to feed. Because we want the HP to have a COP (Coefficient of Performance) of 4 at least, the difference in temperature between the heat waste and the heat needs must stay around 40°C.

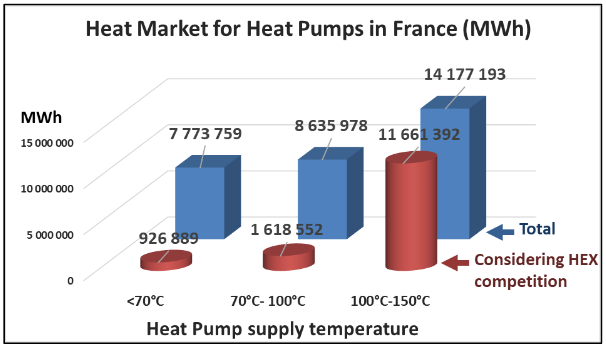
Considering for each sectors both the waste heat and heat needs amount at specific temperature ranges, we can assess to potential of heat production by each heat pumps : HP 1, 2, and 3.

This methods was detailed in [5]

Then, we considered that available waste heat at a temperature range higher than the temperature range of the heat needs could be used through heat exchangers to produce that heat needs. This assumption was then taken into account to revise the HP potential and find its lower limit corresponding to a 100% used of heat exchangers prior to the use of HP.

Results and Discussion

We could assess the HP potential in France (see figure below) that lies between the upper value (total of 31 TWh) and the lower value that considers the use of heat exchanger (total of 14 TWh).



Conclusion and perspective

It is interesting to notice that the HP potential at high temperature (>100°C) is both much more important than at lower temperature and also less sensitive to the competition of heat exchanger. The recent efforts to bring such heat pumps to the markets should help in improving the HP market penetration and the expected CO2 emissions reduction.

This method and these results could be successfully extended to other countries. As industrials processes are very similar from one country to another, as the heat needs are mainly powered by steam through gas or coal and as the present market share of HP in industry is low, we could extend these results to other countries, sector by sector, using relevant key point of comparison such as the volume of production or the energy consumption of the given sector and country.

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High Temperature Heat Pumps market assessment in industry sectors

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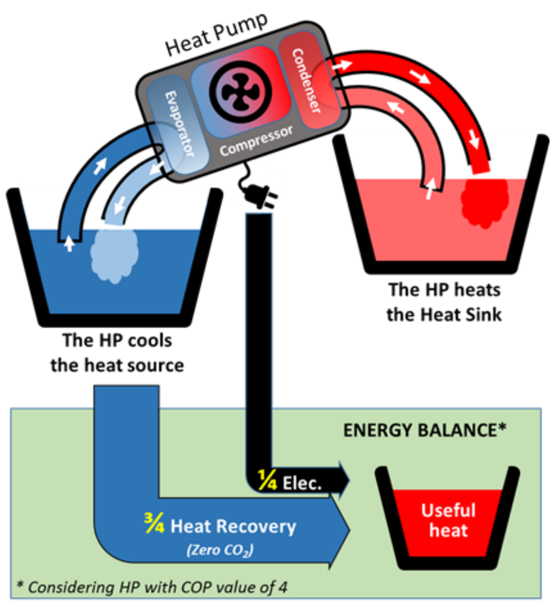
JM FOURMIGUE – EDF R&D

1

Heat Pumps at a glance

- The source of energy is cooled by the evaporator
- The heat pump transfers energy from the cold source (waste heat in this study) to the heat sink (condenser side)
- In this study, the temperature difference between the heat source and the heat sink is kept around 40°C
- The Coefficent Of Performance (COP) is then about 4 which ensures a good economical balance.



Energy data bases from CEREN

"heat needs" and "waste heat"

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3

French Energy databases – Players and method

- ❑ Provided by the CEREN (Centre d'Etudes et de Recherches Economiques sur l'Energie)
- ❑ Financed by French Energy Player (EDF, ENGIE) and ADEME (French Environnemental Agency)
- ❑ Studies on the energy consumption in France for all sectors

For industry, data is structured

- ✓ by industrial sectors (steel, food, chemical...),
- ✓ by energy types (electricity, natural gas, fuel...)
- ✓ by energy uses (furnaces, boilers, dryers ...).

The industry database is built :

- ✓ From about thousand investigations on the largest industrial sites.
- ✓ Completed with over six thousand annual phone surveys on the SME.
- ✓ About seventy specific questionnaires to evaluate the energy in the fumes of the furnaces
- ✓ And sixty others to evaluate energy contained in the steam of dryers.

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French Energy databases – Waste heat origin

Waste heat is characterized by
- ✓ energy value (MWh)
- ✓ type of effluent (fumes, water, mist)
- ✓ temperature level
- ✓ availability of heat
- ✓ industrial sector

Included warm effluents are:
- combustion gases from furnaces (except blast furnaces),
- combustion gases from boilers
- steam from dryers
- cooling fluids from air compressors
- cooling fluids from refrigeration compressors
- cooling fluids from refrigeration condensers
- cooling fluids from heat exchangers in desuperheaters of refrigeration groups
- hot water from clean-in-place (CIP effluents)

Not included effluents are not
- concentrators
- network of fluids of cooling
- energy contained inside the products

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French Energy databases – Heat needs description

Process energy needs characterization:

- ✓ Consumption per year
- ✓ number and installed power capacity
- ✓ type of equipment (boiler, furnace, …)
- ✓ industrial function (cooking, drying, annealing, …)
- ✓ Technology (Induction, resistance, direct gas, …)
- ✓ Age
- ✓ Temperature level
- ✓ Running time per year
- ✓ Industrial sector

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French Energy databases – 130 sectors for industry : quasi process scale

Exemple :

Food industry is divided into 17 sectors

- Breweries
- Chocolate, sweet
- Cider, waters, fruits juices, sodas
- Pastas
- Vegetables and fruits
- Fatty substances
- Slautherhouses
- Fish
- Dairy
- Malt
- Flour
- Meat based products
- Alcoolic beverages
- Salt
- Starch products
- Spirits, champagne, vermouths, winemaking
- Sugar

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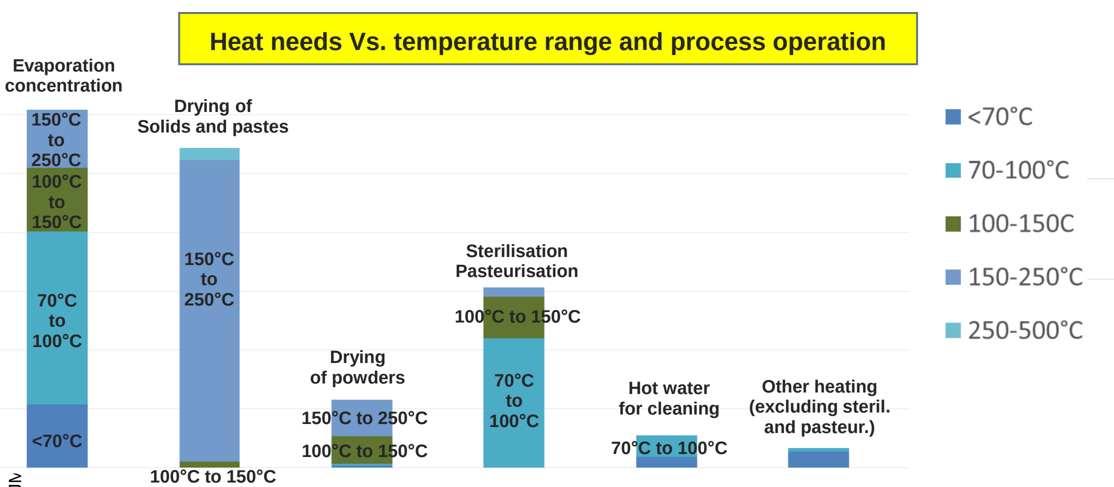
Exemple of the milk Industry



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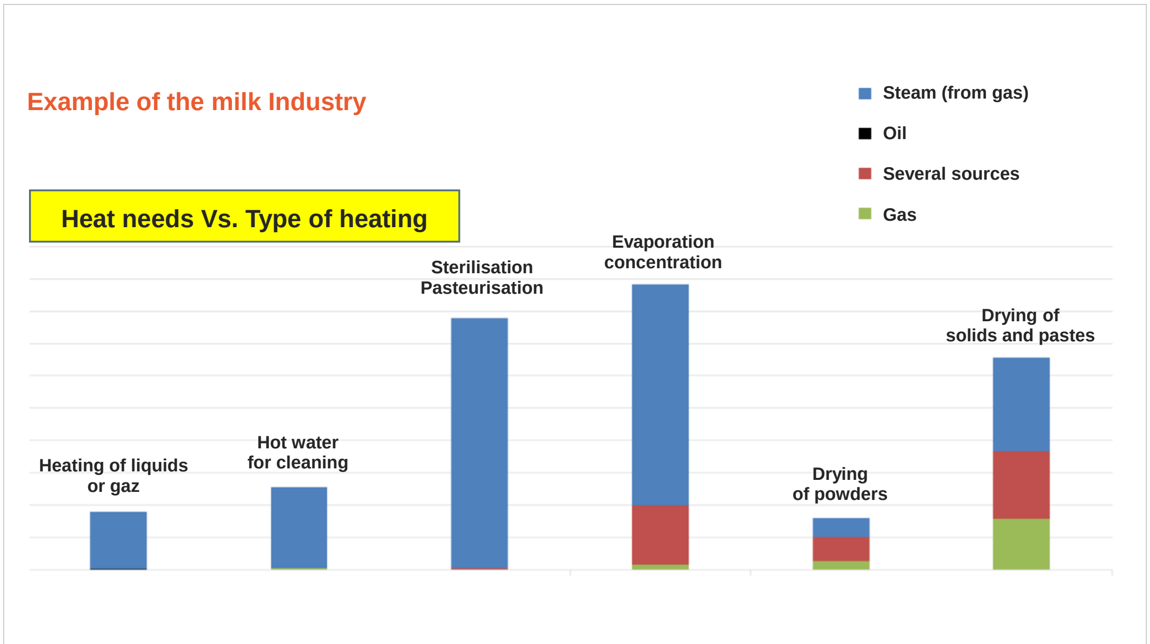
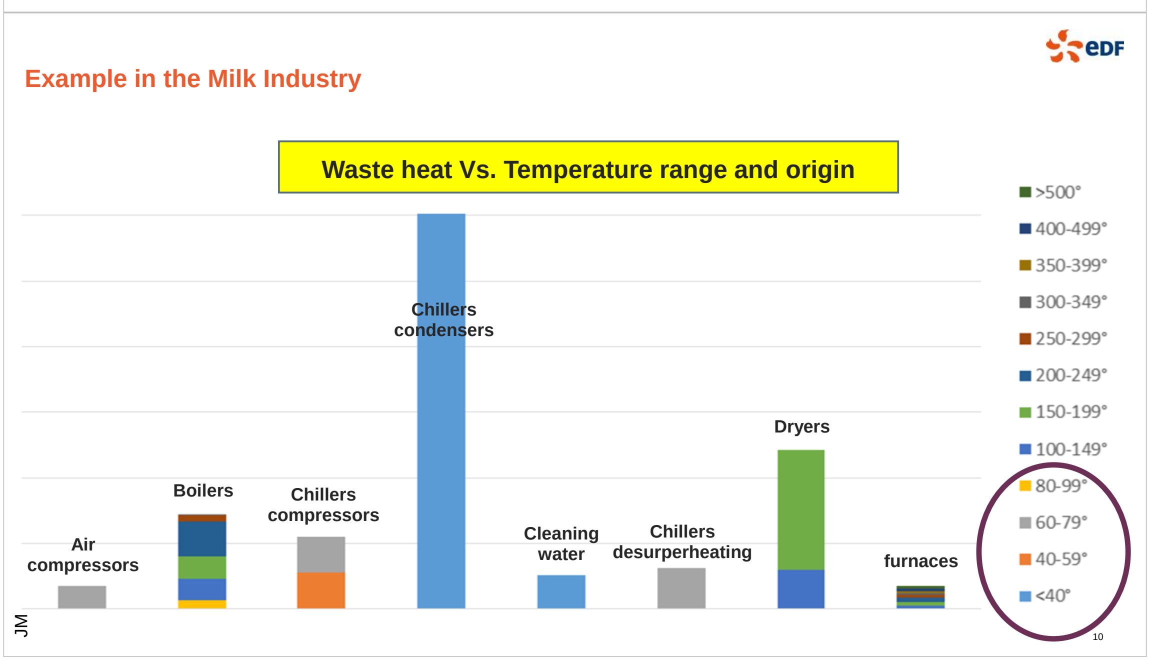
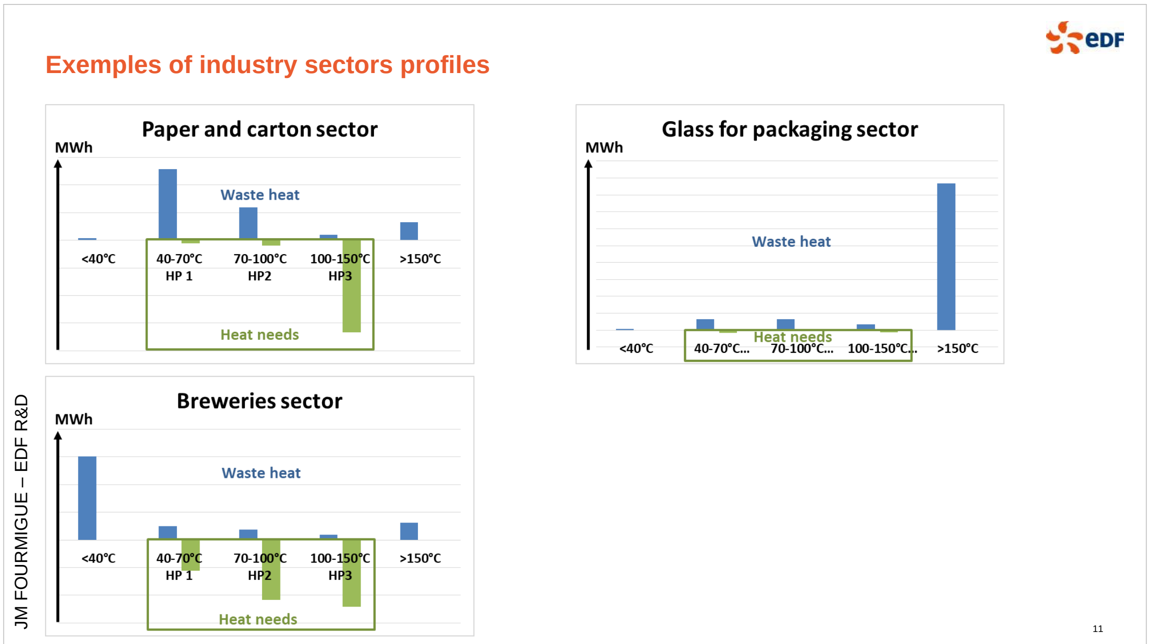

Example of the milk Industry
Heat needs Vs. Type of heating
Steam (from gas)
Oil
Several sources
Gas
Heating of liquids or gaz
Hot water for cleaning
Sterilisation Pasteurisation
Evaporation concentration
Drying of powders
Drying of solids and pastes


Example in the Milk Industry
Waste heat Vs. Temperature range and origin
>500°
400-499°
350-399°
300-349°
250-299°
200-249°
150-199°
100-149°
80-99°
60-79°
40-59°
<40°
Air compressors
Boilers
Chillers compressors
Chillers condensers
Cleaning water
Chillers desurperheating
Dryers
furnaces
JM
10


Exemples of industry sectors profiles
Paper and carton sector
MWh
Waste heat
<40°C
40-70°C
HP 1
70-100°C
HP2
100-150°C
HP3
>150°C
Heat needs
Glass for packaging sector
MWh
Waste heat
<40°C
40-70°C...
Heat needs
70-100°C...
100-150°C...
>150°C
Breweries sector
MWh
Waste heat
<40°C
40-70°C
HP 1
70-100°C
HP2
100-150°C
HP3
>150°C
Heat needs
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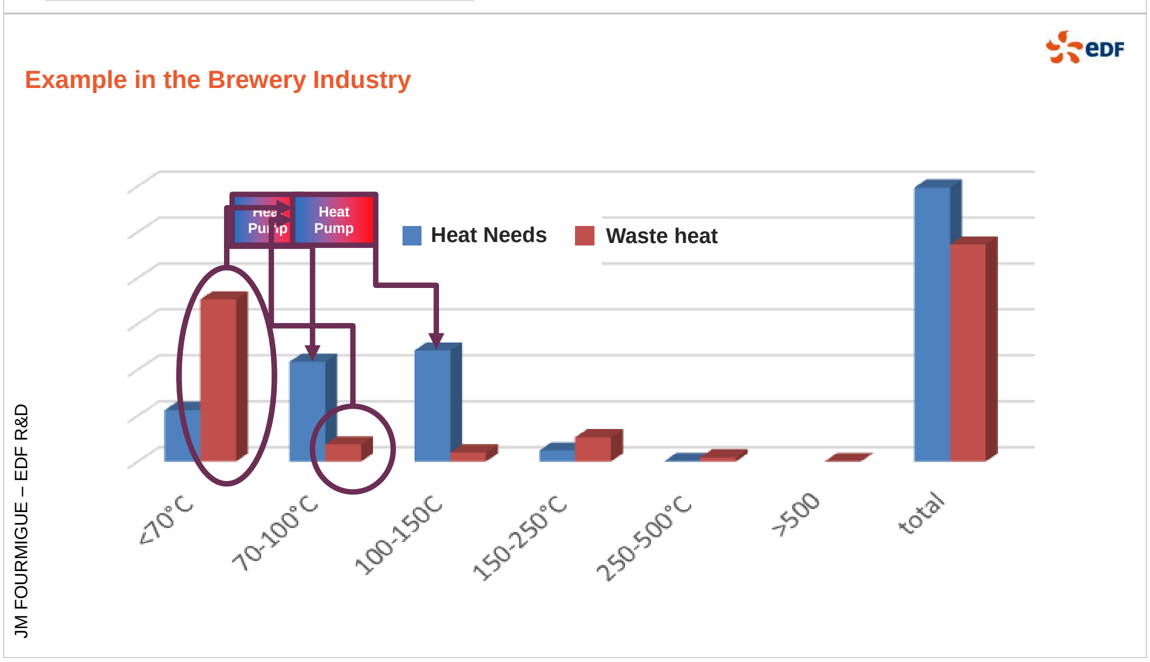
Example in the Brewery Industry
Heat
Pump
Heat Needs
Waste heat
<70°C
70-100°C
100-150C
150-250°C
250-500°C
>500
total
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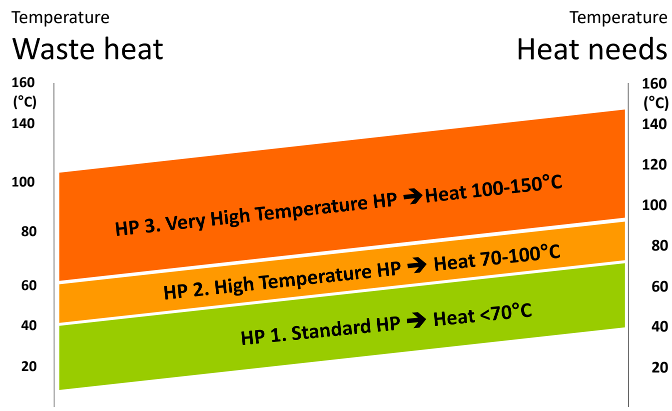
Heat Pumps segmentation for the study

This segmentation have to fit with the temperature intervals of the energy databases



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Key points to target the HP's opportunities in industry

Key point 1 **Overall potential of heat pumps for each industry sector**

Key point 2 **In each industry sector, competition between Heat Exchangers and HP's**

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Key point 1: Potential of energy that can be produced by a heat pump

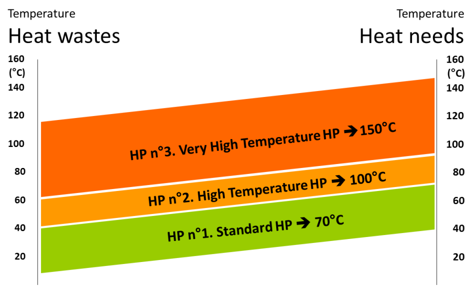
Method : The energy production of HP is limited by

- either the energy needs
- or the energy wasted of the industry sector

HP 3 = min (Heat Waste $_{[60\text{-}100°C]}$; Heat Needs $_{[100\text{-}150°C]}$)

HP 2 = min (Heat Waste $_{[40\text{-}70°C]}$; Heat Needs $_{[70\text{-}100°C]}$)

HP 1 = min (Heat Waste $_{[<50°C]}$; Heat Needs $_{[<70°C]}$)



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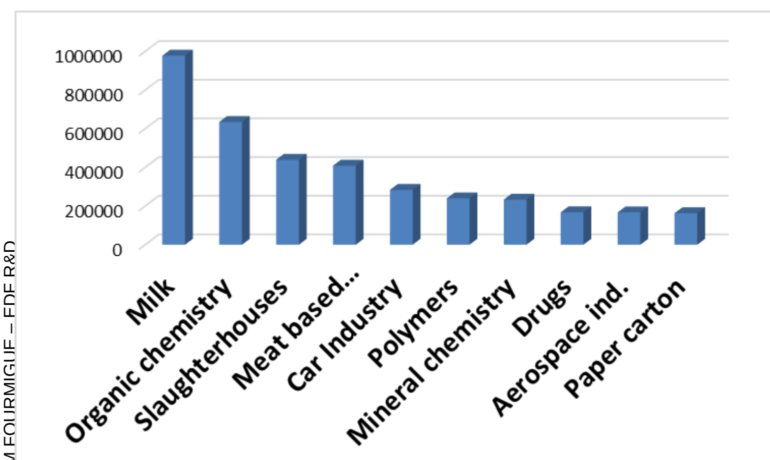
Key point 1

Overall Potential (France) for HP 1 (heat up to → 70°C)



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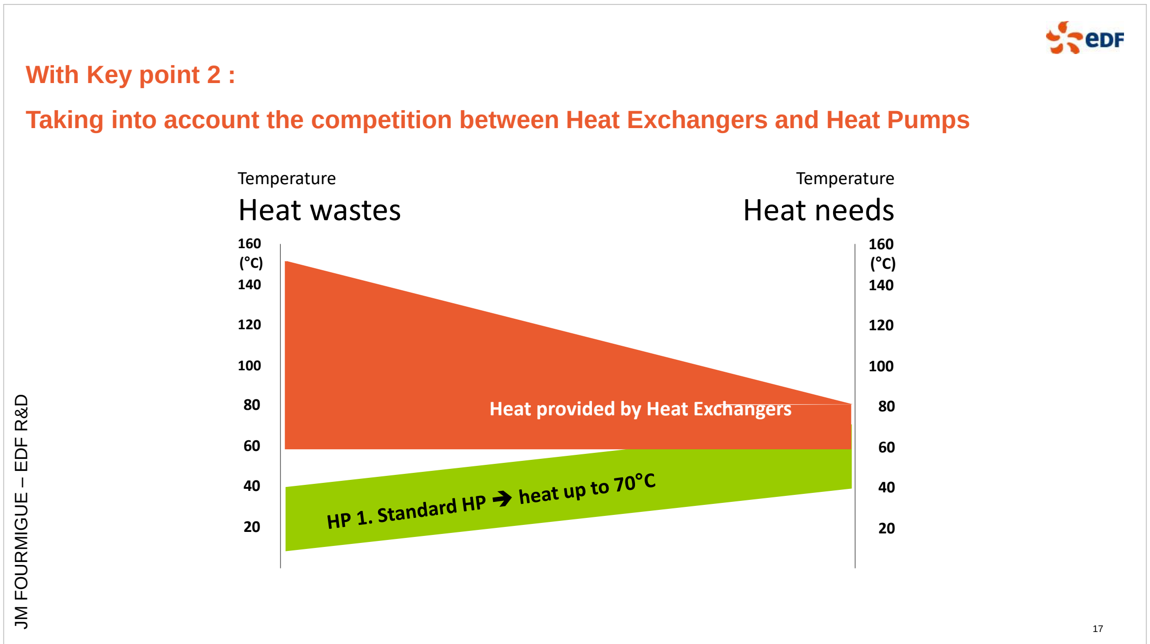
eDF
With Key point 2 :
Taking into account the competition between Heat Exchangers and Heat Pumps
Temperature
Heat wastes
Temperature
Heat needs
160 (°C) 140 120 100 80 60 40 20
160 (°C) 140 120 100 80 60 40 20
Heat provided by Heat Exchangers
HP 1. Standard HP ➔ heat up to 70°C
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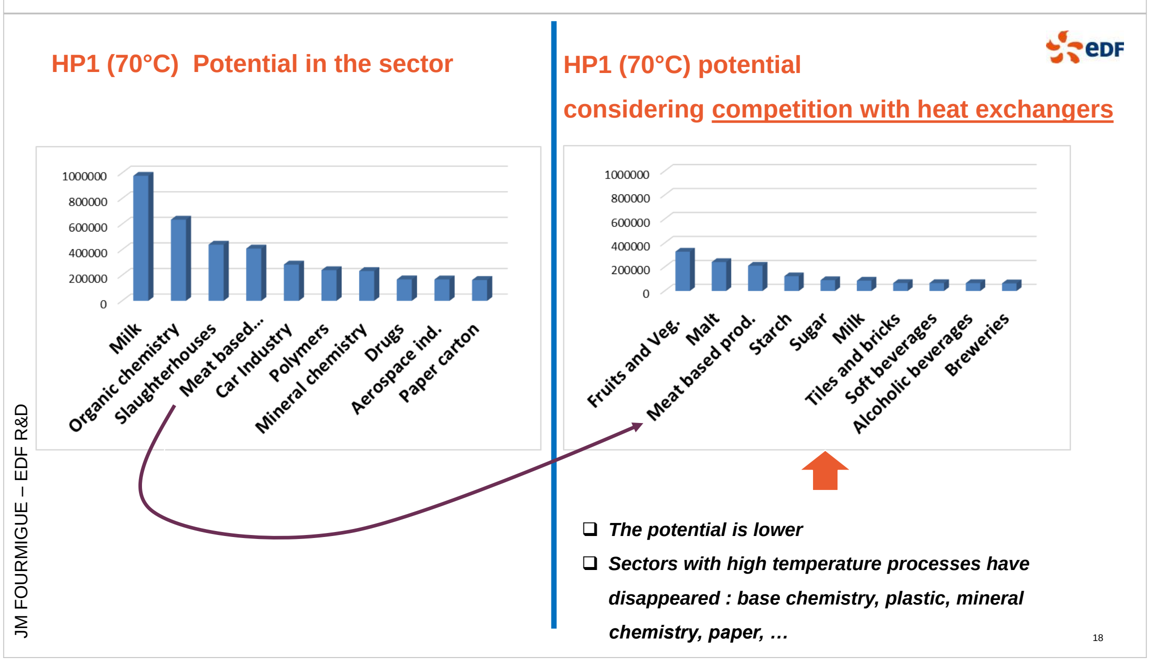
HP1 (70°C) Potential in the sector
1000000 800000 600000 400000 200000 0
Milk
Organic chemistry
Slaughterhouses
Meat based...
Car Industry
Polymers
Mineral chemistry
Drugs
Aerospace ind.
Paper carton
HP1 (70°C) potential
considering competition with heat exchangers
eDF
1000000 800000 600000 400000 200000 0
Fruits and Veg.
Malt
Meat based prod.
Starch
Sugar
Milk
Tiles and bricks
Soft beverages
Alcoholic beverages
Breweries
The potential is lower
Sectors with high temperature processes have disappeared : base chemistry, plastic, mineral chemistry, paper, …
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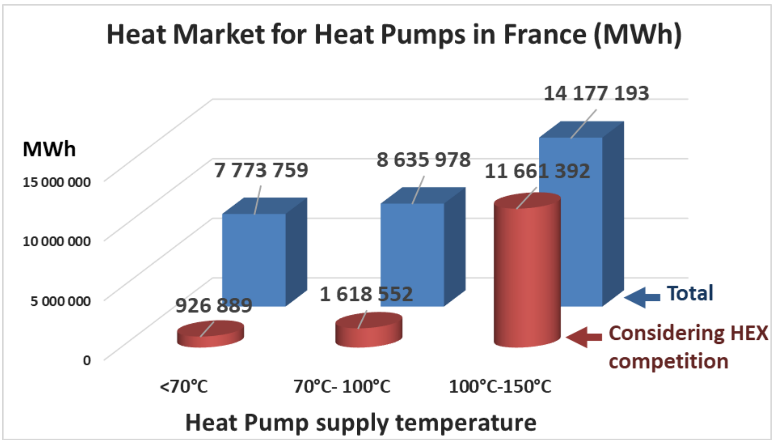
Industrial Heat Pumps potential in France

Total potential : 31 TWh

Potential considering the use of heat exchanger : 14 TWh



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How extending this analysis to other European countries ?

Industry sectors	Milk	Malt	Meat	Car	Pet food	Paper	Chemistry	Sugar
Size in France (unit to be defined)	100	20	15	200	20	60	40	10
Size in other country (unit to be defined)	50	…	…	…	…	…	…	…
HP 1, 2, 3 market in France	1 TWh	2 TWh	1 TWh	4 TWh	1 TWh	3 TWh	5 TWh	2 TWh
HP market other country	0,5 TWh	…	…	…	…	…	…	…

Data calculated using the data of the french market

Data to be found
Eg. Production in litersof milk /year

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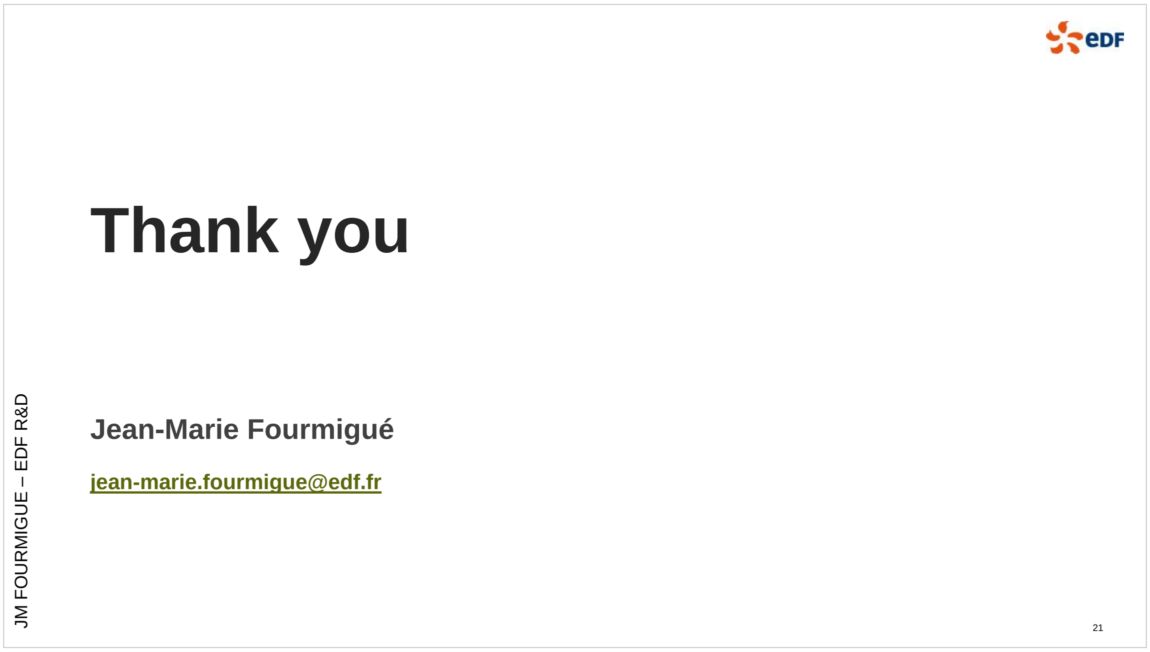
Thank you
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The potential of heat pumps in the electrification of the Danish industry

Fabian Bühler[1], *Benjamin Zühlsdorf*[2], *Fridolin Müller Holm*[3] *and Brian Elmegaard*[1]

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Keywords:
High temperature heat pump, electrification, industry, Denmark

Introduction

Reaching the goals set by the Paris Agreement (UNFCCC, 2015) requires the energy sector to have net-zero CO_2-emissions the latest by 2060 (Philibert, 2017). The power sector changes from fossil fuels to renewable energy sources, providing increasing amounts of clean energy. The decarbonisation of the industry sector is however often overseen, despite the industry accounting for 21 % of the direct global greenhouse gas emissions in 2010 (IPCC, 2014). A decarbonisation of the industry can happen on a large scale following three main technology options, (i) the replacement of fossil fuels with bioenergy, (ii) the electrification of processes and (iii) the implementation of carbon capture and storage technologies (Åhman et al., 2012). Electrification of processes reduces energy-related CO_2-emissions, but it can also reduce the final energy use by integrating heat pumps (HP). The choice of Power-to-Heat technologies generally depend on process and temperature requirements. Some promising electrification technologies, such as high temperature heat pumps (HTHP) or heat pump-assisted distillation, have currently a low technology readiness level (den Ouden et al., 2017), while other available technologies, such as electric boilers and Mechanical Vapour Recompression (MVR), can be infeasible under current economic conditions. The potential for HTHPs was investigated for the European industry (Kosmadakis, 2019), where it was found that HTHP can cover about 1.5 % of the industries heat consumption.

This work derives an overview of the potential of heat pump-based process heat supply for the electrification of thermal processes in the Danish industry.

Energy use in the Danish manufacturing industry

In Denmark, the share of electricity in final energy use of the manufacturing industry has increased from 27.1 % in 1990 to 32.5 % in 2017 (Danish Energy Agency, 2018). The share of natural gas has increased in the same period from 31.3 % to 20.8 %, while the use of oil drastically decreased. The total share of fossil fuel directly used for heating in the industry was still around 50 % in 2017.

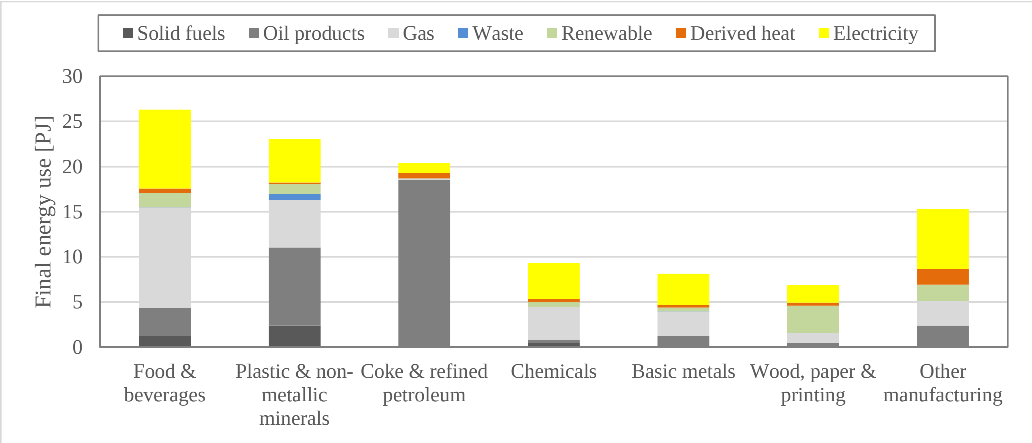

Figure 1: Total final energy use in the Danish industry by industrial sector and energy carrier in 2016 (Denmark Statistics, 2017)

In Figure 1 it can be seen that there are three main industrial sectors in terms of energy use, namely the food and beverage sector, mainly consisting of the production of dairy, meat, beverages and other food products. In the second most energy intense category, the manufacturing of concrete and bricks represents the highest share in terms of fuel use and the most energy intense sector overall with a total fuel use of 17 PJ in 2016.

Processes energy use in the Danish industry

In Figure 2 the process heat demand in the Danish manufacturing industry is shown by temperature level and process. It can be seen that below 100 °C heat is required amongst others for drying and distillation. Heat at higher temperatures is used for also for baking and melting. The peak between 180 °C and 220 °C originates from heating process in refineries and process heat supply. Figure 1

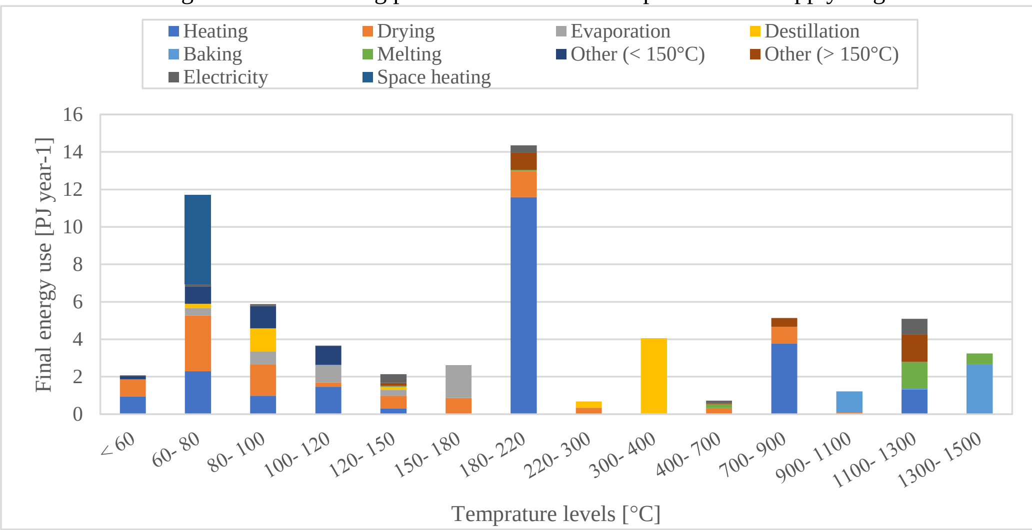

Figure 2: Final energy use of the 21 most energy intense manufacturing industries by temperature and process (Bühler et al., 2016a; Sørensen et al., 2015).

Heat pump potential for waste heat recovery in the Danish manufacturing industry

Figure 3 and 4 describe the amount of heat that could be supplied to industrial processes considering that the available excess heat and temperatures. It further shows the median obtainable COP and the distribution shown in the 1st and 3rd quartile (25 and 75 percentile)

The potential for utilising excess heat with heat pumps in the Danish industry was done based on the work published in (Bühler et al., 2016b). First the amount of excess heat was estimated based on process energy use (Sørensen et al., 2015) for each process in the 22 industrial sector with the highest energy use. Excess heat and process heat were split in up into three temperature levels for each process. Based on these numbers the potential for upgrading the excess heat with a heat pump was evaluated. A simplified heat pump model consisting of the Lorenz COP and a heat pump efficiency of 0.55 was used. It was assumed that excess heat can only be used for the same process type. As the temperatures required generalisation, it was assumed that the excess heat is always cooled down to a reference temperature of 15 °C. Two cases where assumed for the heat sink: (i) it was assumed that the sink is heated from the excess heat temperature to the maximum heat pump supply temperature or the process heat temperature if below the maximum supply temperature, (ii) it was assumed that all heat is supplied at the lower of the maximum heat pumps supply temperature or required process heat temperature. The results are shown for the cases in Figure 3 and Figure 4.

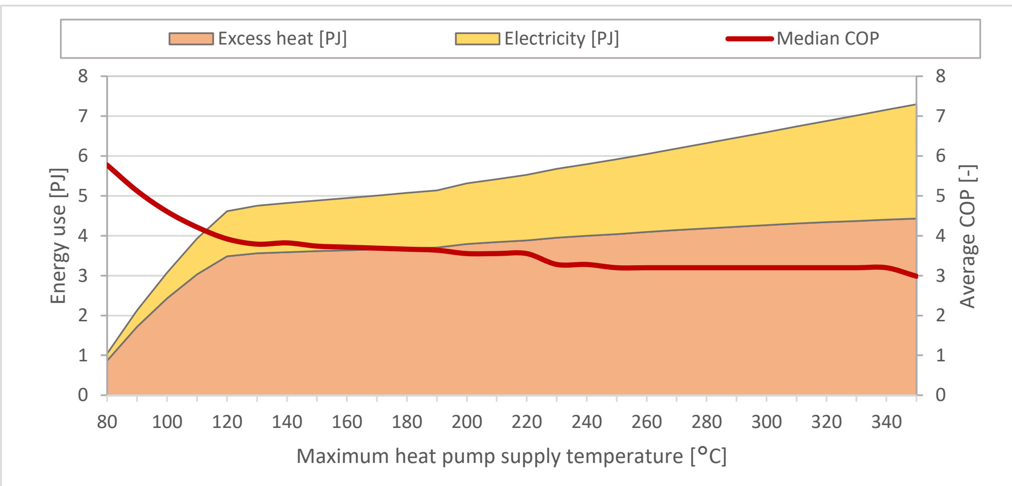

Figure 3: Utilisation potential of excess heat with heat pumps and average COP in the Danish manufacturing industry assuming the heat pump sink is heated from the excess heat temperature to the maximum supply temperature.

It can be seen that for a glide on the heat sink, there is a very sharp increase in the amount of process heat that could be supplied by heat pumps that are recovering excess heat. The lower increase above 120 °C is caused by the requirement of processing heating below this temperature and the decreasing availability of excess heat at temperatures above. From 120 °C onwards, the used excess heat increases slightly while the COP decreases, increasing the overall amount of process heat supplied over proportionally.

The same increase until 120 °C can be observed in Figure 4, where the heat is provided at the highest temperature without glide. However the amount of utilised excess heat is almost constant thereafter, as the obtainable COP are very low and thereby increase the heat supply covering the process heat demands.

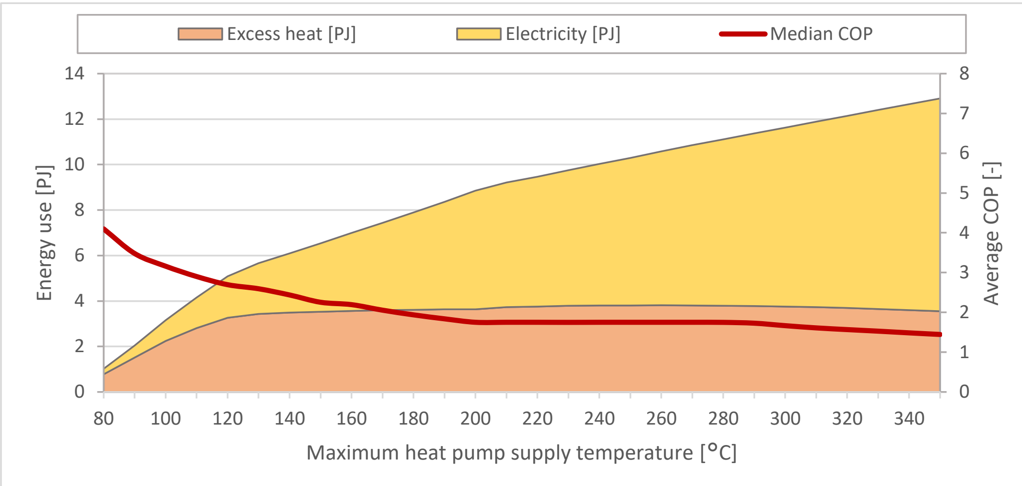

Figure 4: Utilisation potential of excess heat with heat pumps and average COP in the Danish manufacturing industry assuming all heat is supplied at the maximum heat pump supply temperature.

The differences in Figure 3 and 4 are mainly related to the different assumptions with respect to the heat supply. While it was assumed in Figure 3, that the heat is supplied to a medium that is heated from the excess heat temperature to the maximum process heat temperature, it was assumed in Figure 4 that the entire heat is supplied at the maximum process heat temperature. The assumption from Figure 3 corresponds to e.g., heating a single phase medium such as drying air, while the assumption from Figure 4 may be correct in case of process heat supply by steam.

Excess heat sources for heat pumps

The total excess heat found was further spatially distributed to individual production sites following the approach described in (Bühler et al., 2017). Finally, production profiles for industrial sectors were created to determine the peak excess heat. Profiles were created to describe main industry activities and to represent the size of industries (e.g. number of shifts). This approach was based on (Bühler et al., 2018; Wiese and Baldini, 2018). Initially this data and methods were used to find the potential of utilising excess heat for district heating, but are used in the following to give an impression of excess heat rates in the industrial sector.

Figure 5 and Figure 7 show the distribution of excess heat sources across heat rate intervals by temperature of the excess heat and by main industry sector. Similarly, Figure 6 and Figure 8 show the total excess heat potential by temperature and main industrial sector in these intervals. While the majority of the sources are below 100 kW, the highest excess heat potential is found in sources above 1 MW. Temperatures are relatively even distributed, however the small sources are mainly from food, chemical and wood processing industries. The large sources are exclusively found in oil refineries and non-metal mineral processing. It is however possible that in industries, excess heat from small sources is bundled and emitted from a single source.

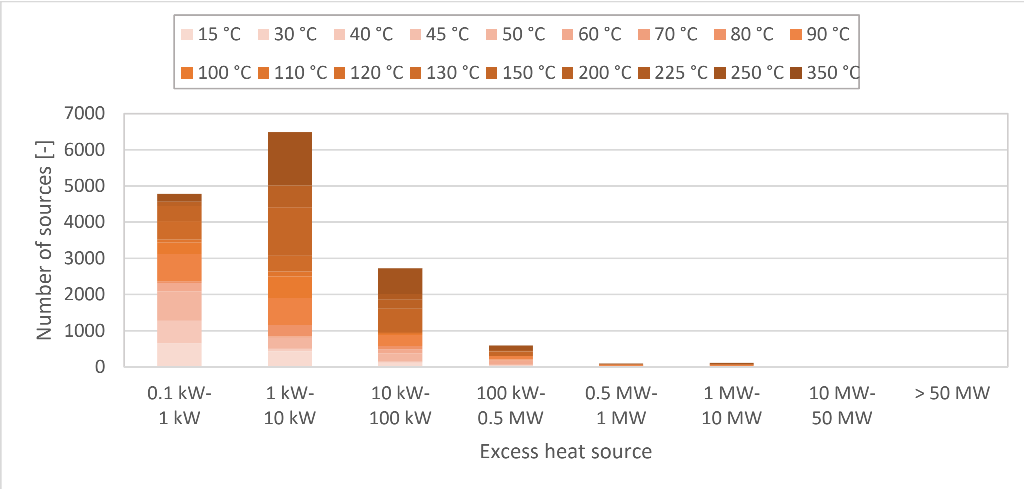

Figure 5: Distribution of excess heat sources from thermal processes in the Danish manufacturing industry (number of sources).

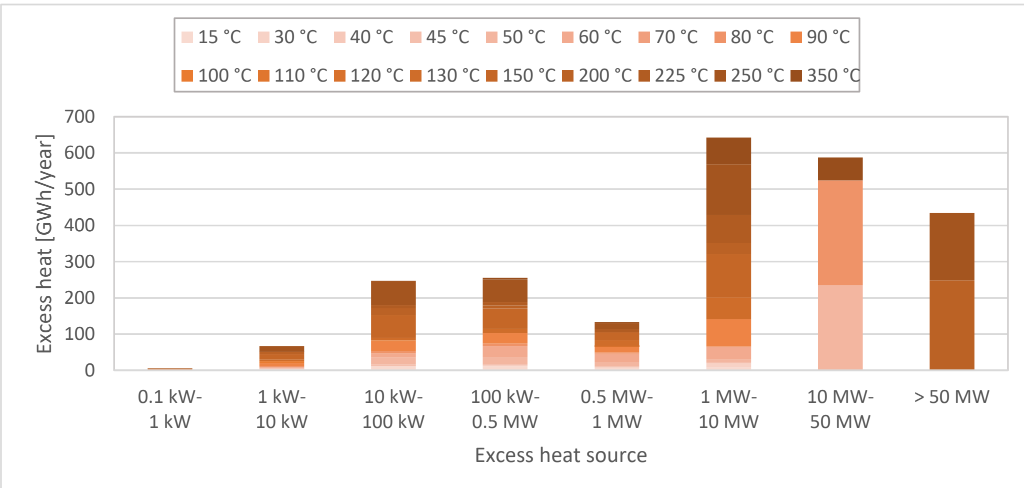

Figure 6: Distribution of excess heat sources from thermal processes in the Danish manufacturing industry (excess heat potential).

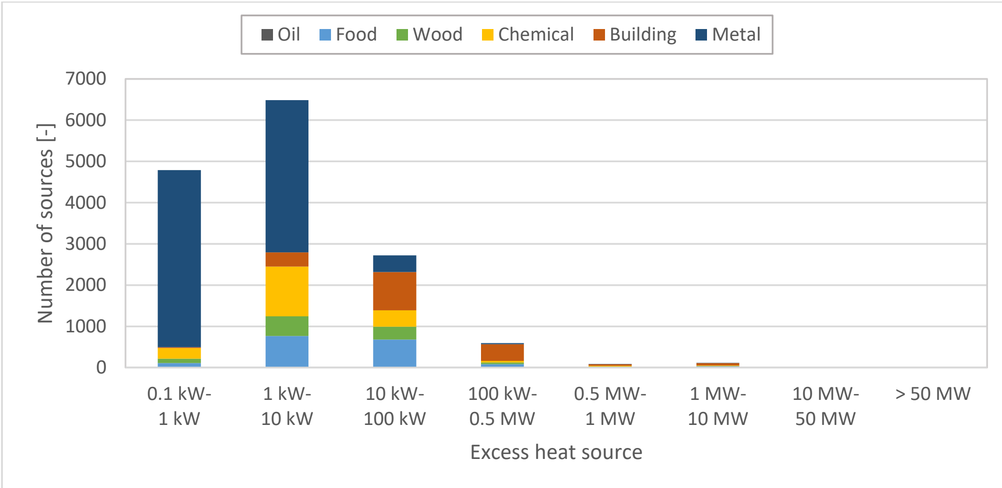

Figure 7: Distribution of excess heat sources from thermal processes in the Danish manufacturing industry (number of sources).

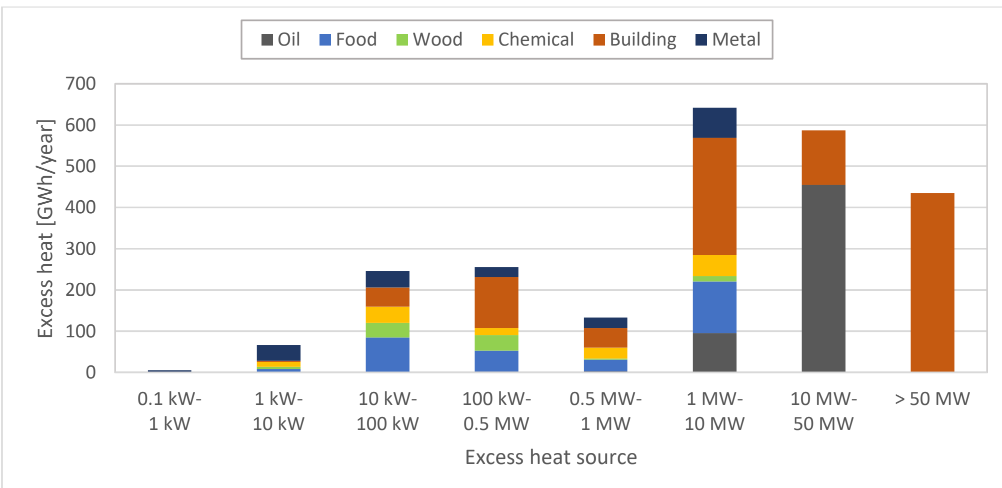

Figure 8: Distribution of excess heat sources from thermal process in the Danish manufacturing industry.

Conclusion

This work showed that heat pumps in the Danish industry, which can provide process heat at up to 120 °C can utilise a significant amount of excess heat to supply process heat. Technologies supplying higher temperatures have a possible potential if lower COPs are accepted. The final potential depends however on the matching of heat source and sink, as well as the heat sink characteristics. For high temperature heat pumps, due to economy of scale, particularly large excess heat sources will be of interest. The number of excess heat sources from thermal processes in the industry above 1 MW is limited to a few (below 150), however they represent a significant amount of the total available heat in Denmark.

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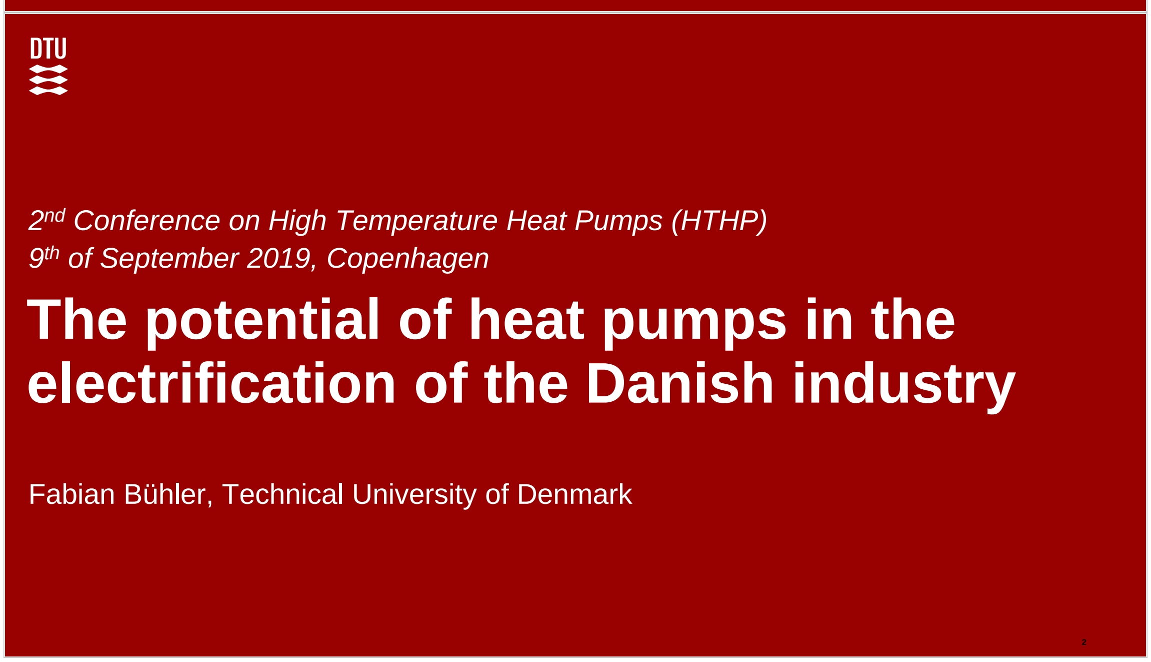
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9th of September 2019, Copenhagen
The potential of heat pumps in the electrification of the Danish industry
Fabian Bühler, Technical University of Denmark
2

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Agenda

Introduction to ELIDI Project

Case study: Electrification of a dairy factory

Industrial energy use and electrification potential in Denmark

Heat pumps in the electrification

Final remarks

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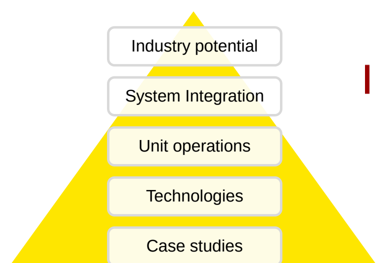
ELIDI - Project
Electrification of processes in the Danish Industry

Objective
- Reduce energy use in thermal processes
- Electrification options and technologies
- Potential for electrification in industry
- Challenges and boundary conditions for electrification



DTU Mechanical Engineering
Department of Mechanical Engineering

viegand maagøe
energy people

DANISH TECHNOLOGICAL INSTITUTE

DFD

SAN Electro Heat

ELFORSK

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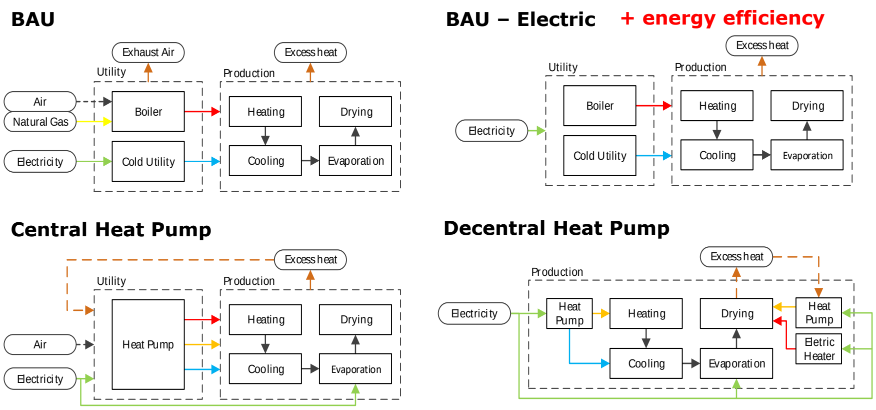
Electrification of dairy factory (milk powder)

Possible strategies



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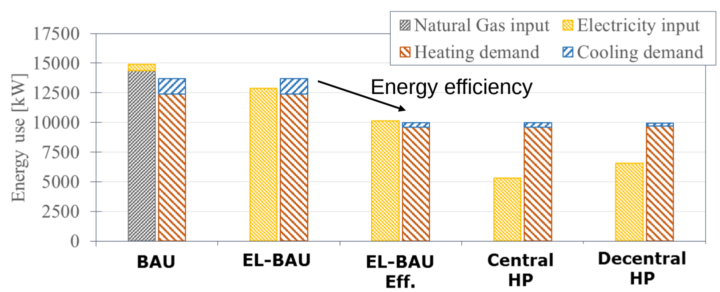
Electrification – Dairy factory

Energy analysis



- Central heat pump: System COP of 1.95
- Decentral heat pump: System COP of 1.57
  Individual COP between 1 and 5.2
- Cooling demand covered with HP

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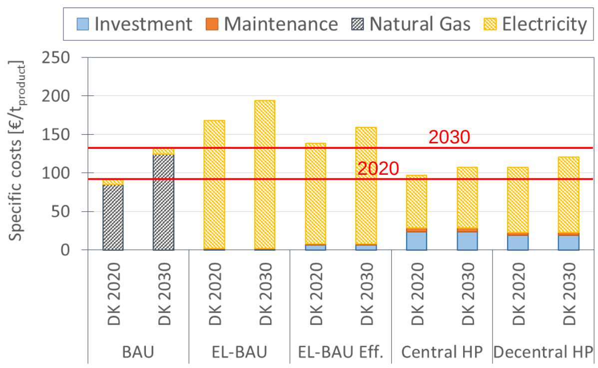
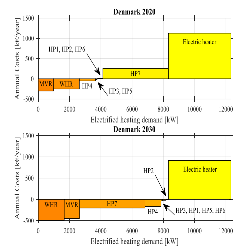
Electrification – Dairy factory
Economic analysis



Simple payback time

- DK 2030 (rplacing existing boiler):
  - Central 8 years
  - Decentral 10 years



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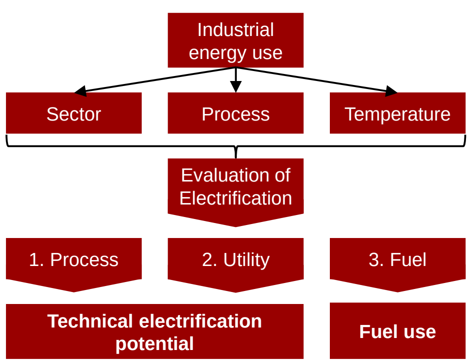
Electrification potential in Denmark
Approach



Process and utility

- Heat Pumps for process heat
  - Heat demand < 80 °C
    - Heat sources at 10 °C
  - Heat demand > 80 °C
    - Heat source gradually increasing up to 80 °C
- MVR, VRC HIDiC
- Electric heaters, boilers

Special processes

- Brick production (Microwave assisted kiln)
- Cement production (Clinker heating)
- Drying of metal paints (IR drying)
- Melting (Induction furnaces)

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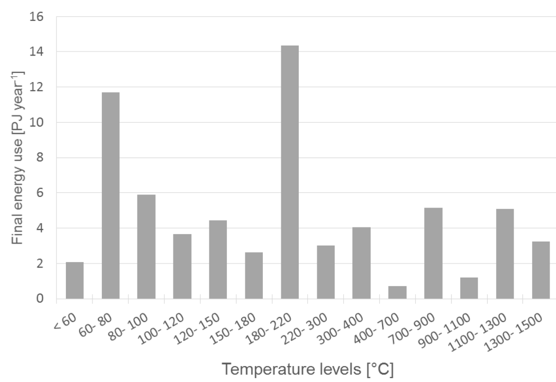
Industrial energy use in Denmark



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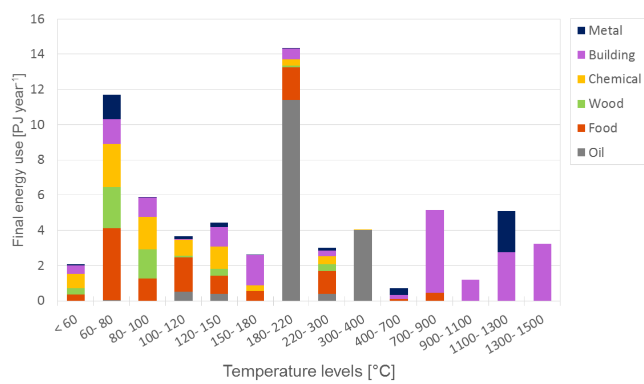
Industrial energy use in Denmark



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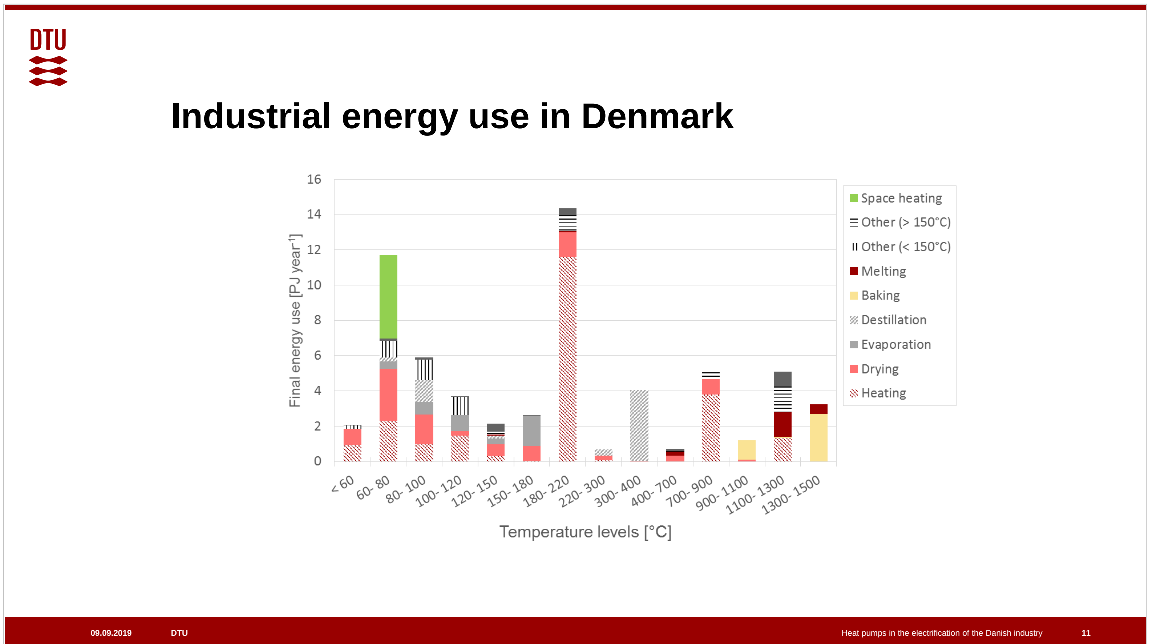
DTU
Industrial energy use in Denmark
Final energy use [PJ year⁻¹]
16
14
12
10
8
6
4
2
0
< 60
60- 80
80- 100
100- 120
120- 150
150- 180
180- 220
220- 300
300- 400
400- 700
700- 900
900- 1100
1100- 1300
1300- 1500
Temperature levels [°C]
Space heating
Other (> 150°C)
Other (< 150°C)
Melting
Baking
Destillation
Evaporation
Drying
Heating
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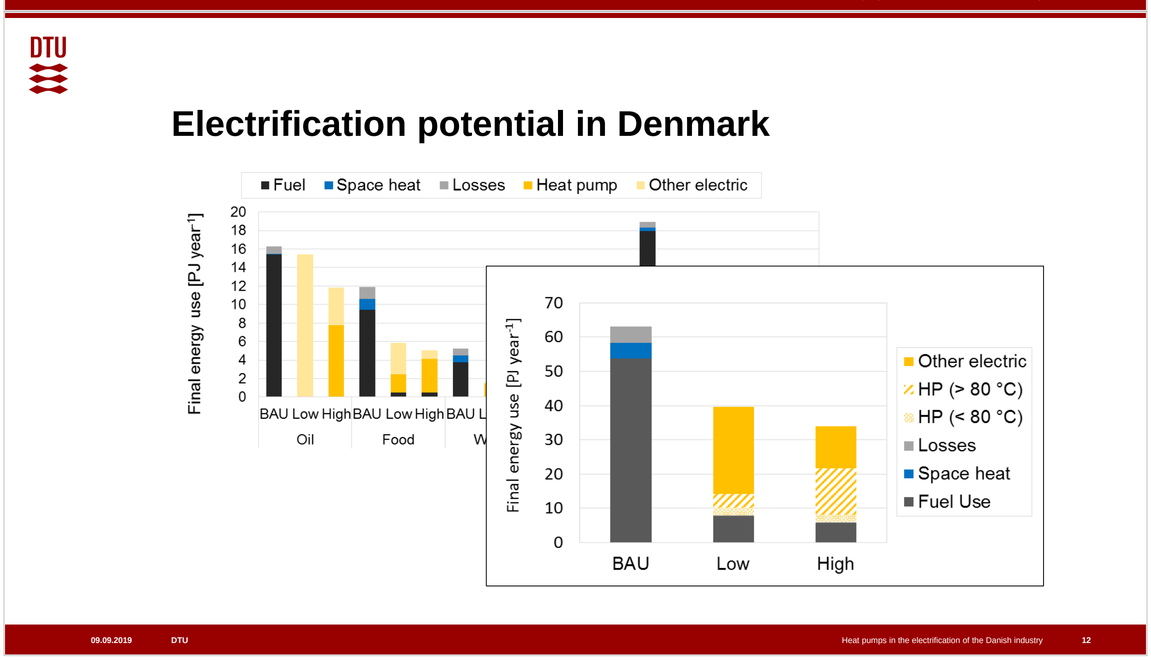
DTU
Heat pumps in the electrification of the Danish industry
11


DTU
Electrification potential in Denmark
Fuel
Space heat
Losses
Heat pump
Other electric
Final energy use [PJ year⁻¹]
20
18
16
14
12
10
8
6
4
2
0
BAU
Low
High
Oil
Food
Final energy use [PJ year⁻¹]
70
60
50
40
30
20
10
0
BAU
Low
High
Other electric
HP (> 80 °C)
HP (< 80 °C)
Losses
Space heat
Fuel Use
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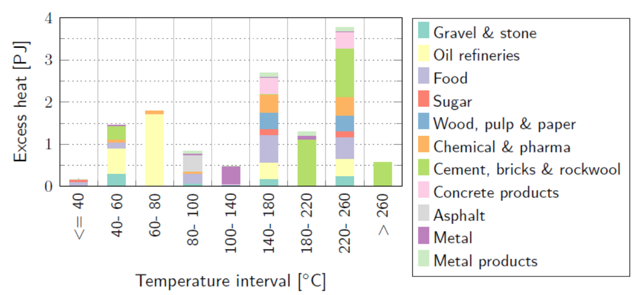
Heat pumps in the electrification of the Danish industry
12

DTU

Excess heat in Denmark



Excess heat in Denmark	212.0	PJ per year
↳ manufacturing industry	23.0	PJ per year
↳ thermal processes	12.6	PJ per year

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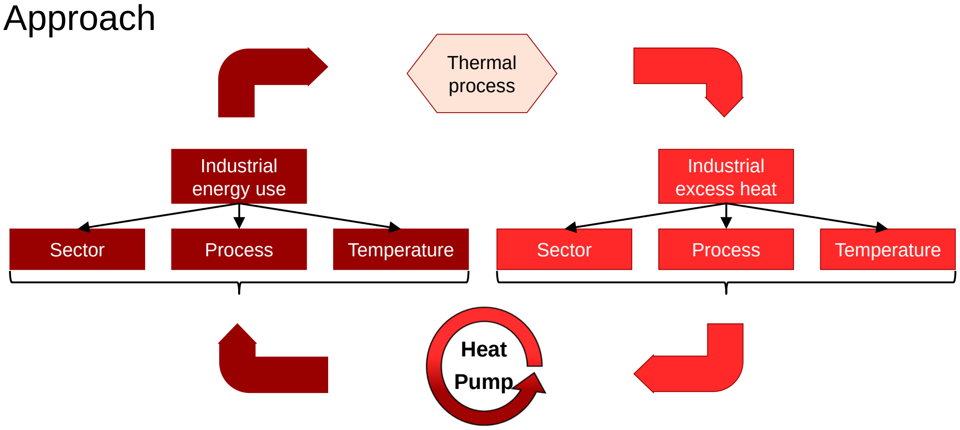
Electrification potential in Denmark

Approach



- Max. supply temperature
- Temperature lift

- Lorenz Cycle

- Outlet temperature

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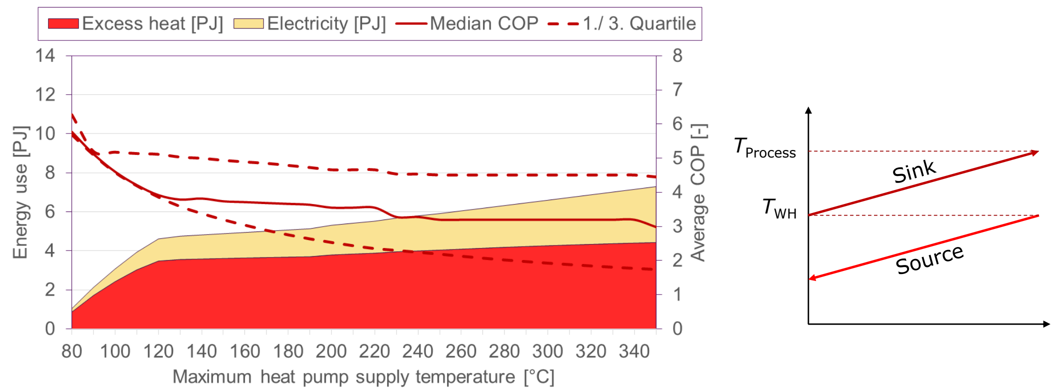
Heat pump potential in Denmark



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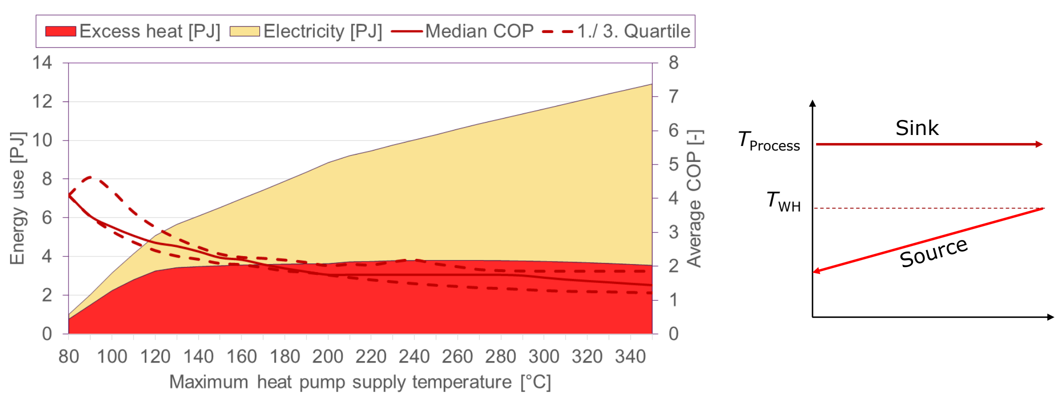
Heat pump potential in Denmark



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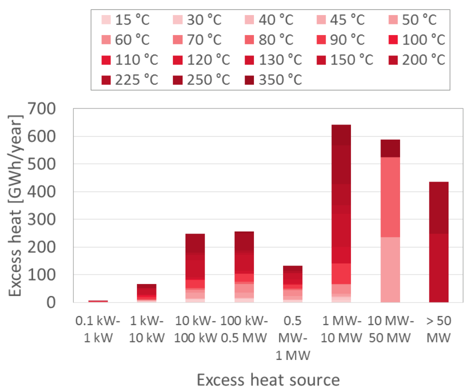
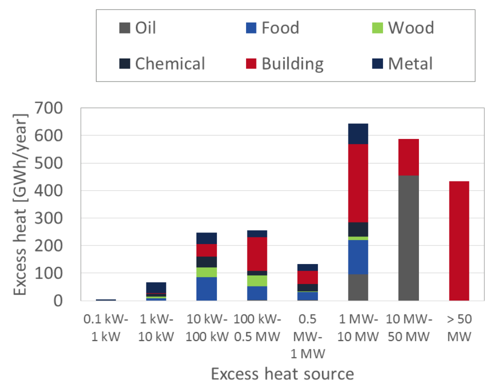
Excess heat sources in Denmark





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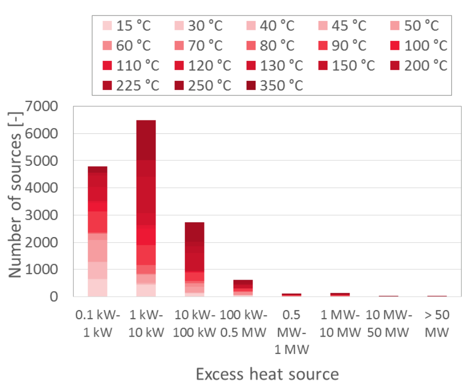
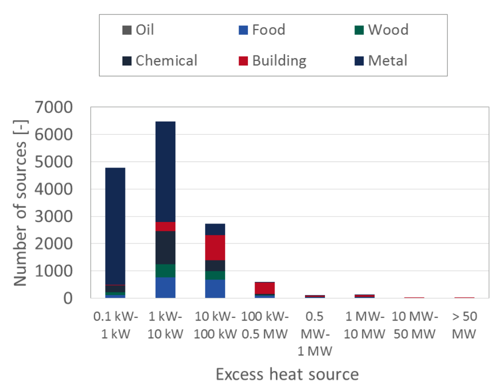
Excess heat sources in Denmark





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Concluding remarks

- Electrification through heat pump integration
  - Reduction in final energy use and energy related operating costs
  - ELIDI project will demonstrate feasibility for other case studies
- Denmark with high amount of low temperature heating demand
  - High electrification potential with existing technologies
  - Reduction in final energy use
- Excess heat utilization for process heating in Denmark with heat pumps
  - Majority of excess heat useable for process heat up to 150 °C
  - High number of very small excess heat sources
  - Recovery potential in food and non-metal minerals between 1 and 10 MW

DTU

Thank you for your attention!

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High-temperature heat pumps in pumped heat energy storage systems

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Keywords:
High temperature heat pump, R718, compressor technology, Carnot battery, Power-to-heat-to-power

Introduction

In northern Europe, the primary sources of renewable energy are wind power and photovoltaics. As a consequence future energy systems will mainly be based on electrical energy from renewable sources. The drawback of wind power and PV is the missing dispatchability, so storages for electrical energy will become crucial for a high share of renewable energy sources. Today the majority of storage capacity for electrical energy is provided by pumped hydro energy storages (PHES). Although PHES constitute an established and efficient storage technology, the potential is limited due to geological restrictions. Therfore, pumped thermal energy storage (PTES) systems could be a promising alternative. Here electrical energy is converted to thermal energy by means of a heat pump. The thermal energy is stored in a high-temperature thermal energy storage, which is, upon demand, converted back to electrical energy by a thermal power cycle. In an ideal, reversible implementation of this concept, the roundtrip efficiency would be 100 %. In contrast, if the high-temperature storage is charged by resistance heating, the roundtrip efficiency is limited to the thermal efficiency of the power cycle. A PTES based on a subcritical Rankine cycle is also referred to as Compressed Heat Energy Storage (CHEST), described in [1]. An overview on PTES technologies is given in [2].

Methods

In in a real implementation of a CHEST-cycle exergy losses lead to a reduction of the roundtrip efficiency. The most significant exergy losses are caused by

- the heat transfer to and from the high-temperature thermal energy storage
- the heat transfer from the heat source to the heat pump
- the heat transfer to the heat sink
- and the isentropic efficiency of the heat pump compressor and the power cycle expander

Therefore, while designing a PTES, it is crucial to avoid exergy losses. This can be achieved by minimizing the temperature differences during heat transfer, using a high-temperature storage system that matches, as well as possible, between the thermal profiles of condensation and subcooling at the heat pump side as well as preheating and evaporation on the power cycle side. The impact of the necessary temperature differences on the efficiency can be reduced by choosing a high temperature lift between the thermodynamic mean temperatures of evaporation in the heat pump or rather the heat sink of the power cycle and the high-temperature storage. This seems to be counterproductive on the first view, as increasing the temperature lift will reduce the COP of the heat pump. But at the same time the efficiency of the power cycle increases, while the specific amount of thermal energy that has to be stored in the high-temperature storage decreases. Consequently, the specific system size in relation to the electrical power can be reduced.

A CHEST variant with water as the working fluid is shown in Figure 1. For charging, thermal energy from a cold sensible water storage is used to evaporate water at 70 °C. The steam is compressed up to 100 to 107 bar in a six-stage compressor and subsequently condensed in a latent heat thermal energy storage (LH-TES). Sodium nitrate with a melting temperature of 306 °C is used as the storage material. As water is a wet working fluid, the condensate is flashed and fed back between every compressor stage to desuperheat the steam. An expansion valve reduces the condensate pressure to the evaporator pressure. Alternative configurations use an ammonia heat pump instead of the cold sensible water storage and a cascade of sensible storages to store the thermal energy from subcooling for preheating. For discharging, water is evaporated at 81 to 87 bar in the LH-TES and expanded in a 3 to 4-stage wet-steam cycle. A fraction of the steam is extracted from the turbines to recharge the cold sensible water storage. The rest is either used to provide district heating or it is further expanded to generate additional electrical energy.

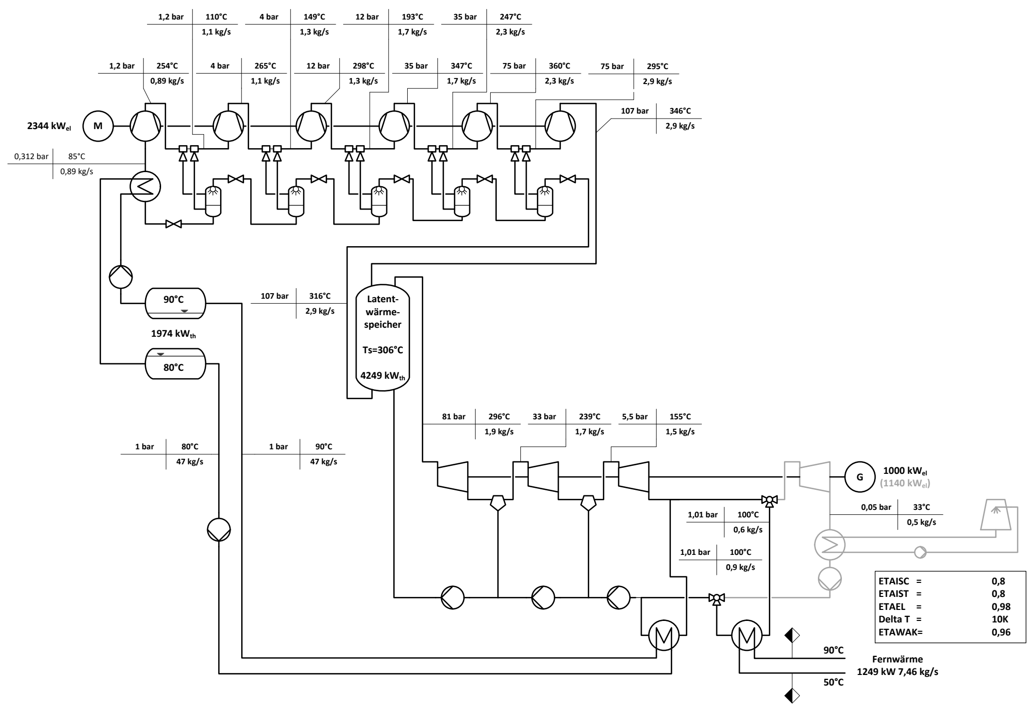

Figure 1: Flow diagram of the CHEST-System.

Results and Discussion

Several variants were simulated using the thermodynamic cycle calculation tool EBSILON PROFESSIONAL. At the cold side either the additional heat pump or a cold sensible water storage was considered. Also different configurations for the high- temperature storage and the related cycle set up are distinguished. The results are summarized in Table 1.

Table 1: Resulting roundtrip efficiencies for different implementations of the CHEST system.

Cold side	**Additional NH_3-heat pump**				**Cold sensible water storage**		
High temperature storage system (HT-TESS)	Latent+ sensible	Latent+ sensible	Latent	Latent	Latent+ sensible	Latent	Latent
Condensate injection	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Flash-evaporation + condensate injection	No	No	No	Yes	No	No	Yes
Sensible storage for intercooling	Yes	No	No	No	No	No	No
$Eta_{el.}$	0,64	0,64	0,49	0,56	0,61	0,48	0,57
$Eta_{el.,with\ district\ heating}$	-	0,44	0,34	0,39	0,56	0,41	0,51
Utilization factor	-	1,79	1,38	1,57	0,95	0,96	0,96

The highest roundtrip efficiency of 64 % is obtained with the ammonia heat pump as the heat source and a thermal energy storage cascade consisting of the LH-TES, a molten salt two tank storage covering the temperature range between 300 °C and 177 °C and a pressurized water-storage between 177 ° and 80 °C. The reason therefore is the optimal matching of the TES to the half cycles. A simplified process variant using only the LH-TES and flash-steam/condensate injection obtains a roundtrip efficiency of 56 %. Using the cold sensible water-storage, the efficiency is reduced to 61 % for the cascaded high-temperature storage variant. In combination with the flash-steam/condensate injection, the simple variants efficiency is slightly higher with 57 %. All other variants reach only efficiencies below 50 %. Operating the system in combined heat and power mode, the electrical roundtrip efficiency is reduced by 15 to 20 percent points for the variant with the ammonia heat pump and by 5 to 6 percent points for the variant with the cold sensible water storage. But the utilization factor is very high for the ammonia heat pump variant as additional thermal energy from the environment is supplied to the cycle.

Conclusion and References

PTES systems can be a promising, site-independent and cycle stable alternative for bulk electrical energy storage. The presented CHEST-concept provides roundtrip efficiencies up to 64 %. In combined power and heat operation, PTES provide an interface to the thermal energy sector and can reach very high total energy utilization factors, acting as an energy hub. However further research and development has to be conducted to design high efficiency water-steam compressors and sophisticated LH-TES concepts.

[1] Steinmann W.-D. The CHEST (Compressed Heat Energy STorage) concept for facility scale thermos- mechanical energy storage. Energy 2014; 69:543–52. doi:10.1016/j.energy.2014.03.049.

[2] Steinmann W.-D. Thermo-mechanical concepts for bulk energy storage. Renew Sustain Energy Rev 2017; 75:205–19. doi:10.1016/j.rser.2016.10.065.

High-temperature heat pumps in pumped heat energy storage systems

Working principles and system requirements

2nd International Conference on High-Temperature Heat Pumps, Copenhagen

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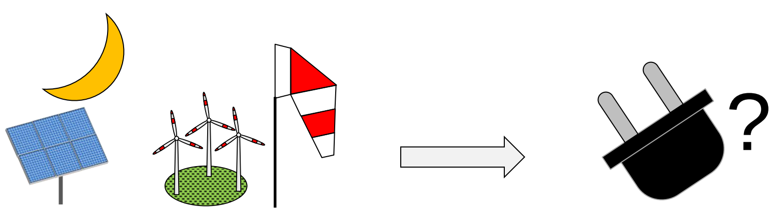
Motivation: Why power-to-heat-to-power?



- Electrical energy generated from wind and PV not dispatchable
- High share of renewable energy sources require storages
- Potential for pumped-hydro energy storages geologically limited
- Battery storages offer a limited number of cycles

→ Power-to-Heat-to-Power storage systems are a promising alternative

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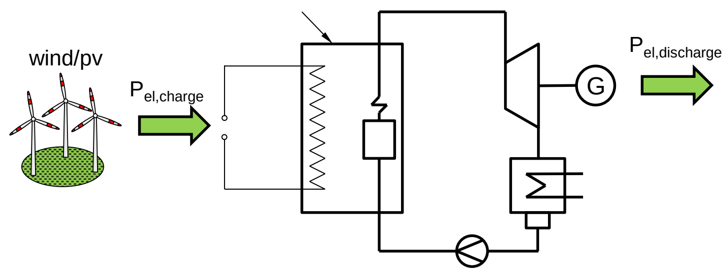
Power-to-Heat-to-Power (PHP) based on resistance heating



- Ideal efficiency: Carnot-factor of heat engine
- Real efficiency: ~35-40 %

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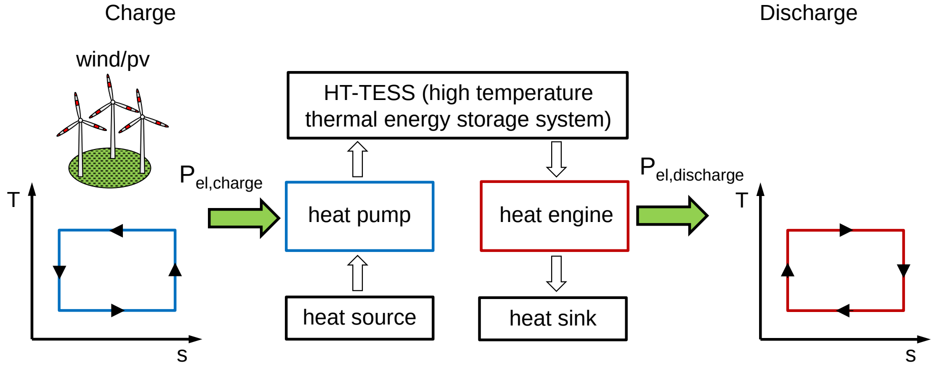
PHP based on Pumped Thermal Energy Storage (PTES)



→Ideal cycle: roundtrip efficiency = 100 %
→They provide an interface for thermal energy integration

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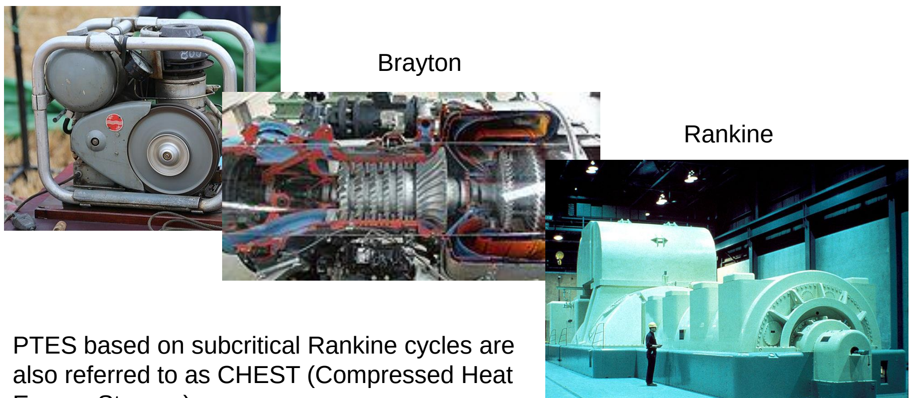
Options for basic thermal cycle

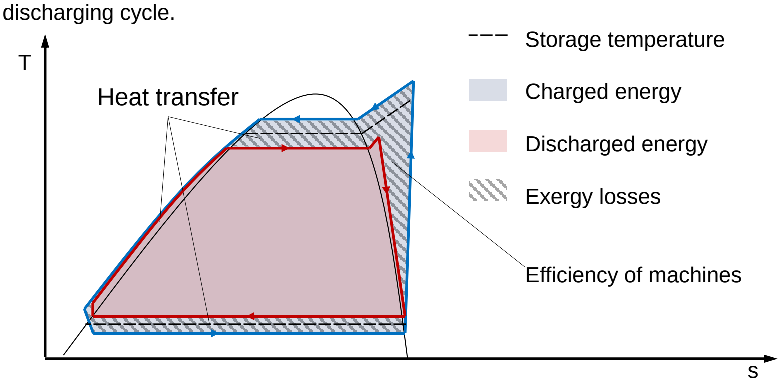
Closed thermal cycles with external heat supply



PTES based on subcritical Rankine cycles are also referred to as CHEST (Compressed Heat Energy Storage)

Exergy losses in real CHEST-Systems

- Exergy losses are caused by
  - heat transfer and
  - limited efficiency of machines.
- Ratio between sensible and latent heat thermal energy is different for charging and discharging cycle.

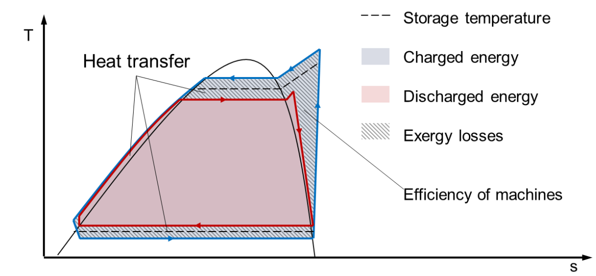


$$\text{Roundtrip efficiency } \eta_{RT} = \frac{P_{el,discharging}}{P_{el,charging}} = COP \cdot \eta_{th}$$

Design requirements

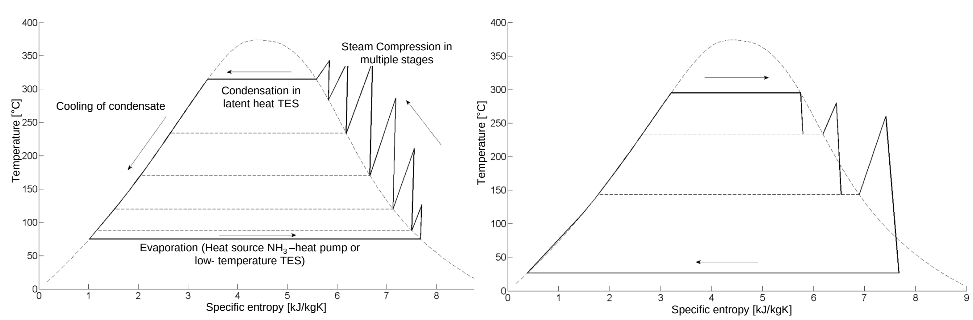
- Charging and discharging cycles must be as congruent as possible.
  - High-temperature thermal energy storage system (HT-TESS) has to be matched carefully between the thermal profiles of heat pump and power cycle.
  - High temperature lift
    - to minimize impact of exergy losses in heat exchangers
    - to keep COP of the heat pump low → reduction of thermal storage capacity in HT-TESS required per kW/h_{el}
  - Use of high efficiency machines
  - Keep thermal energy that can't be used for reconversion into electrical energy in the system

$$\eta_{RT} = COP \cdot \eta$$



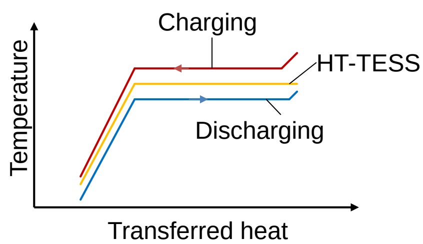
The CHEST (Compressed Heat Energy STorage) concept

Charging | Discharging

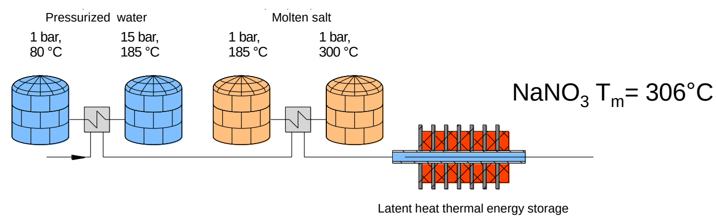


Steinmann, Energy 69 (2014) 543-552

First design challenge: Matching of HT-TESS between heat pump and power cycle

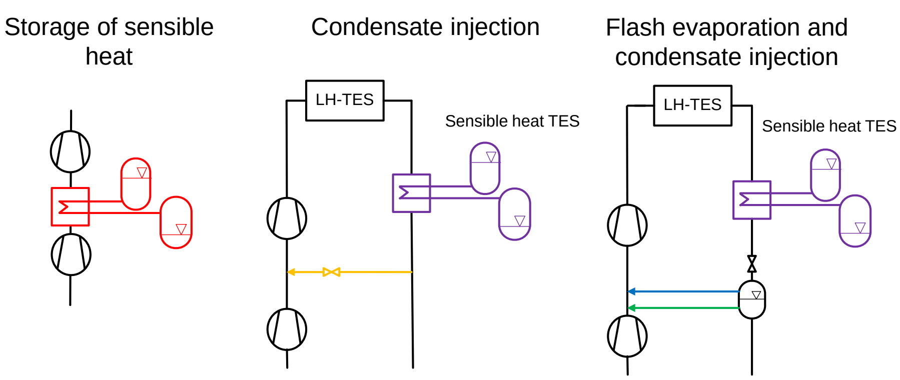


- HT-TESS as cascade of sensible and latent heat TES.
- Alternatively only latent heat TES.



Second design challenge: Intercooling/ desuperheating

- Water steam compression requires cooling between compressor stages.



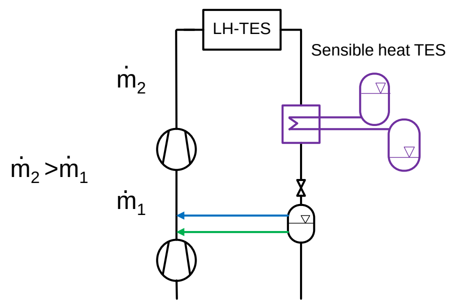
Third design challenge: Ratio of sensible to latent heat

- Ratio of sensible to latent heat thermal energy varyies between Charging and Discharging.
- Typically the heat pump provides too much sensible thermal energy.
- Excess sensible thermal energy can‘t be used in power cycle.

  - Flash-evaporation
  - Condensate injection

  Reduction of compression work in the low pressure stages .

- It‘s possible to use only the LH-TES



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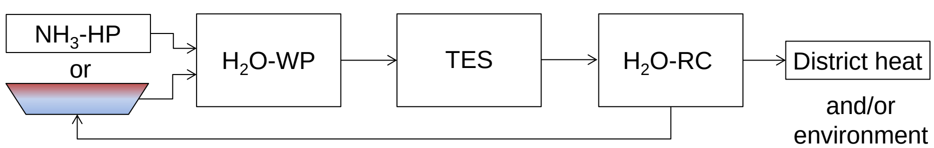
Options for the cold side



Options cold side charging

- Use of environment or waste heat as a heat source
  - Additional NH3- heat pump required
  - Only feasible if sea or lake water is available.
  - Ambient air is not suitable due to parasitic power consumption of fans and icing during winter.
- Pit-water-storage

Options cold side discharging

- District heating
- Pit-water-storage

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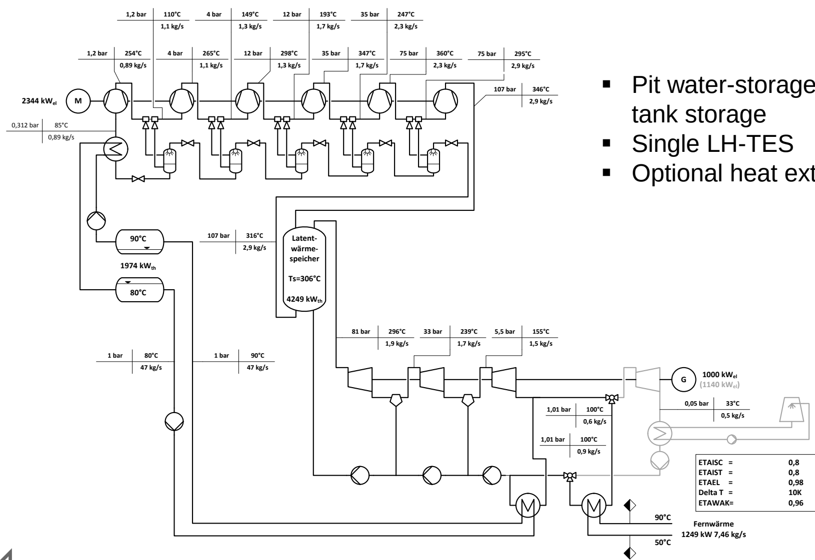
Flow diagram of a water- based CHEST-system



- Pit water-storage/Two-tank storage
- Single LH-TES
- Optional heat extraction

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Achievable efficiencies of system variants

Cold side	Additional NH_3-heat pump				Pit-water-storage		
High temperature storage system (HT-TESS)	Latent+ sensible	Latent+ sensible	Latent	Latent	Latent+ sensible	Latent	Latent
Condensate injection	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Flash-evaporation + condensate injection	No	No	No	Yes	No	No	Yes
Sensible storage for intercooling	Yes	No	No	No	No	No	No
$Eta_{el.}$	0,64	0,64	0,49	0,56	0,61	0,48	0,57
$Eta_{el.,with\ district\ heating}$	-	0,44	0,34	0,39	0,56	0,41	0,51
Utilization factor	-	1,79	1,38	1,57	0,95	0,96	0,96

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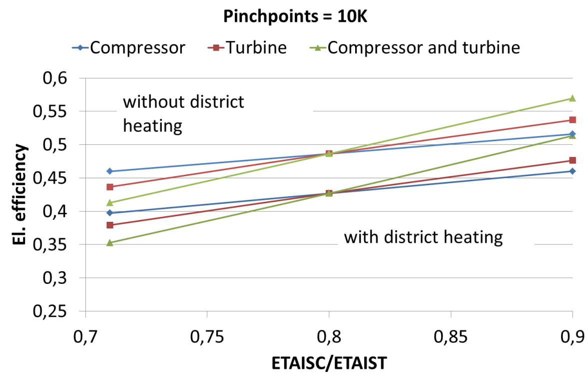
Impact of isentropic machinery efficiency



- Efficiency of turbine is dominant.
- Reduction of isentropic efficiency from 0.9 to 0.8 reduces Eta_{el} by 9 percent points on average.

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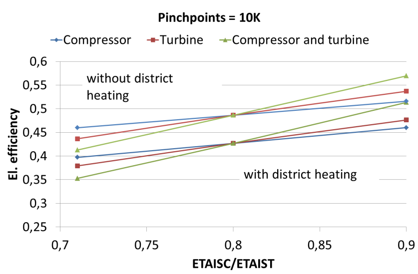
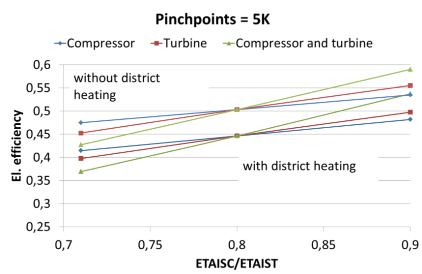
Impact of temperature difference in pinch-points





Reduction of temperature difference in pinch-points of heat exchangers from 10 K to 5 K increases Eta_{el} on average by 2 percent points.

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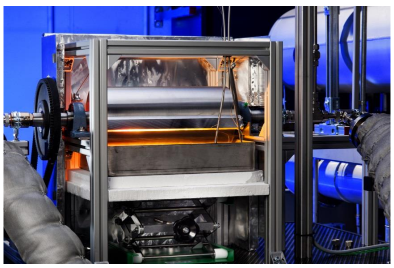
Identification of R&D fields

- Current LH-TES have a limited feasibility for high power/ high capacity applications
- Development of active LH-TES with separation of heat exchanger and storage tanks

- No off-the-shelf water-steam compressors with intercooling available.
- For a prototype size (0.5 to 1 MW) available compressors could be modified.



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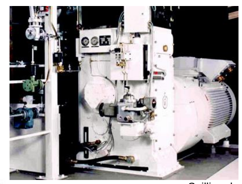

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Conclusion and outlook

Conclusion

- PTES- storage systems could be an alternative to pumped hydro and battery storages.
- They are independent from geological restrictions and provide cycle stability.
- Roundtrip efficiencies around 60% are achievable for water based PTES.
- By sector coupling to the thermal energy sector, PHP-systems can provide a high total energy utilization factor.

Outlook

- Key components such as water steam compressors and active LH-TESS require further R&D.

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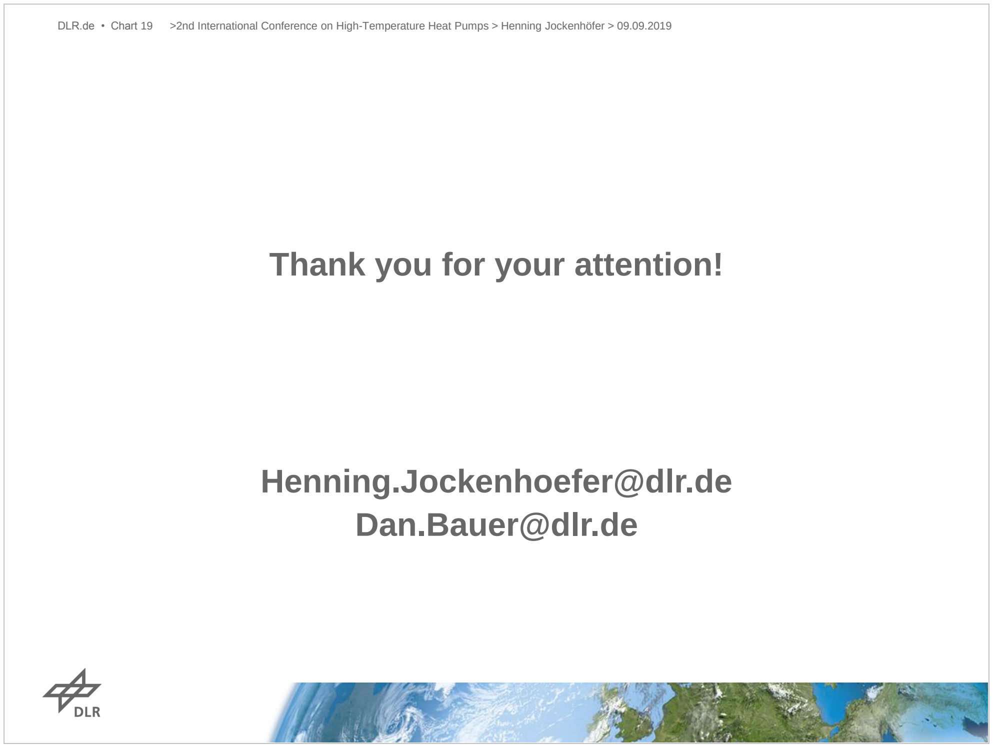
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Industrial heat pumps in the Netherlands – developments and demonstrations

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Keywords:
Climate agreement, Industrial heat pump R&D

Abstract

The 2015 Paris Climate Agreement initiated the discussions on the Dutch national level to define greenhouse gas reduction targets. To combat climate change, the Dutch government aims to reduce the Netherlands' greenhouse gas emissions by 49% by 2030, compared to 1990 levels [1]. The government is now taking measures and has made agreements with other parties to achieve this ambitious goal. The National Climate Agreement, finalised June 2019 contains agreements with the sectors on what they will do to help achieve the climate goals. The participating sectors are: electricity, industry, built environment, traffic and transport, and agriculture.

For the industrial sector the target is to reduce greenhouse gas emissions by 55%, forcing the sector to implement more energy efficient production processes and to incorporate low carbon or renewable fuels and feedstocks in their processes. Policy measures to support the stakeholders in the energy transition are being drafted, and expected to come into force in 2020.

In the search for more energy efficient and cleaner processes, heat pump technologies offer an interesting option; the primary energy consumption of heating processes can be reduced and a switch from fossil fuels to renewable electricity can be made.

Two large scale heat pumps are currently in development in the Netherlands. In the district heating network of the city of Utrecht, a 25 MW_{th} heat pump is being designed to deliver heat at 75°C, covering 10% of the heat demand [2]. A wastewater treatment facility provides the heat source, as well as the needed space to install the heat pump and 5000m^3 thermal water storage volume.

At the DOW Terneuzen site a steam recompression system is built as a pilot project [3]. This heat pump is designed to deliver 12.5 barg steam at 266°C at a capacity of 12 ton/hour (~8MW). The inlet steam is at 3 barg, 170°C, and is upgraded by a 2-stage centrifugal compressor. The pilot plant is commissioned in Q2-2019 and operation will be monitored through 2020.

The number of applications of industrial heat pumps in NL is limited and are predominantly found in moderate temperature food processing applications. As the industry is pushed to reduce their CO_2 emissions, the interest for heat pump solutions is increasing. To fulfil todays requirements of efficient heating in industrial processes, heat pump technologies need to be able to deliver heat at temperatures

above 100°C, operate reliably and efficient and become available at an acceptable initial investment cost. That is, the technology needs to become competitive against the current gas fired heating systems. The R&D program ‘industrial heat’ of ECN.TNO focuses its research and development of heat pump technologies to deliver heat in the temperature range of 120-180°C, and scaling up of the heat pump technologies towards higher technology readiness level. Several projects on closed cycle compression heat pumps are ongoing in collaboration with industrial end-users and heat pump component and system suppliers. The 200 kW_{th} butane pilot heat pump [4] at a paper process proved the ability to deliver steam at 120°C from a waste heat source at 60°C. The learnings from this project formed the basis for the development roadmap of industrial heat pumps. Some of the ongoing projects will be further highlighted in the presentation, covering the developments of a 2 MW_{th} heat pump with synthetic and natural refrigerants, and the development of a high temperature heat pump unit for testing of innovative cycles, components and working fluids.

These heat pump developments are further strengthened by new testing infrastructure brought together in the Carnot lab. This lab consists of test-rigs to operate and characterize the performance of novel heat pump concepts and components from 1 kW_{th} up to 2 MW_{th}.

References / Sources

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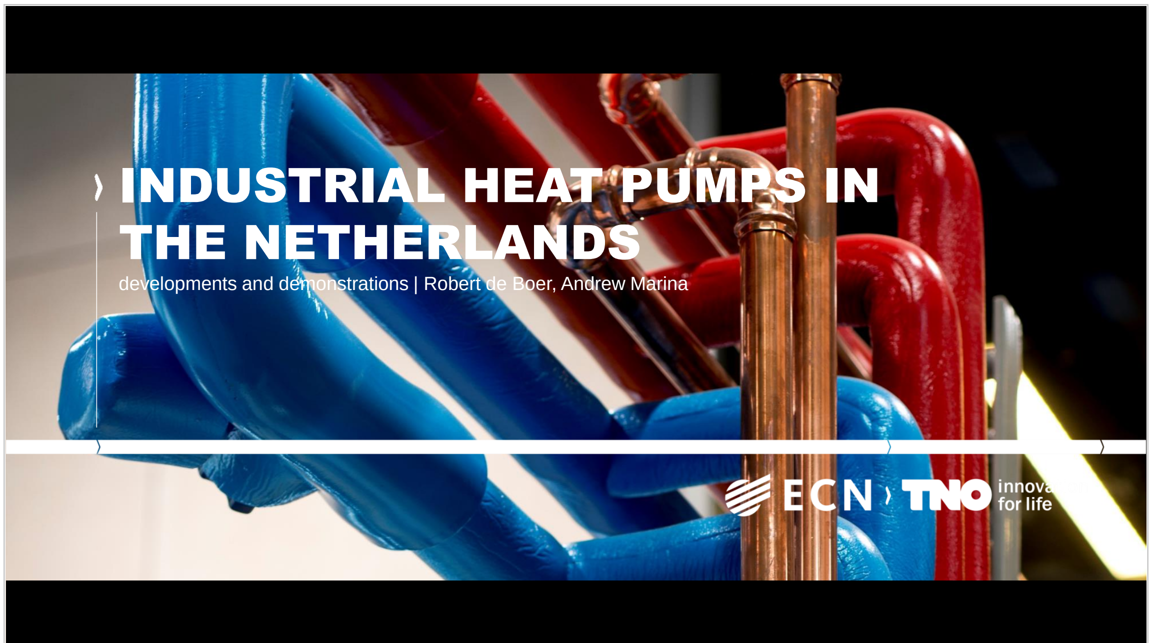


CONTENT

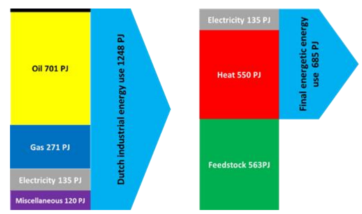
- Background
- Heat Pump applications
- Industrial heat pump developments

2 | Industrial heat pumps in the Netherlands

ECN › TNO innovation for life

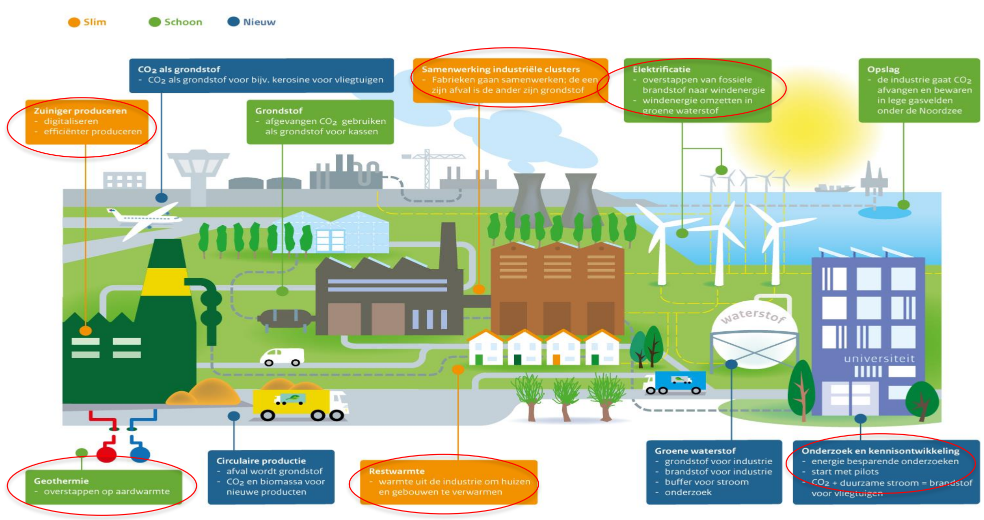
BACKGROUND

- 2015 Paris Climate agreement → 2019 Dutch National Climate Agreement (Klimaat akkoord)
- **2030**: Overall 49% CO_2 emission reduction (1990 reference)
  - Electricity – 70% renewable (SolarPV, Wind-onshore/offshore,)
  - Buildings – 1.5 Million houses made sustainable / net zero energy
  - Industry – 55% CO_2 emission reduction



- Starting 2020: New subsidy schemes in place to stimulate implementation of low CO_2 technologies
  - Industrial heat pumps

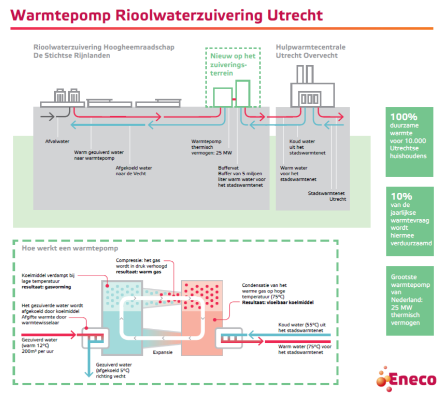
3 | Industrial heat pumps in the Netherlands



INDUSTRIE Klimaat-akkoord

HEAT PUMP IN DISTRICT HEAT NETWORK

- 25 MW_{th} Heat pump in district heating of city of Utrecht
- ENECO district heat operator
- Wastewater effluent as heat source
- Largest DH-heat pump in NL
- Thermal storage included
- Heat delivery at 75°C
- Operational in 2022

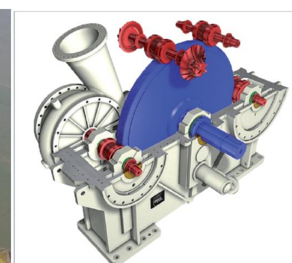


Source: Eneco

5 | Industrial heat pumps in the Netherlands

HEAT PUMP IN INDUSTRIAL STEAM SYSTEM

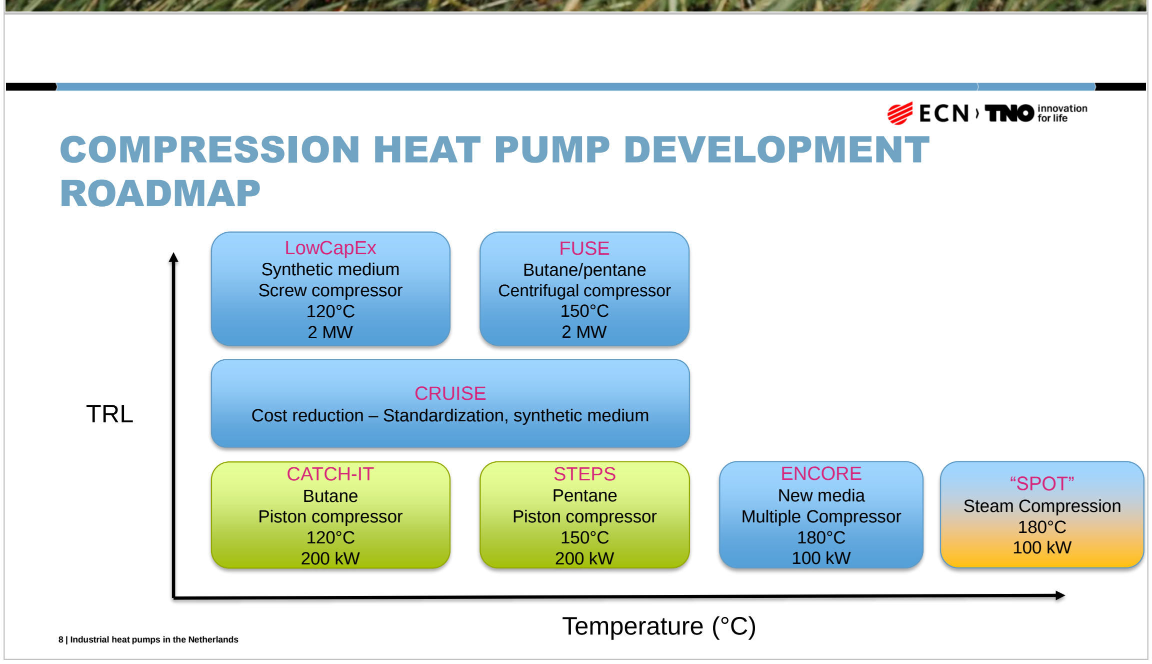
- Demonstration project @DOW Terneuzen
  - Capacity: 12 tonne/hr
  - Inlet conditions 170°C, 3 bar(g)
  - Outlet conditions 266°C, 12.5 bar(g)
  - Electrical power 1350 kW
  - COP 5.89
  - Compressor radial, 2 stages, AtlasCopco
  - Timeline Feasibility study 2016, Detail engineering 2017-2018; Compressor order Q4-2017, Delivery Q4-2018; Commisioning Q2-2019, Monitoring 2019-2020



Source: Klop, Eschweiler, BWK bd 70(2019)9

6 | Industrial heat pumps in the Netherlands


ECN › TNO innovation for life
› HEAT PUMP DEVELOPMENTS
7 | Industrial heat pumps in the Netherlands


ECN › TNO innovation for life
COMPRESSION HEAT PUMP DEVELOPMENT ROADMAP
TRL
LowCapEx
Synthetic medium
Screw compressor
120°C
2 MW
FUSE
Butane/pentane
Centrifugal compressor
150°C
2 MW
CRUISE
Cost reduction – Standardization, synthetic medium
CATCH-IT
Butane
Piston compressor
120°C
200 kW
STEPS
Pentane
Piston compressor
150°C
200 kW
ENCORE
New media
Multiple Compressor
180°C
100 kW
"SPOT"
Steam Compression
180°C
100 kW
Temperature (°C)
8 | Industrial heat pumps in the Netherlands

ECN › TNO innovation for life

LOW CAPEX: 2 MW_{TH} HEAT PUMP @120°C

- Technical
  - Synthetic working fluid – R-1233zd(E)
  - Parrallel construction of the high pressure side
  - Potential for further lowering capex
  - Step towards industrial demonstrations
- After laboratory tests, heat pump will be integrated in an industrial process as demonstrator unit.

9 | Industrial heat pumps in the Netherlands

ECN › TNO innovation for life

FUSE PROJECT (2019-2021)

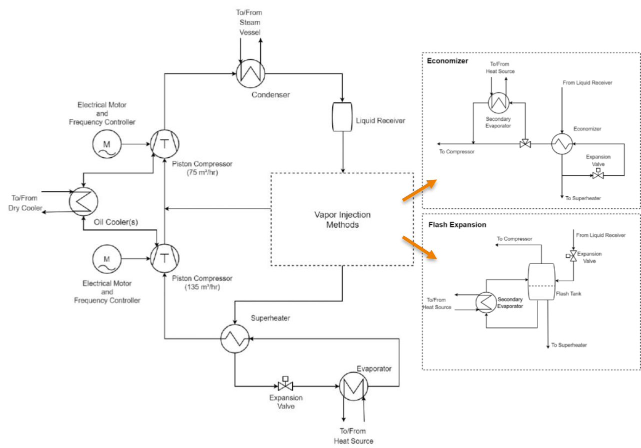
- Project outline
  - 1-2 MW steam producing, industrial heat pump with natural working fluid
  - Waste heat 60°C-90°C to produce medium pressure steam (range of 2-5 bar) 150°C
  - Specific investment cost of < 200 €/kWth (excluding integration)
  - Develop a modular compression heat pump design utilizing a limited number of standardized components that can be configured in numerous ways to cover > 70% of the industrial heat pump market. This enables series production and therewith low costs (< 200 €/kW_{th}) and offers still flexibility with respect to operating conditions (power, temperature)
  - Establish heat pump performance under a variety of simulated (industrial process data) static and dynamical conditions at ECN part of TNO, and under field test conditions at the DOW Terneuzen site.
  - Establish a Dutch manufacturer for the standardized, modular, flexible compression heat pump.

10 | Industrial heat pumps in the Netherlands

ECN › TNO innovation for life

ENCORE: HIGH TEMPERATURE HEAT PUMP UNIT

- Testing innovative working fluids, cycles, components, compressors, etc.
  - System is modifiable to achieve this
- Gain sufficient practical experience to implement innovations in full scale design/demo's
- 100 $kW_{th(out)}$ direct expansion system
- Possible working fluids:
  - Pentane
  - Hexane
  - R1336mzz(Z)



11 | Industrial heat pumps in the Netherlands

ECN › TNO innovation for life

CARNOT LAB

- Building of Carnot lab facilities currently ongoing
  - Planned completion Q1-2020
- R&D infrastructure for conversion and storage of heat
  - Support for technology development of the R&D program
  - Testing of materials, componenents, new concepts and systems
  - Development and test facility for industry
  - Data coupling with existing (industrial) processes for dynamic tests and simulations.
- Measurement equipment
- Heat/cold infrastructure (1 kW ….. 2 MW)
- Development platform for compression heat pump
- Modeling software

2MW full scale test rig

12 | Industrial heat pumps in the Netherlands

WRAP-UP

- Climate agreement expected to raise demand for industrial heat pump technology in NL
- Few industrial heat pump implementations currently in development
- Heat pump R&D in NL targets at
  - Increasing delivery temperatures to match industrial process needs
  - Reducing the CAPEX of industrial heat pumps to 200€/kW_{th}
  - Test facilities for IHP performance characterization available / under construction

14 | Industrial heat pumps in the Netherlands

2 Industrial cases and examples of successful integration of heat pumps

1.1 High-temperature heat pumps in Japan - Potential, development trends and case studies, Takenobu Kaida, CRIEPI

1.2 High-temperature heat pumps in Austria: Demonstration and application examples, Veronika Wilk, AIT

1.3 Combined heating and cooling: Integrated ammonia-water heat pump in modern dairy production, Stein Rune Nordtvedt, Hybrid Energy

1.4 Hydrocarbon heat pumps with combined process cooling and heating at 115 °C, Christian Schlemminger, SINTEF

1.5 Two-phase vane compressor for supply of industrial process steam, Nikolai Slettebø, Tocircle

High Temperature Heat Pumps in Japan
- Potential, Development Trends, and Case Studies -

Takenobu Kaida

Central Institute of Electric Power Industry (CRIEPI),
Energy Innovation Center (ENIC), Yokosuka, Japan,
kaida@criepi.denken.or.jp

Keywords:
High temperature heat pump, Potential, Development trend, Case study, Distillation process

Abstract

For significant reduction of greenhouse gas emissions, it is important to decrease the emission factor of electricity, to electrify heat usage, and to use electricity with high efficiency. In particular, industrial sector has much potential for electrifying heat usage and for reducing greenhouse gas emissions. Heat pump is recognized as one of the key technologies for the industrial electrification.

When focusing on steam demand and hot water demand for heat pump application, it is reported that the heat demand between 50°C and 150°C is estimated at 300 PJ/year in Japan [1]. The Japanese Government expects the cumulative shipments of industrial heat pumps achieve the heating capacity of 1,673 MW by FY2030 compared to the actual cumulative shipments of 11 MW in FY2013 [2]. The effect is estimated at 1.35 million ton-CO_2 reduction.

In recent years, various types of industrial heat pumps have been developed and commercialized in Japan [3]. When focusing on high temperature heat pumps (HTHPs) over 100°C, the following 5 products are available in the market; a hot air supply heat pump (Eco Sirocco) by MAYEKAWA, steam supply heat pumps (SGH120 and SGH165) by KOBELCO, a pressurized hot water supply heat pump (ETW-S) by MHI Thermal Systems, and a steam supply heat pump by Fuji Electric. KOBELCO has already prepared the product line-up up to 165°C. Fuji Electric, MAYEKAWA and MHI Thermal Systems prepared HTHPs around 120°C and are developing HTHPs over 150°C by NEDO projects [4].

Generally, coefficient of performance (COP) of heat pump becomes smaller as increasing supply temperature. It depends on the price and emission factor of electricity how much COP is necessary for customer's benefit. Today, in Japan, the emission factor of electricity is about 0.5 kg-CO_2/kWh [5] and the price ratio of electricity to city gas is about 2.8 [6]. These values make a little difficult for applying HTHPs.

The author has performed the experimental performance evaluation of the SGH165 [7]. The performance data were acquired under various conditions on the assumption of actual conditions. As well as extracting technical issues, the competitive condition was clarified compared to existing boiler

system. In addition, the author visited installed sites of SGHs and conducted interview survey to user companies and engineering companies about application of steam supply heat pumps. The effects by applying SGHs were as planned or better than planned, so the users satisfied them and had no additional requests to the heat pumps. The engineering companies recognized the good operability of the heat pumps. But for applying it furthermore, they request higher COP and lower initial cost.

As examples of good practices, two case studies for applying heat pumps to distillation processes are shown in this presentation. Existing distillation column for ethanol or methanol needs the temperature above 100°C. The first case applied 120°C steam supply heat pump for ethanol distillation [8]. The effects were 43% CO_2 reduction and 54% energy cost reduction. On the other hand, the second case applied 90°C hot water supply heat pump for methanol distillation [9]. This could be realized by decompression of distillation column to vacuum pressure. The effects were 60% CO_2 reduction and 63% energy cost reduction. In this way, decreasing heat demand temperature by changing process shows the better effectiveness of heat pump and gives the more opportunity of heat pump even when electricity situation is not so good economically and environmentally like at this time in Japan.

In conclusions, toward electrification and decarbonisation for future, both higher temperature heat pump for extending the territory of heat pump application and process innovation for shifting heat demand to lower temperature are important realistically for spreading industrial heat pumps.

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[8] JEHC, Collection of case studies for industrial electrification, Vol. 4, March, 2015. [in Japanese] http://www.jeh-center.org/asset/00032/monodukurinidenki/vol4_hokkaidobaioetanoru.pdf.pdf

[9] JEHC, Collection of case studies for industrial electrification, Vol. 6, March, 2019. [in Japanese] http://www.jeh-center.org/asset/00032/monodukurinidenki/vol6_meitousangyou.pdf

CRIEPI
Central Research Institute of Electric Power Industry

High Temperature Heat Pumps in Japan

- Potential, Development Trends, and Case Studies -

2nd Conference on High Temperature Heat Pumps

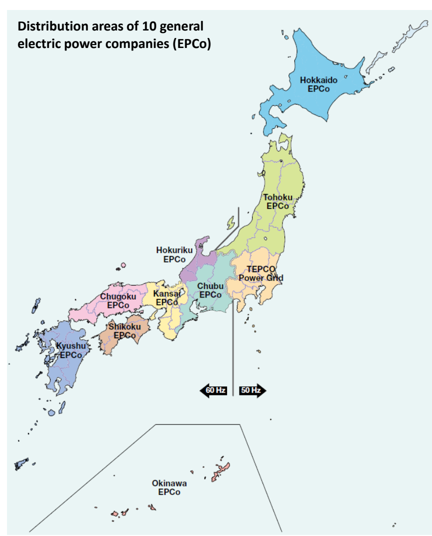
9th September, 2019 | Copenhagen, Denmark

Takenobu KAIDA

Central Research Institute of Electric Power Industry (CRIEPI)

What is CRIEPI?

- **Common research institute**
  of 10 general electric utilities in Japan
- **Academic research institute**
  based on objective scientific knowledge
- **In-house research institute**
  with own experimental facilities and research tools



Based on: JEPIC, The Electric Power Industry in Japan 2019.

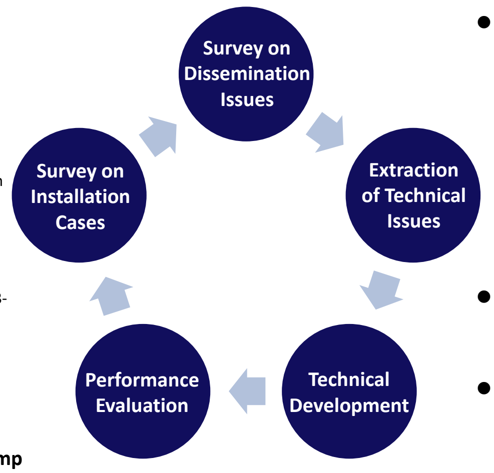
2

CRIEPI's Activities about Industrial Heat Pumps

- **Survey of installation cases and dissemination issues**
  (IEA HPT Annex 48, FY2016-2018)
  (Sharing information with JEHC & HPTCJ)

  ***JEHC** = Japan Electro-Heat Center
  ***HPTCJ** = Heat Pump & Thermal Storage Technology Center of Japan

- **Performance evaluation of heating tower defrosting**
  (Joint research with KUKEN, FY2013-2014)
- **Performance evaluation of steam supply heat pump**
  (Contract research from electric utilities, FY2014-2015)
- **Performance evaluation of DX-type industrial heat pump**
  (Contract research from electric utility, FY2016)



- **Extraction of technical issues**
  (FEPC, METI, enecho, NEDO)

  ***FEPC** = Federation of Electric Power Companies of Japan
  ***METI** = Ministry of Economy, Trade and Industry
  ***enecho** = Agency for Natural Resources and Energy
  ***NEDO** = New Energy and Industrial Technology Development Organization

- **Development of lumber drying system with heat pump**
  (Joint research with Shimane prefecture and Toshiba Carrier, FY2013-2015)
- **Application of low GWP refrigerant to high temperature heat pump**
  (Independent research with the cooperation of AGC and KOBELCO, FY2017-2018)

 3

Outline

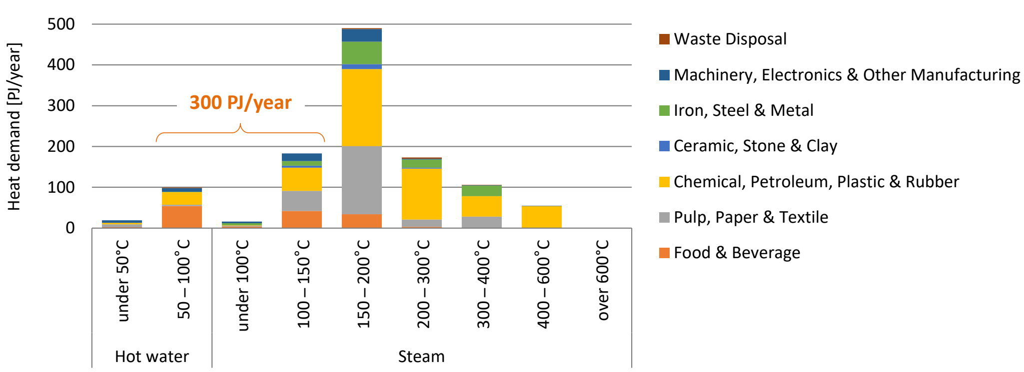
- **Potential & Development Trends**
  - Overview of Industrial heat pumps (IHPs)
  - Recent development trends of high temperature heat pumps (HTHPs)
- **Case Studies**
  - Performance evaluation of steam grow heat pump « SGH165 »
  - Interview survey about SGHs
  - Application for distillation process

 4

Potential & Development Trends

Potential of IHPs in Japan

- Focusing on **steam demand** and **hot water demand** for HP application
- Heat demand potential of IHPs: **300 PJ/year** (= 83.3 TWh/year)
- Large Potential industry: « Chemical », « Food », « Paper », « Machinery »

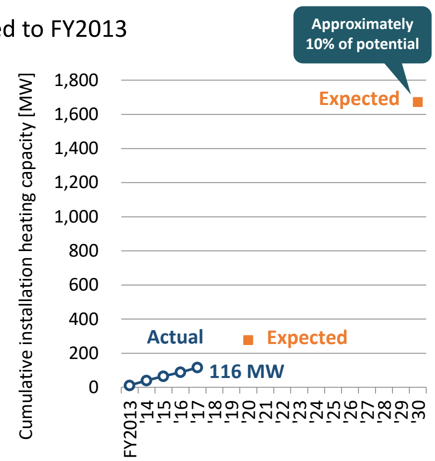


Based on: MRI, Survey report on heat demand and supply equipment, commissioned by enecho, 2018. [in Japanese]

6

Expectation of Spreading IHPs

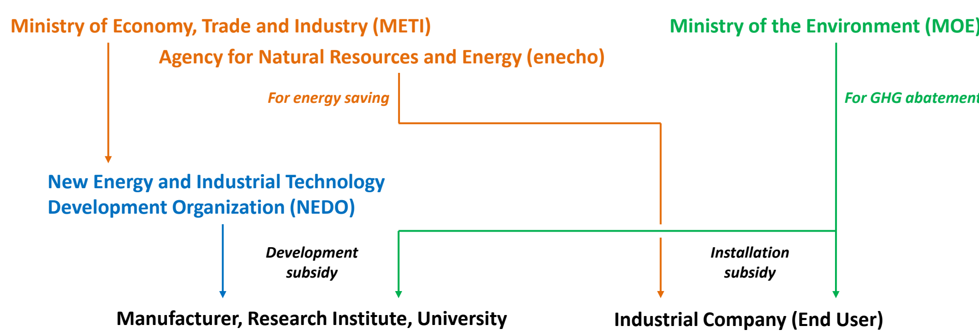
- "The Plan for Global Warming Countermeasures"
  - Decided by the Cabinet on 13th May, 2016
  - 26% reduction of GHG emissions by FY2030 compared to FY2013 (= 367 million ton-CO_2 reduction)
- Role of IHPs
  - **Over 150 times spread** (11 MW in FY2013 to 1,673 MW in FY2030)
  - 1.35 million ton-CO_2 reduction (= about 0.4% of total GHG emissions reduction)



 Based on: METI & MOE Joint Meeting, 1st March, 2019. [in Japanese] 7

FYI | Government Support in Japan

- Two types of subsidies
  - Development subsidy (Higher efficiency, Higher temperature, Lower GWP)
  - Installation subsidy



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Commercialized HTHPs over 100°C

	MAYEKAWA	KOBELCO	KOBELCO	MHI Thermal Systems	Fuji Electric
External Appearance					
Commercialized Year	2009	2011	2011	2011	2015
Product Name	Eco Sirocco	SGH120	SGH165	ETW-S	—
Heat Source/Sink	Water/Air	Water/Steam	Water/Steam	Water/Water	Water/Steam
Supply Temperature	60-**120**°C	100-**120**°C	135-**175**°C	**130**°C	100-**120**°C
Heat Source Temperature	0-40°C	25-65°C	35-70°C	55°C	60-80°C
Heating Capacity (Steam Rate)	110 kW[*1]	370 kW[*2] (0.51 ton/h)	624 kW[*3] (0.89 ton/h)	627 kW[*4]	30 kW[*5] (45 kg/h)
COP	3.7[*1]	3.5[*2]	2.5[*3]	3.0[*4]	3.5[*5]
Refrigerant	R744 (CO_2)	R245fa	R245fa+R134a	R134a	R245fa
Compressor	Reciprocating	Screw	Screw	Centrifugal	Reciprocating
Heat Pump Cycle	Transcritical	Subcritical	Subcritical + Steam Compression	Transcritical	Subcritical

*1 Heat source: 30-25°C, Heat sink: 20-100°C *2 Heat source: 65-60°C, Heat sink: 20-120°C *3 Heat source: 70-65°C, Heat sink: 20-165°C *4 Heat source: 55-50°C, Heat sink: 70-130°C *5 Heat source: 80-75°C, Heat sink: 20-120°C

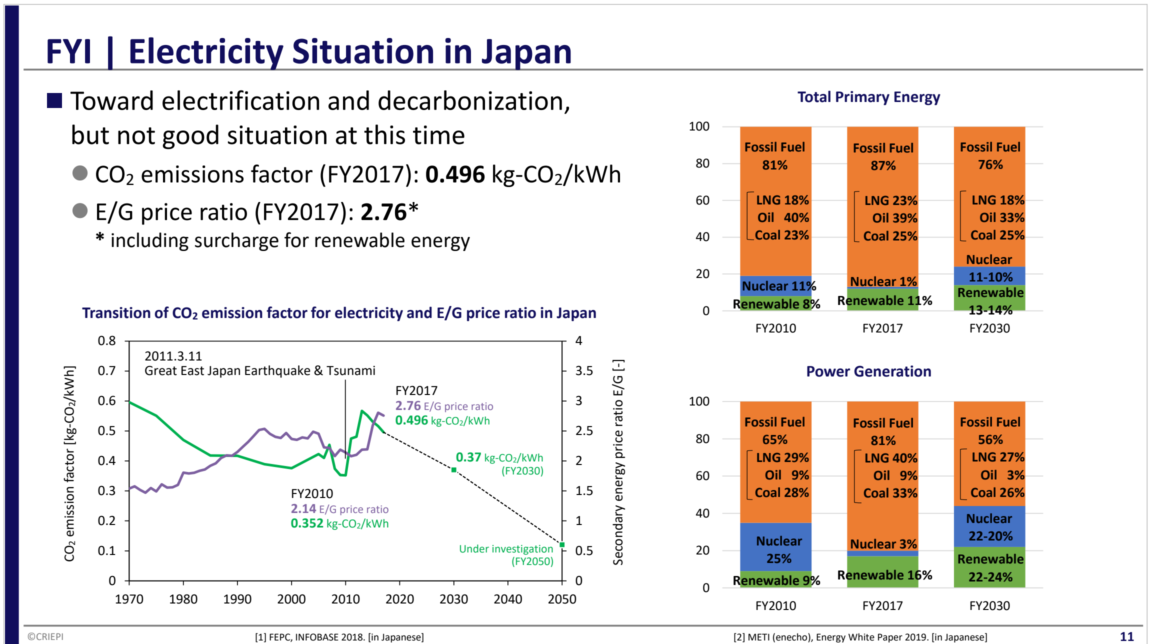
Based on: Manufacturers' brochure
9

Recent National R&D Projects for HTHPs over 150°C

- NEDO Projects
  - Program for Strategic Innovative Energy Saving Technology
    - Steam supply heat pump (**Fuji Electric**, FY2015-FY2018)
  - R&D Project for Innovative Thermal Management Materials and Technologies (TherMAT)
    - High temperature heat pump (**MAYEKAWA**, FY2015-FY2022)
    - High temperature heat pump (**MHI Thermal Systems**, FY2015-FY2022)

	Fuji Electric	MAYEKAWA		MHI Thermal Systems	
Supply Temperature	**150**°C	**160**°C	**200**°C	**160**°C	**200**°C
Heat Source Temperature	70-90°C	80°C	100°C	80°C	100°C
Heating Capacity	30 kW	300 kW	—	600 kW	—
Target COP	≥ 3.3	≥ 3.5	≥ 3.5	≥ 3.5	≥ 3.5
Refrigerant	R1336mzz(Z)	R600	—	R1336mzz(Z)	—
Compressor	Scroll	Centrifugal	Centrifugal	Centrifugal	Centrifugal
Heat Pump Cycle	Subcritical	Transcritical	—	Subcritical	—

Based on: Document by NEDO
10


FYI | Electricity Situation in Japan
Toward electrification and decarbonization, but not good situation at this time
CO2 emissions factor (FY2017): 0.496 kg-CO2/kWh
E/G price ratio (FY2017): 2.76*
* including surcharge for renewable energy
Transition of CO2 emission factor for electricity and E/G price ratio in Japan
CO2 emission factor [kg-CO2/kWh]
Secondary energy price ratio E/G [-]
2011.3.11
Great East Japan Earthquake & Tsunami
FY2017
2.76 E/G price ratio
0.496 kg-CO2/kWh
0.37 kg-CO2/kWh (FY2030)
FY2010
2.14 E/G price ratio
0.352 kg-CO2/kWh
Under investigation (FY2050)
Total Primary Energy
Fossil Fuel 81%
LNG 18% Oil 40% Coal 23%
Nuclear 11%
Renewable 8%
Fossil Fuel 87%
LNG 23% Oil 39% Coal 25%
Nuclear 1%
Renewable 11%
Fossil Fuel 76%
LNG 18% Oil 33% Coal 25%
Nuclear 11-10%
Renewable 13-14%
FY2010
FY2017
FY2030
Power Generation
Fossil Fuel 65%
LNG 29% Oil 9% Coal 28%
Nuclear 25%
Renewable 9%
Fossil Fuel 81%
LNG 40% Oil 9% Coal 33%
Nuclear 3%
Renewable 16%
Fossil Fuel 56%
LNG 27% Oil 3% Coal 26%
Nuclear 22-20%
Renewable 22-24%
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[1] FEPC, INFOBASE 2018. [in Japanese]
[2] METI (enecho), Energy White Paper 2019. [in Japanese]
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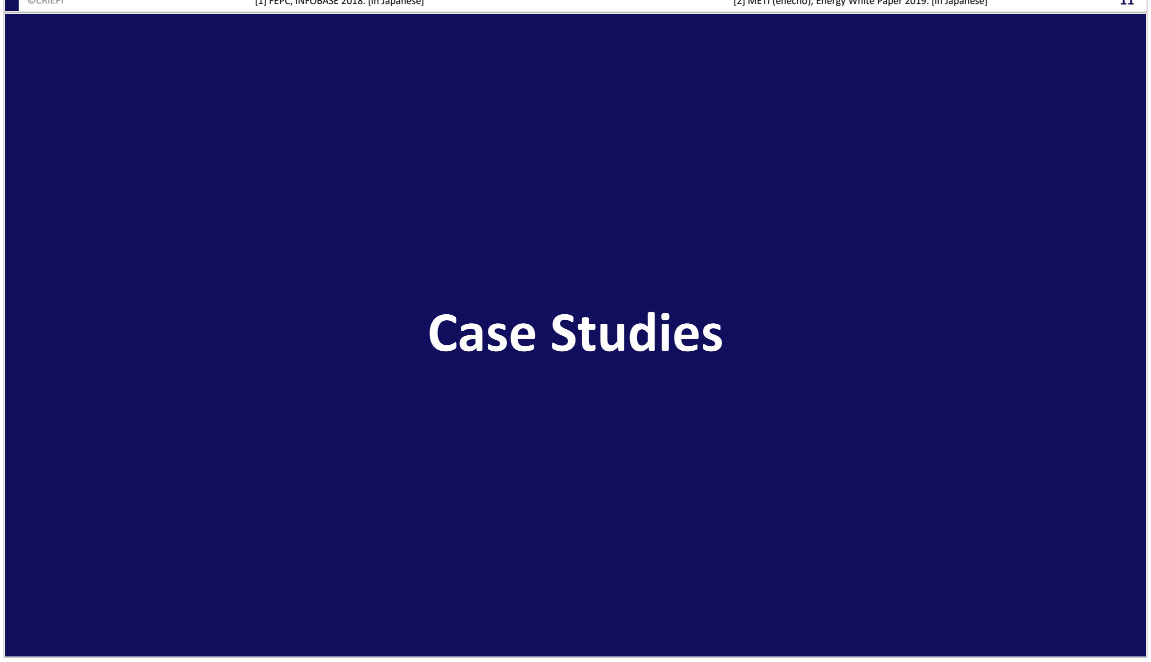
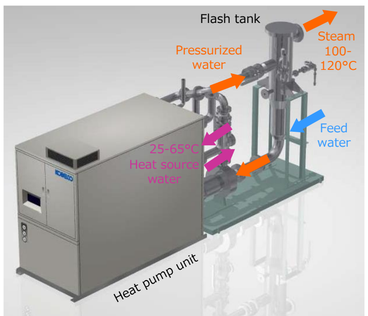

Case Studies

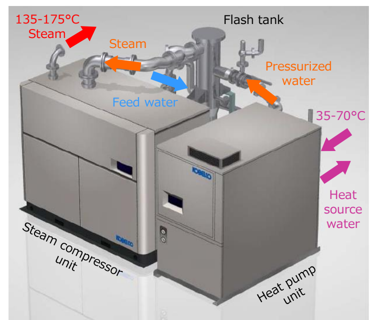
Steam Grow Heat Pumps (SGHs)

- Developed by **KOBELCO** and Japanese electric utilities
- Commercialized in 2011 (just before 3.11 disaster)



SGH120

R245fa
Two-stage screw compressor
COP = 3.5, *Q* = 370 kW (0.5 ton/h)
Steam = 120°C, Heat source = 65°C



SGH165

R245fa + R134a (mixture)
Single-stage screw compressor + Steam compressor
COP = 2.5, *Q* = 660 kW (0.9 ton/h)
Steam = 165°C, Heat source = 70°C

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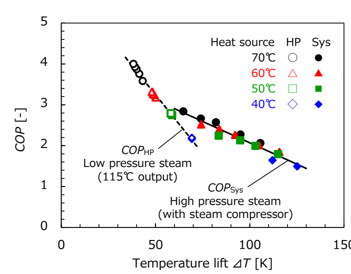
Performance Evauation of SGH165

- Performance evluation of SGH165
  - Tested at **CRIEPI** Lab in FY2014
  - Energy performance
    - Influence of heat source temperature, steam discharge pressure, feed water temperature
    - Part-load performance
    - Effect of waste heat recovery from drainage
  - Operating characteristics
    - Start-up, shut-down
    - Condensed water blow
    - Load following capability
- Interview survey about SGHs
  - Visited to installed sites in FY2015
  - To user companies and engineering companies



©CRIEPI Ref: T. Kaida et al., 9th International Conference on Compressors and their Systems, 2015. https://iopscience.iop.org/article/10.1088/1757-899X/90/1/012076 14

Interview Survey about SGHs

		Case 1	Case 2	Case 3
Outline	Industry	Chemical	Chemical (Medicine)	Waste Disposal
	Process	**Distillation** of ethanol	**Sterilization** of chemical container	**Drying** sewage sludge
	Installed HP	**SGH 120** × 5 units	**SGH165** × 1 unit	**SGH165** × 1 unit
	Installed year	2012	2014	2016
	Installation type	Renewal from steam boiler	Addition to existing steam line	Newly established
User company	Background	• Considered improvement measures • Suggestion from engineering company	• Considered improvement measures • Suggestion from engineering company	Not yet
	Objective (needed payback period)	• Reduction of running cost (**3.5 years with subsidy**)	• Reduction of running cost (**3 years without subsidy**)	Not yet
	Initial concerns	• Reliability and durability	• Durability	Not yet
	Effects	• **Better than planned**	• **As planned**	Not yet
	Requests	• Nothing	• Nothing	Not yet
Engineering company	Handling	• Easy because unitized	• Easy because unitized	• Easy because unitized
	Reason of selection	• Shorter delivery date and better operability compared to MVR	• Effective use of waste heat	• Effective use of waste heat
	Requests	• **Higher COP** • Smaller capacity (around half size) • Specification for explosion resistance	• **Lower initial cost** • Larger capacity (around 2 times size) • Specification for explosion resistance	• **Lower initial cost** • Increasing tolerance of water quality • Simpler drain piping

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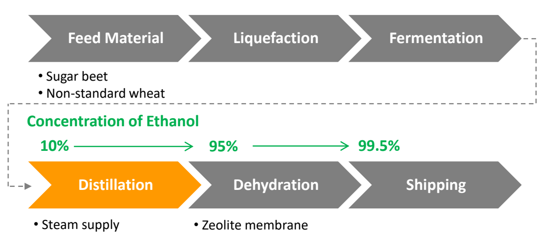
Case 1 | Distillation Process in Bio-ethanol Production

Outline of Installation

Industry	Chemistry
Production	Bio-ethanol
User company	Hokkaido Bioethanol Co., Ltd.
Installed year	2012
Process	Distillation of ethanol
Application	**Steam (120°C)**
Engineering company	Japan Chemical Engineering & Machinery Co., Ltd.
HP manufacturer	Kobe Steel, Ltd. (KOBELCO)
HP System	**SGH120** (× 5 units) Refrigerant: R245fa Steam flow: 2 ton/h
Effects	CO_2 reduction: 43%*1 Primary energy reduction: 40%*2 Energy cost reduction: 54%

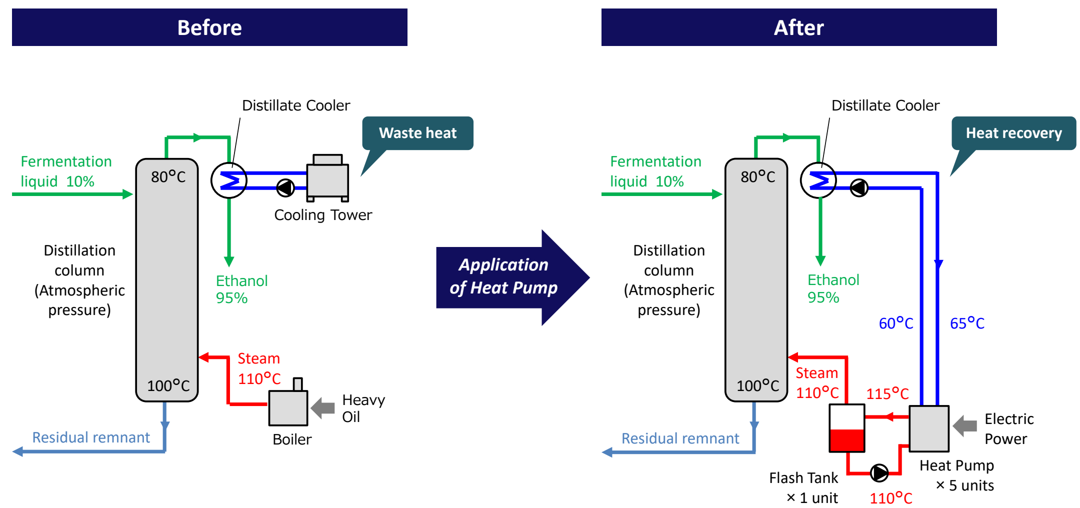
*1 Electricity: 0.681 kg-CO_2/kWh (Hokkaido EPCo, FY2013), Heavy oil: 2.71 kg-CO_2/L
*2 Electricity: 9.76 MJ/kWh, Heavy oil: 39.1 MJ/L

Process and Application



 Ref: JEHC, Collection of case studies for industrial electrification, Vol. 4, March. 2015. [in Japanese] 16

Case 1 | Distillation Process in Bio-ethanol Production



Ref: JEHC, Collection of case studies for industrial electrification, Vol. 4, March. 2015. [in Japanese]

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Interview Survey about SGHs

		Case 1	Case 2	Case 3
Outline	Industry	Chemical	Chemical (Medicine)	Waste Disposal
	Process	**Distillation** of ethanol	**Sterilization** of chemical container	**Drying** sewage sludge
	Installed HP	**SGH 120** × 5 units	**SGH165** × 1 unit	**SGH165** × 1 unit
	Installed year	2012	2014	2016
	Installation type	Renewal from steam boiler	Addition to existing steam line	Newly established
User company	Background	• Considered improvement measures • Suggestion from engineering company	• Considered improvement measures • Suggestion from engineering company	Not yet
	Objective (needed payback period)	• Reduction of running cost (**3.5 years with subsidy**)	• Reduction of running cost (**3 years without subsidy**)	Not yet
	Initial concerns	• Reliability and durability	• Durability	Not yet
	Effects	• **Better than planned**	• **As planned**	Not yet
	Requests	• Nothing	• Nothing	Not yet
Engineering company	Handling	• Easy because unitized	• Easy because unitized	• Easy because unitized
	Reason of selection	• Shorter delivery date and better operability compared to MVR	• Effective use of waste heat	• Effective use of waste heat
	Requests	• **Higher COP** • Smaller capacity (around half size) • Specification for explosion resistance	• **Lower initial cost** • Larger capacity (around 2 times size) • Specification for explosion resistance	• **Lower initial cost** • Increasing tolerance of water quality • Simpler drain piping

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Interview Survey about Steam Supply Heat Pumps

■ To engineering companies for distillation and concentration

	Company A	Company B	Company C
Strong field	Distillation of alcohol	Concentration of food and beverage	Water treatment
Estimated steam unit price	N/A	4,000 JPY/ton	5,000 JPY/ton
Estimated payback period	• Basically **3 years** • Sometimes 5 years with subsidy	• Basically **3 years** • Sometimes 5 years with subsidy	• 2 years for semiconductor industry • up to 5 years for food industry
Comparison to competing technologies	• **MVR** applied for large capacity (alcohol production: 200 kL/day) • **Double-effect evaporator** for middle capacity • **HP** considered for small capacity (alcohol production: 20–50 kL/day)	• **MVR** applied (steam 5–20 ton/h) • **SGH120** has the similar economic effect with **double-effect evaporator.** • **HP** will be considered for the processes **MVR** cannot apply: concentration of hydrofluoric acid, hydrochloric acid, etc.	• **MVR** considered first of all • **HP** will be considered for the processes **MVR** cannot apply: processes which may splash solvent or need high Boiling Point Rising (BPR), etc.
Market trends in Japan	• Number of newly-established or renewal of distillation columns: only several units/year in Japan • Small sized distillation columns use steam of several hundreds kg/h	• Intermediary material of food or medicine (ex. molasses)	• Increasing concentration needs for waste liquid treatment • Increasing factories of Zero Liquid Discharge (ZLD): concentration, drying, solidification

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Case 4 | Distillation Process in Dextran Production

Outline of Installation

Industry	Chemistry
Production	Dextran (Polysaccharides)
User company	Meito Sangyo Co., Ltd.
Installed year	2017
Process	Distillation of methanol
Application	**Hot water (90°C)**
Engineering company	Kimura Chemical Plants Co., Ltd.
HP manufacturer	Kobe Steel, Ltd. (KOBELCO)
HP System	**HEM-HR90** (× 2 units) Refrigerant: R134a+R245fa Heating capacity: 800 kW
Effects	CO_2 reduction: 60%*1 Primary energy reduction: 60%*2 Energy cost reduction: 63%

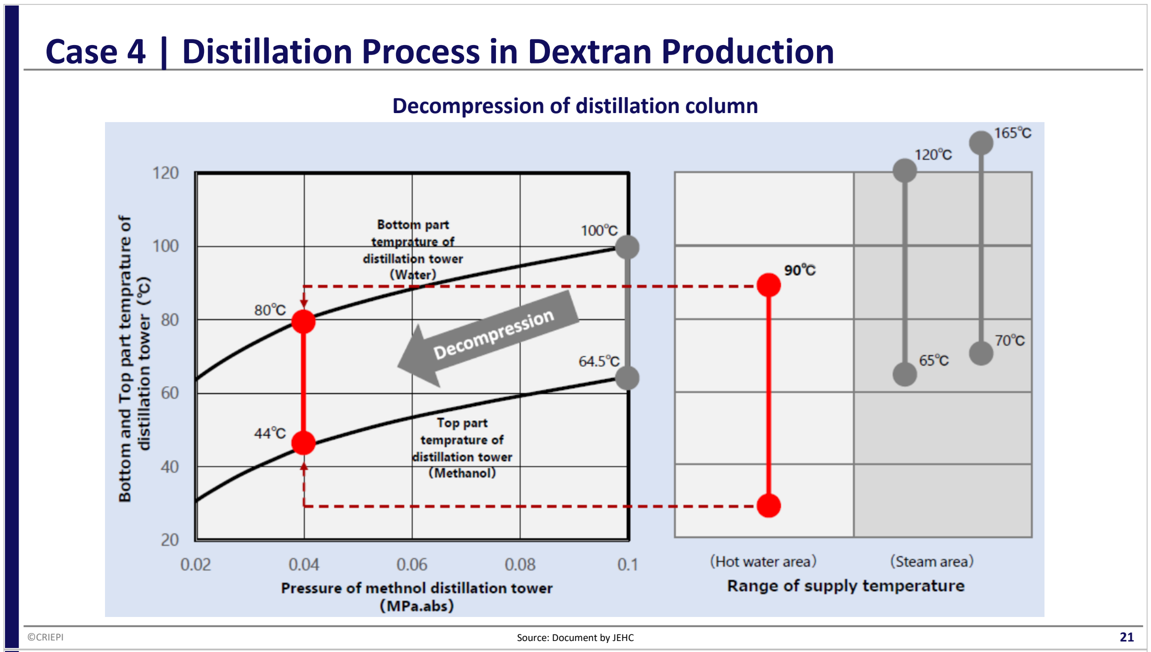
*1 Electricity: 0.491 kg-CO_2/kWh (TEPCO, FY2015), City gas: 2.29 kg-CO_2/Nm^3
*2 Electricity: 9.97 MJ/kWh, City gas: 45.0 MJ/Nm^3

Process and Application

Feed Material → Fermentation → Collection of Precipitation → Acid Hydrolysis
• Sugars
• Distillation of methanol

→ Collection of Precipitation → Vacuum Concentration → Spray Drying → Dextran (Powder)
• Distillation of methanol

 Ref: JEHC, Collection of case studies for industrial electrification, Vol. 6, March. 2019. [in Japanese] 20


Case 4 | Distillation Process in Dextran Production
Decompression of distillation column
Bottom and Top part temprature of distillation tower (℃)
120
100
80
60
40
20
0.02
0.04
0.06
0.08
0.1
Pressure of methnol distillation tower (MPa.abs)
Bottom part temprature of distillation tower (Water)
Top part temprature of distillation tower (Methanol)
100℃
80℃
64.5℃
44℃
Decompression
90℃
120℃
165℃
65℃
70℃
(Hot water area)
(Steam area)
Range of supply temperature
©CRIEPI
Source: Document by JEHC
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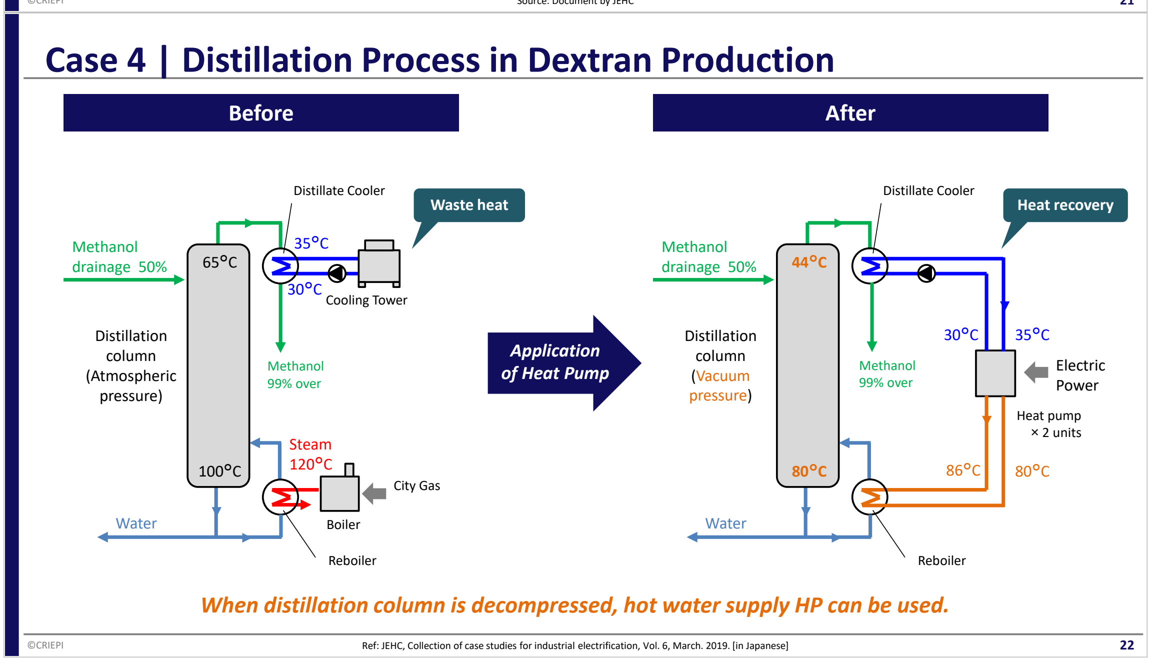
Case 4 | Distillation Process in Dextran Production
Before
After
Distillate Cooler
Waste heat
Methanol drainage 50%
65°C
35°C
30°C
Cooling Tower
Distillation column (Atmospheric pressure)
Methanol 99% over
Steam 120°C
100°C
City Gas
Boiler
Water
Reboiler
Application of Heat Pump
Distillate Cooler
Heat recovery
Methanol drainage 50%
44°C
Distillation column (Vacuum pressure)
Methanol 99% over
30°C
35°C
Electric Power
Heat pump × 2 units
80°C
86°C
80°C
Water
Reboiler
When distillation column is decompressed, hot water supply HP can be used.
©CRIEPI
Ref: JEHC, Collection of case studies for industrial electrification, Vol. 6, March. 2019. [in Japanese]
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Case 4 | Distillation Process in Dextran Production

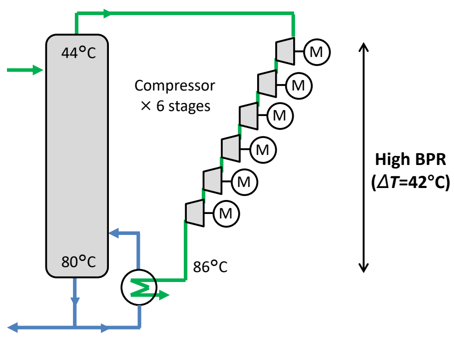
Why MVR was not selected?

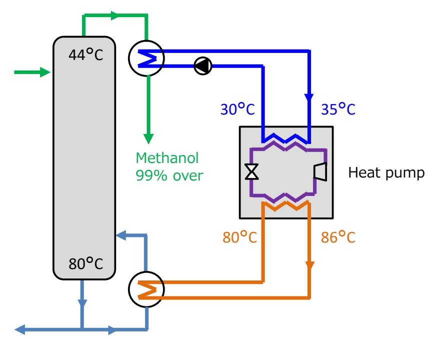
MVR

- Require a special specification compressor because of methanol vapor compression
- Require 6 stage compressors because of high BPR
→ high cost and complicated operation

Heat Pump

- Lower risk of methanol flammability
- Use a general-purpose heat pump
→ lower cost and better operability

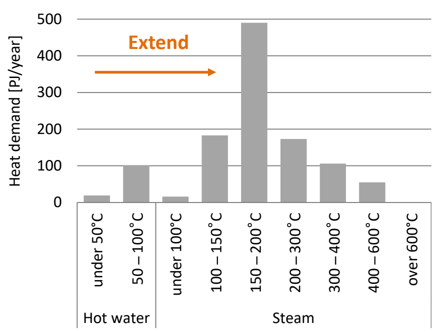


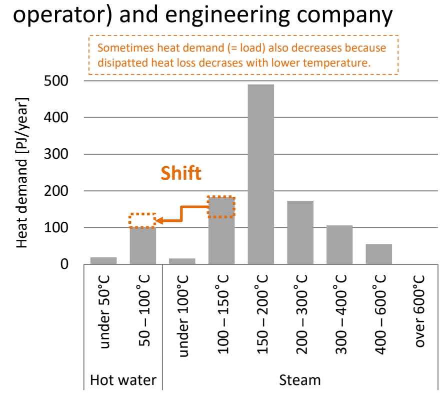


 Based on: Document by JEHC 23

For Spreading IHPs

- Higher Temperature HP
  - Extend the territory of HP application
  - Much potential
  - Not much customer's benefit yet
- Process Innovation
  - Shift heat demand and use not so high temperature HP for better COP
  - Need to involve customer (process operator) and engineering company





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Conclusions

Conclusions

- HTHPs potential in Japan
  - Industrial heat demand: **300 PJ/year under 150°C**
  - Expectation of spreading IHPs: **1,673 MW by FY2030** (over 150 times installation by FY2013)
- HTHPs development trends in Japan
  - **KOBELCO**: already prepared the product line-up **up to 175°C**
  - **Fuji Electric**, **MAYEKAWA** and **MHI Thermal Systems**: already prepared HTHPs **around 120°C**, developing HTHPs **over 150°C** (up to 200°C) by **NEDO** projects
- Electricity situation in Japan
  - Toward electrification and decarbonization, but not good situation at this time
  - CO_2 emissions factor (**0.5 kg-CO_2/kWh**) and economic factor (**E/G = 2.8**)
- Realistic another approach for applying IHPs at this situation
  - Shifting heat demand by **process innovation**
  - An example: decompression of distillation column and application of 90°C hot water supply HP

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Thank you for your attention.

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High temperature heat pumps in Austria: demonstration and application examples

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veronika.wilk@ait.ac.at

Keywords:
High temperature heat pump, diary, steel and rolling mill, utilities, drying, demonstration

Abstract

Austrian industry consumed 385 PJ of final energy in 2016 [1]. Approximately 25% thereof were covered by natural gas. It was used for industrial applications that are relevant for heat pumps, such as space heating and air conditioning, steam generation and industrial ovens. Space heating and air conditioning are typical fields of applications for heat pumps. Industrial ovens comprise all kinds of ovens ranging from low-temperature applications, such as drying to high-temperature processes, such as sintering. Steam generation also covers a broad range of temperatures. Both applications are therefore partially relevant for heat pumps. The integration of heat pumps into industrial processes is still in a rather early diffusion phase in Austria despite the large technical potential according to the national technology and implementation roadmap for heat pumps. This roadmap was developed in a comprehensive participatory stakeholder process and was published in 2016. It is based on the strengths of the Austrian heat pump sector and the users' needs. Industrial processes were identified as one of four main fields of applications for heat pumps. The recommendations for research and development institutions comprise the implementation of model solutions and pilot systems, heat pumps for higher supply temperatures and new concepts to enable widest possible market penetration. [2]

Three application examples of high temperature heat pumps are presented, covering the food industry, metal industry and utilities. All examples are brown-field installations, where the heat pumps were integrated into existing processes to recover waste heat from different sources, such as flue gas condensation, chillers and cooling water. In these examples, the heat provided by the heat pumps is fed into district heating grids. The supply temperatures range from 78 – 95°C, the heating capacities from 4 – 40 MW. These heat pumps were commissioned in the last four years, also reflecting the increasing spread of industrial heat pumps in Austria.

European legislation aiming at an increase in renewables in electricity supply and reduction of CO2 emissions, as well as further development of the technology according to the needs of industrial applications are important drivers to spread heat pumps in industry. Current research activities focus on high temperature heat pumps, new refrigerants and efficiency measures, as well as holistic planning approaches for industrial sites. Among other projects, DryFiciency, an H2020 project, is presented in

more detail. Two heat pump demonstrators are developed, constructed and operated in a real industrial environment. They are closed loop compression heat pumps operated on OpteonMZ supplying up to 400 kW heat at 160°C. The heat pumps are integrated in industrial drying process in two Austrian companies, Agrana Stärke GmbH (starch drying) and Wienerberger AG (brick drying). The heat pumps are currently about to be commissioned. Then, extensive monitoring of the operation will start to evaluate efficiency and other important process parameters, as well as stability of refrigerant and lubricant when exposed to high temperatures. There is increasing demand for industrial heat pumps in Austria, as they allow for waste heat recovery, efficiency increase and electrification and will therefore play a major role in the future energy system. With heat pumps that deliver high temperature heat up to 160°C, a larger range of applications in industry can be covered. To satisfy the needs of industry, high availability and short payback periods are required. It is therefore essential to come up with reliable and cost-efficient solutions for the technological challenges for high temperature heat applications, such as temperature resistant materials and components. Successful demonstration projects are an important basis to establish trust in new technologies and for further roll out.

References

[1] Statistics Austria, STATcube – Statistische Datenbank von STATISTIK AUSTRIA, Nutzenergieanalyse 2016 (useful energy analysis), accessed on 23.03.2018

[2] Hartl, M., Biermayr, P., Schneeberger, A., Schöfmann, P., Österrei-chische Technologie- und Umsetzungsroadmap für Wärmepumpen, Berichte aus Energie- und Umweltforschung Nr. 8/2016, im Auftrag des Bundesministeriums für Verkehr, Innovation und Technologie, Juni 2016

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HIGH TEMPERATURE HEAT PUMPS IN AUSTRIA:

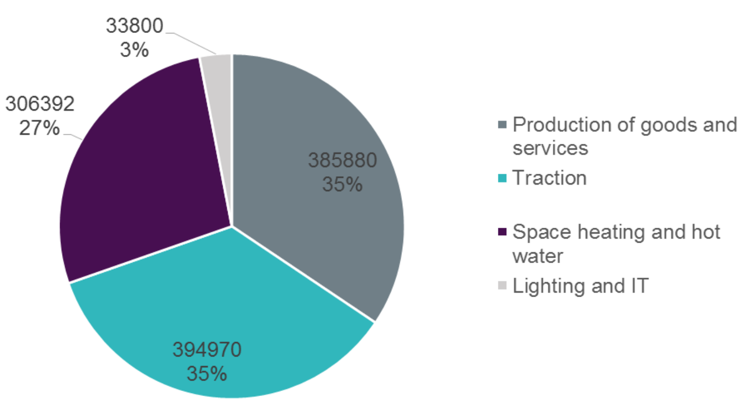
Demonstration and application examples

Veronika Wilk, Michael Lauermann, Franz Helminger,
Gerwin Drexler-Schmid, Thomas Fleckl

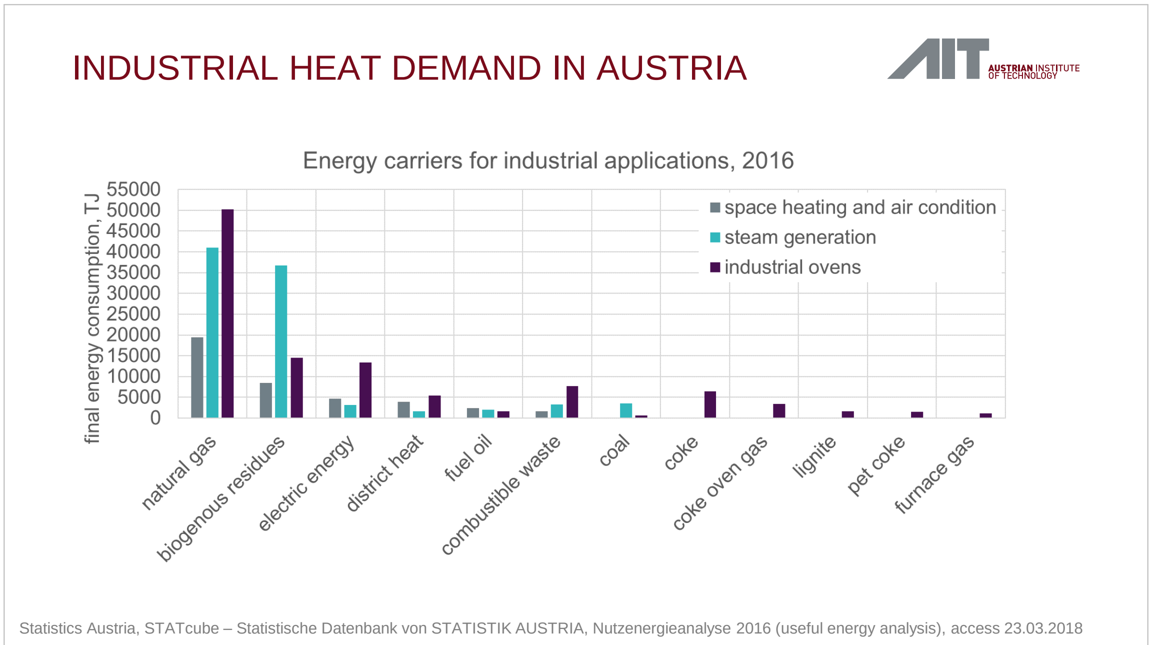
ENERGY USE IN AUSTRIA

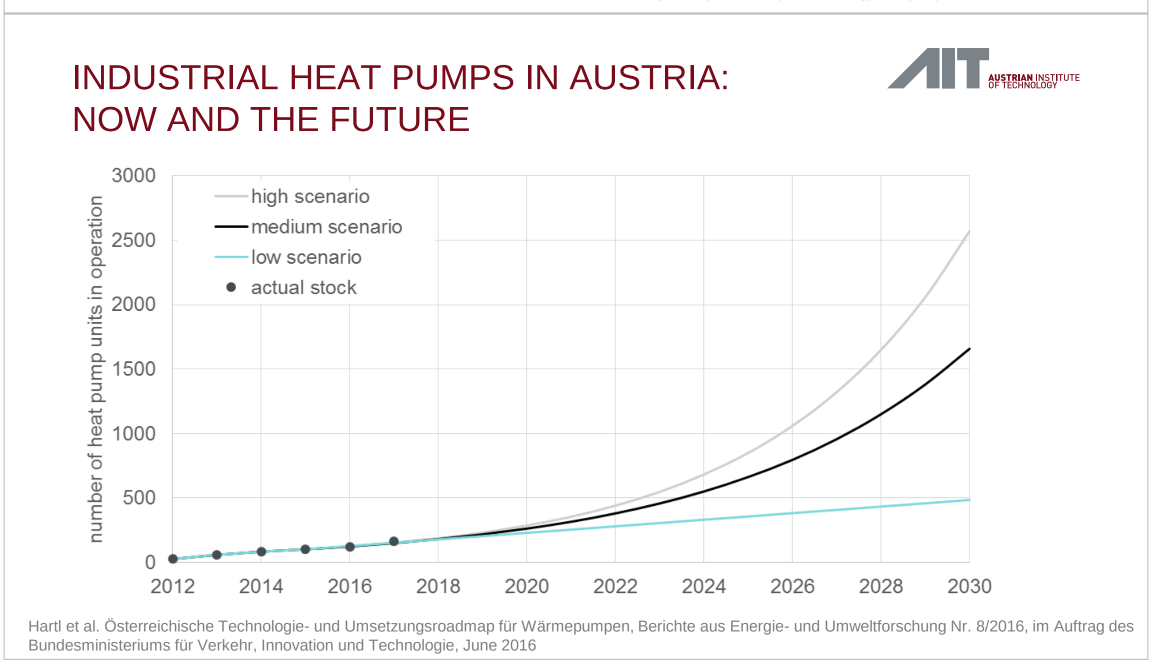
AUSTRIAN INSTITUTE OF TECHNOLOGY

Final energy consumption in Austria in 2016: 1 121 042 TJ



Statistics Austria, STATcube – Statistische Datenbank von STATISTIK AUSTRIA, Nutzenergieanalyse 2016 (useful energy analysis), access 23.03.2018


INDUSTRIAL HEAT DEMAND IN AUSTRIA
AUSTRIAN INSTITUTE OF TECHNOLOGY
Energy carriers for industrial applications, 2016
final energy consumption, TJ
55000
50000
45000
40000
35000
30000
25000
20000
15000
10000
5000
0
space heating and air condition
steam generation
industrial ovens
natural gas
biogenous residues
electric energy
district heat
fuel oil
combustible waste
coal
coke
coke oven gas
lignite
pet coke
furnace gas
Statistics Austria, STATcube – Statistische Datenbank von STATISTIK AUSTRIA, Nutzenergieanalyse 2016 (useful energy analysis), access 23.03.2018


INDUSTRIAL HEAT PUMPS IN AUSTRIA:
NOW AND THE FUTURE
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number of heat pump units in operation
3000
2500
2000
1500
1000
500
0
2012
2014
2016
2018
2020
2022
2024
2026
2028
2030
high scenario
medium scenario
low scenario
actual stock
Hartl et al. Österreichische Technologie- und Umsetzungsroadmap für Wärmepumpen, Berichte aus Energie- und Umweltforschung Nr. 8/2016, im Auftrag des Bundesministeriums für Verkehr, Innovation und Technologie, June 2016

TECHNOLOGY ROADMAP FOR HEAT PUMPS

- published in June 2016
- participatory stakeholder process
- focus on the strengths of the national heat pump sector and the users' needs

- four main fields of applications for heat pumps
  - residential and non-residential buildings
  - smart electric grids
  - thermal grids
  - **industrial processes**

Hartl et al. Österreichische Technologie- und Umsetzungsroadmap für Wärmepumpen, Berichte aus Energie- und Umweltforschung Nr. 8/2016, im Auftrag des Bundesministeriums für Verkehr, Innovation und Technologie, June 2016

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EXAMPLES FOR INDUSTRIAL APPLICATIONS

Food, metal, plastics, power plants…

FOOD INDUSTRY

Dairy

- Berglandmilch eGen / Tirol Milch Wörgl
- joint project with Stadtwerke Wörgl
- installed by Frigopol in 2015

3 heat pumps with a total

- cooling capacity: 3.2 MW
- heating capacity: 4.2 MW

- heat source: flue gas condensation and waste heat from chillers, up to 45°C
- heat sink: 78°C, for district heating

Photo: Frigopol Hochtemperatur-Wärmepumpen, http://www.frigopol.com/wp-content/uploads/54b8ce0cdfdd6.pdf

Further details: A. Baumhakel, Frigopol Kälteanlagen GmbH, www.frigopol.com

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METAL INDUSTRY

Rolling Mill

- steel and rolling mill Marienhütte GmbH
- Energie Graz GmbH & Co KG

2 heat pumps (Friotherm) with a total heating capacity of 11 MW

- heat source: process waste heat (cooling baths)
- heat sink: district heating at 70 and 95°C, residential area (Graz City center and Reininghaus)

Photo: http://www.energie-graz.at/energie/fernwaerme/projekte/reininghaus , 10.05.2017,
Arnitz et al., Waste Heat Recovery at the Steel and Rolling Mill "Marienhütte" Graz (Austria), Heat Pumping Technologies Magazine, Vol37, No2, 2019, https://heatpumpingtechnologies.org/publications/waste-heat-recovery-at-the-steel-and-rolling-mill-marienhutte-graz-austria/

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POWER PLANT

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Wien Energie

- Power Plant Simmering (installed capacity of 1.2 GWel / 1 GWth)
- start up by end of 2018

2 heat pumps (Friotherm) with a total heating capacity of 27 - 40 MW

- heat source: cooling circuit of the power plants, river water also possible, 6-27°C
- heat sink: district heating, up to 95°C
- average COP: 3

C. Segalla, Power 2 Heat Anwendungen für das Fernwärmenetz, Großwärmepumpenforum Vienna, May 2018.

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INDUSTRIAL HEAT PUMPS: NOW AND THE FUTURE

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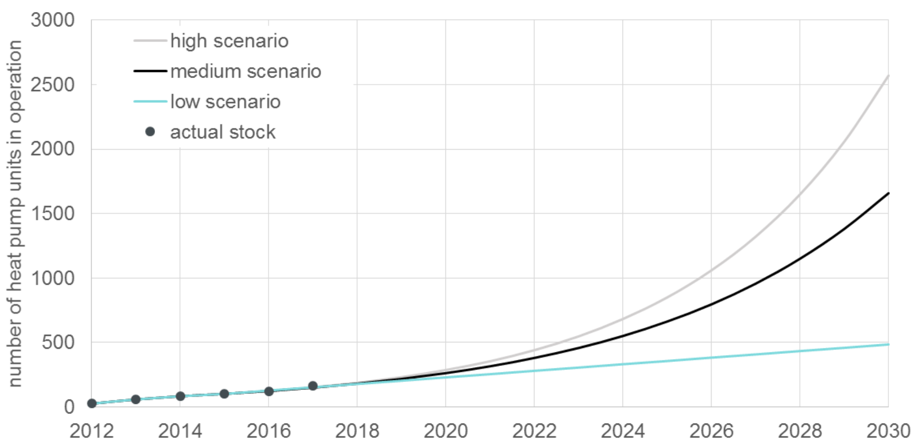

Hartl et al. Österreichische Technologie- und Umsetzungsroadmap für Wärmepumpen, Berichte aus Energie- und Umweltforschung Nr. 8/2016, im Auftrag des Bundesministeriums für Verkehr, Innovation und Technologie, June 2016

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RESEARCH AND DEMONSTRATION ACTIVITIES

AIT Austrian Institute of Technology

RESEARCH QUESTIONS: HIGH TEMPERATURE HEAT PUMPS

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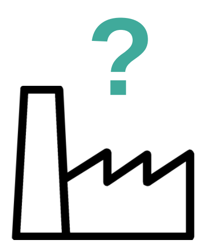
Challenges:

- refrigerants
- temperature resistant materials and components
- heat sources with high temperatures
- efficient cycles, e.g. pressure recovery

Requirements for industrial applications:

- high availability and reliability
- short payback time



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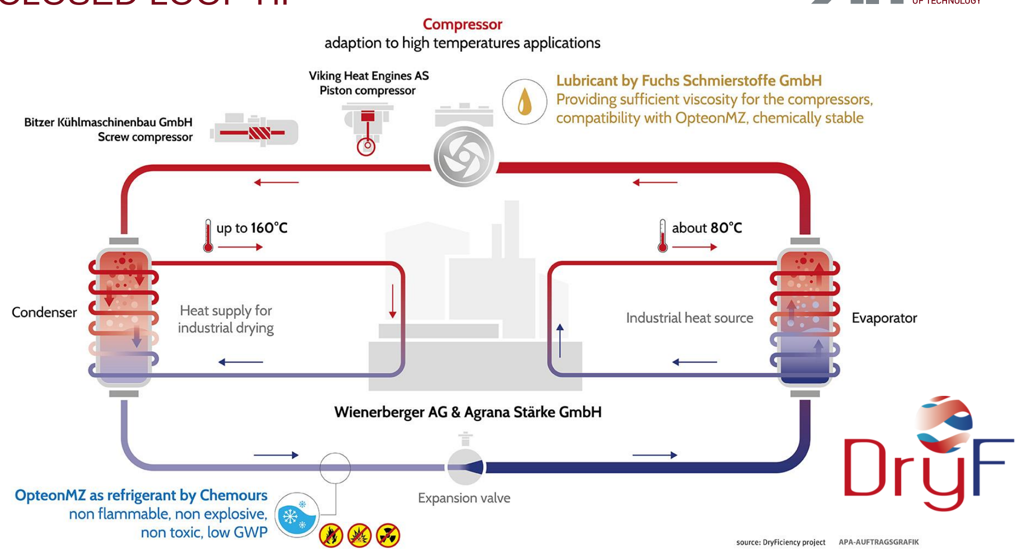
DRYFICIENCY

- H2020 project with 13 partners and 6 Mio € funding
- development and demonstration of three high temperature heat pumps in industrial drying processes
- air drying with closed loop heat pumps
- steam drying with open loop heat pump (mechanical vapor recompression)

The project has received funding from the European Union's Horizon 2020 programme for energy efficiency and innovation action under grant agreement No. 723576.

CLOSED LOOP HP

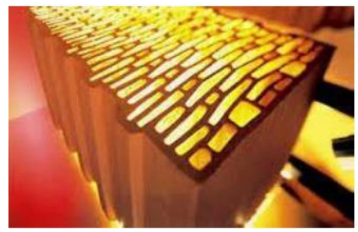


DEMO SITES

STARCH

AGRANA Stärke GmbH
Pischelsdorf (AUT)

BRICKS



Wienerberger AG
Uttendorf (AUT)

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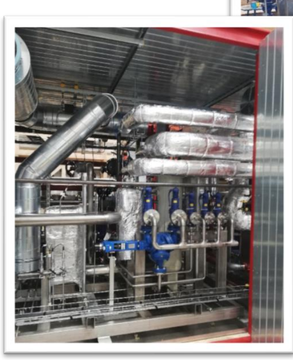
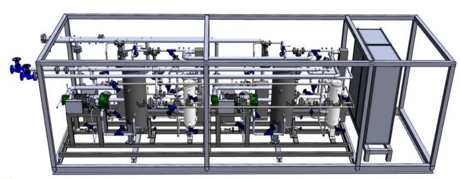
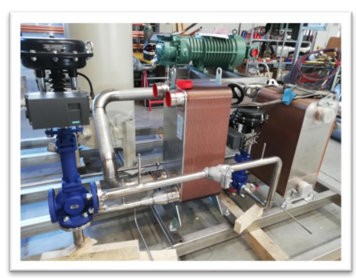
HEAT PUMPS UNDER CONSTRUCTION







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INTEGRATION IN THE BRICK FACTORY
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DryF
Wienerberger
Building Material Solutions
16/09/2019
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HP FOR STARCH DRYING
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DryF
AGRANA
STÄRKE
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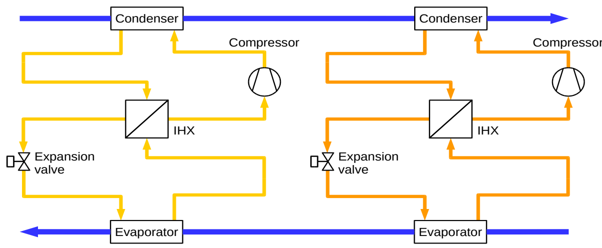
INSIDE…



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NEXT STEPS

- commissioning and start up
- test different operation conditions (heat supply temperatures, part load conditions, etc.)
- monitoring of refrigerant and lubricant quality
- evaluation of efficiency and other important process parameters

www.dryficiency.eu

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NATURAL REFRIGERANTS

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Concept of a new butane high temperature heat pump

FRIGOPOL klima+ energie fonds SINTEF

- lift industrial waste heat from 60 °C to 130 °C
- validated Dymola model for the heat pump
- ejector utilization
- CFD for heat & mass transfer in heat exchangers
- experimental validation

Results

- successful operation of the functional model in the lab
- performance increase with ejector
- validated CFD model

G. Drexler-Schmid et al.: Messung und Simulation einer 50 kW Butan-Hochtemperaturwärmepumpe mit Ejektor, DKV Tagung Kassel 2016.

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RESEARCH QUESTIONS: PROCESS INTEGRATION

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Planning process of industrial heat pumps:

- interaction of multiple stakeholders

Optimization of industrial sites:

- interaction of multiple heat suppliers, storages and consumers
- design optimization
- operation optimization

Method development:

- dynamic simulations: interaction of heat pump and process
- mathematical programming: complex and dynamic systems, discontinuous processes

16/09/2019

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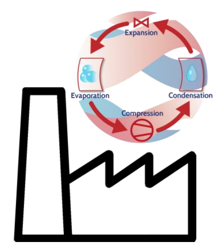
CONCLUSIONS

- increasing interest from industry in industrial heat pumps
  - expect high availability and reliability
  - short payback periods
- solution of technological challenges for high temperature applications essential
  - suitable refrigerants
  - heat sources with high temperatures
  - temperature resistant materials and components
- industrial heat pumps allow for economic and environmental benefits
  - increase on-site efficiency and contribute to decarbonization
  - technology readily available on the market
  - demonstration projects for new developments



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TOMORROW TODAY

THANK YOU!

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Combined Heating and Cooling: Integrated Ammonia-Water Heat Pump in Modern Dairy Production

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Keywords:
High temperature heat pump, hybrid heat pump, R717/R718, heat integration, heat recovery

Abstract

Introduction

TINE SA in Norway continues their work modernizing the dairy industry with their new dairy located at Flesland in Bergen. The new dairy will produce milk, juice and cream.

TINE has been working to find the most modern solutions for energy use and the environment. All heating and cooling are supplied by cooling machines and heat pumps in interaction. TINE is the winner of the Norwegian Heat Pump Award 2019 and the EHPA "Heat Pump City of the Year" Award 2019 in the Decarb Industry Category with this plant.

Hybrid Energy AS emerged from Institute for Energy Technology (IFE) in Norway and was founded in 2004. Hybrid Energy AS have commissioned plants in dairies, slaughterhouses, fish feed producers, biogas production plants, district heating plants and process industry. In total 17 high temperature hybrid heat pumps systems are commissioned, with over 500.000 hours of operation.

The chosen heat integration solution and the high temperature hybrid heat pump is described, and initial operational results for the hybrid heat pump are reported.

Choosing the best heat integration of heating and cooling

The traditional method for providing cooling and heating in a dairy has been to separated cooling and heating. The cooling demands are delivered by chillers and the heating demands are delivered by fossil fuelled boiler providing steam or pressurised hot water. A certain amount of heat recovery was possible.

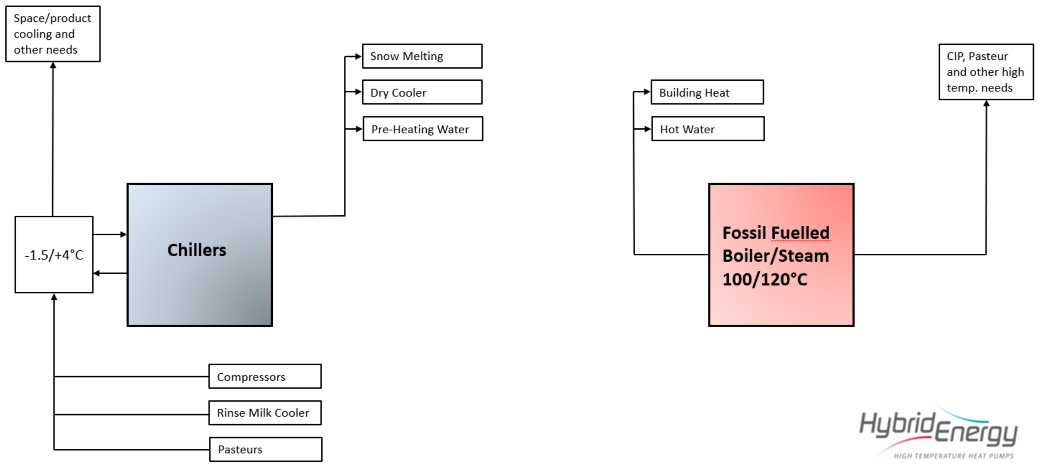


Figure 1 Traditional dairy: Separated Cooling and Heating

At TINE's new dairy a fully integrated cooling and heating system has been installed, see Figure 2. The hot and cold side is connected using thermal storage tanks and heat pumps. Heat is recovered from the chiller condenser side and transferred to useful temperatures by means of two ammonia heat pumps at intermediate temperatures and by a single- stage hybrid heat pump system at high temperature.

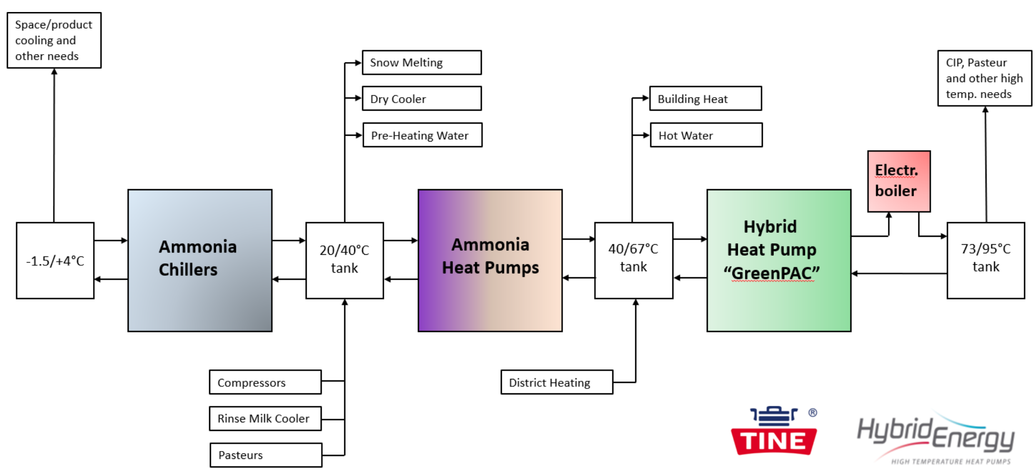


Figure 2 Tine Bergen: Integrated Energy Recovery Using Heat Pumps

Technical details of main components

Table 1 gives an overview of the main components in the cooling and heating system.

Table 1 Main components

Component	Specification
Hybrid heat pump	Capacity: 940kW Heat Source Inlet/Outlet Temp.: 67/60°C Heat Sink Inlet/Outlet Temp.: 83/95°C
Ammonia heat pumps	Total capacity (two HP's): 1577kW Heat Source Inlet/Outlet Temp.: 67/60°C Heat Sink Inlet/Outlet Temp.: 83/95°C
Chiller plant	Total capacity (three Chillers): 2400kW Heat Source Inlet/Outlet Temp.: 4/-1,5°C Heat Sink Inlet/Outlet Temp.: 20/40°C
Thermal storage tanks	Ice water (+0,5°C): 60m³ Glycol (-1,5°C): 60m³ Process water (20°C): 130m³ Process water (40°C): 130m³ Process water (40°C): 130m³ Process water (67°C): 130m³ Process water (95°C): 130m³

Hybrid heat pump

The Hybrid technology is based on an absorption process and a compression process, utilizing a mixture of water and ammonia. A Hybrid Heat Pump is built with standard ammonia compressors, with a design pressure of 25 bar. A traditional heat pump using pure ammonia, can heat water to 50°C at this pressure. A Hybrid Heat Pump can heat water to 120°C using the exact same equipment. It can cover a whole new range of temperatures than traditional heat pumps, meeting the demands of a lot of industrial processes.

Figure 3 displays a simplified flow diagram with the main components of a Hybrid Heat Pump. The working circuit contains a solution pump, pumping a solution low in ammonia from low to high pressure, and a compressor compressing ammonia gas. In the absorber/condenser the solution absorbs the ammonia gas, and heat is released through both absorption and condensation. The solution exiting the absorber/condenser is rich with ammonia and passes through an expansion valve where the pressure drops. As the solution enters the desorber/evaporator, ammonia is boiled out of the solution when heat is absorbed from the heat source, and the process is repeated.

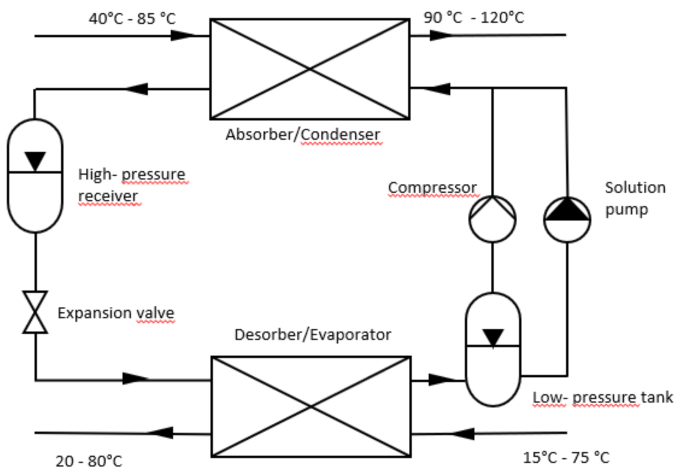


Figure 3 Sketch of hybrid heat pump

The hybrid heat pump system installed at the new dairy at Tine Bergen is at single stage system, i.e. pressurisation of the vapour coming from the low-pressure tank is done in one compressor. The hybrid heat pump was installed early in 2019 and has been in operation for a total of 737 hours. The total delivered heat by the heat pump is 319 MWh, giving an average effect of 432kW. The average coefficient of performance over the total operational time is 5.6 while delivering hot water at a temperature of 95-97°C.

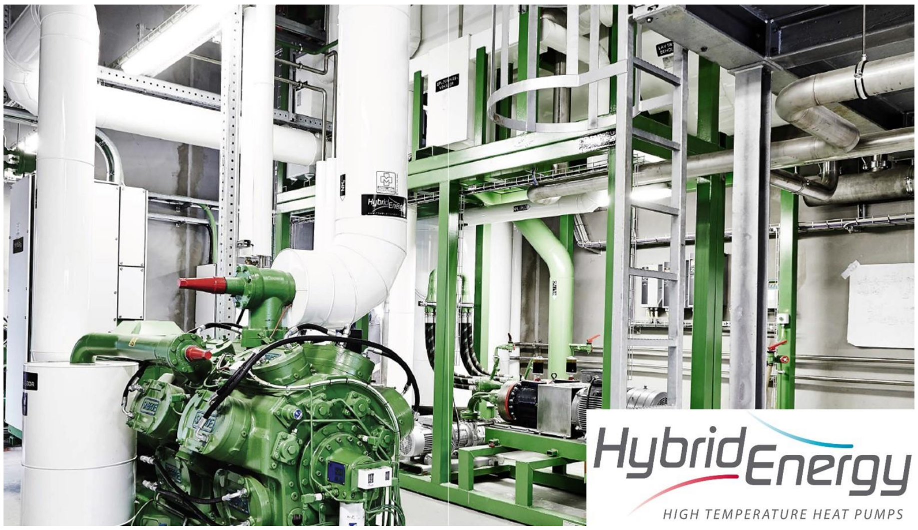
Table 2 Hybrid heat pump performance

Operational time	737 hours
Delivered heat	319 MWh
Mean effect	432 kW
Power consumption	57 MWh
Output temperature	95-97°C
COPheating	5.6

Key success factors

The key success factors for the project are cooperative work between TINE, SINTEF through HeatUP and HighEff research projects, state funding by ENOVA, engineering companies (Sweco, AF Group) and suppliers (Krones AG, Milkron GmbH, Hybrid Energy AS, Johnson Controls Norway)

The solutions are based on conventional knowledge, but the overall composition ensures that the project goes beyond the best available technology.



Combined Heating and Cooling: Integrated Ammonia-Water Heat Pump in Modern Dairy Production

Dr. Stein Rune Nordtvedt, CTO

Hybrid Energy AS

- Emerged from Institute for Energy Technology (IFE) in Norway
- Founded in 2004
- Commissioned plants in dairies, abattoirs, fish feed producers, biogas production plants, district heating, process industries

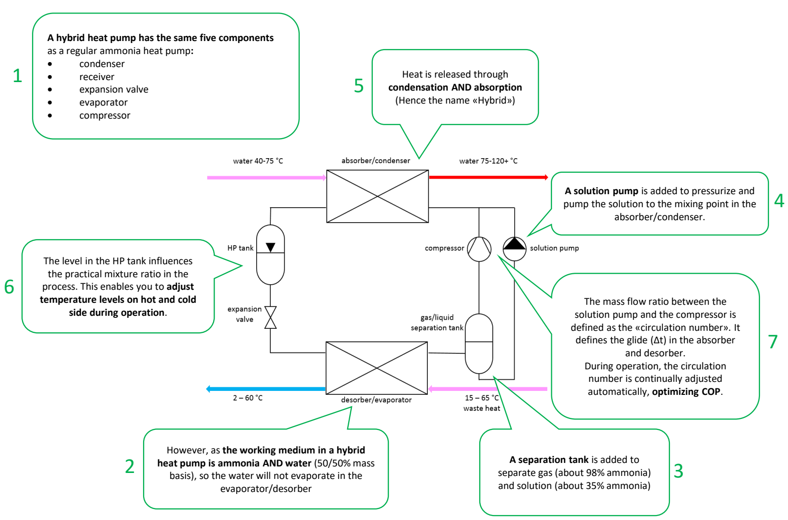
- 17 High Temperature Heat Pump systems commissioned.
- Proven technology with over 500.000 operational hours

What is a Hybrid Heat Pump?

- Natural working medium (50/50 water and ammonia)
- Can deliver 120 °C at low system pressures
- Yields exceptional COP's, especially with large glides (Δt's) on hot and cold side
- Uses Standard refrigeration equipment
- Offers unique flexibility <u>after</u> commissioning

HybridEnergy
HIGH TEMPERATURE HEAT PUMPS

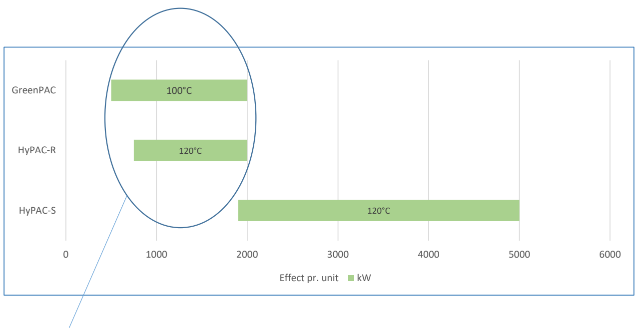
The Technology: Basics



HybridEnergy
HIGH TEMPERATURE HEAT PUMPS

The Hybrid Heat Pump Family

1. The GreenPAC
2. The HyPAC-R
3. The HyPAC-S



Designed for dairy purposes

HybridEnergy

The GreenPAC

Typical applications:

- Washing
- Pasteurization
- CIP
- Process heat
- General heating

Source Temperature:	60°C
Output Temperature:	100°C
Output Effect Range:	500 – 2000 kW
COP:	5-8

HybridEnergy

The HyPAC-R

Typical applications:
- Pressurized water circuits
- Boiler replacement
- Pasteurization
- CIP
- Process heat
- General heating

Source Temperature:	60°C
Output Temperature:	120°C
Output Effect Range:	750 – 2000 Kw
COP:	3-5

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HIGH TEMPERATURE HEAT PUMPS

TINE SA - In Brief

- Cooperative owned by Norwegian milk producers
- 31 dairies in Norway and 7 subsidiaries in six foreign countries.
- Main products: milk, cheese and other dairy products
- Annual turnover in 2018: NOK 23,5 Billion.
- Employees: 5355.

TINE®

HybridEnergy

HIGH TEMPERATURE HEAT PUMPS

TINE Bergen: «The greenest dairy in Europe»

- "Modernizing the dairy industry"
- Worked to find the most modern solutions for energy use and the environment
- All heat and cooling supplied by cooling machines and heat pumps in interaction
- The dairy will have 6000 m^2 of solar cells on the roof to produce its own electricity
- First dairy complete without fossil or direct electric heating
- 40% reduction (approx. 5GWh) in energy consumption
- NOK 16.4 million in support from Enova
- TINE is the winner of the Norwegian Heat Pump Award 2019 and the EHPA "Heat Pump City of the Year Award 2019" The Decarbindustry category

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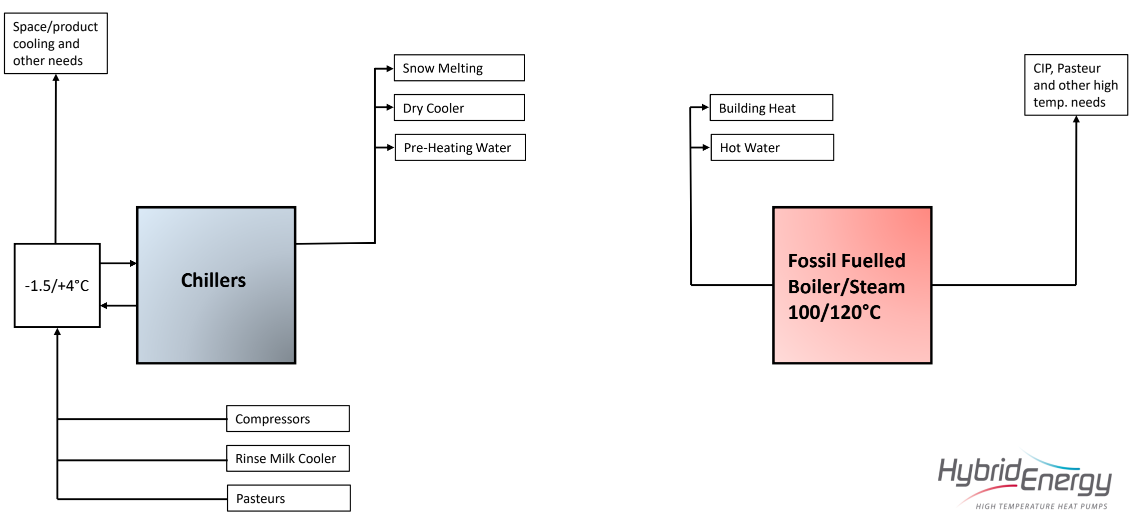
Traditional Dairy – Separated Cooling/Heating



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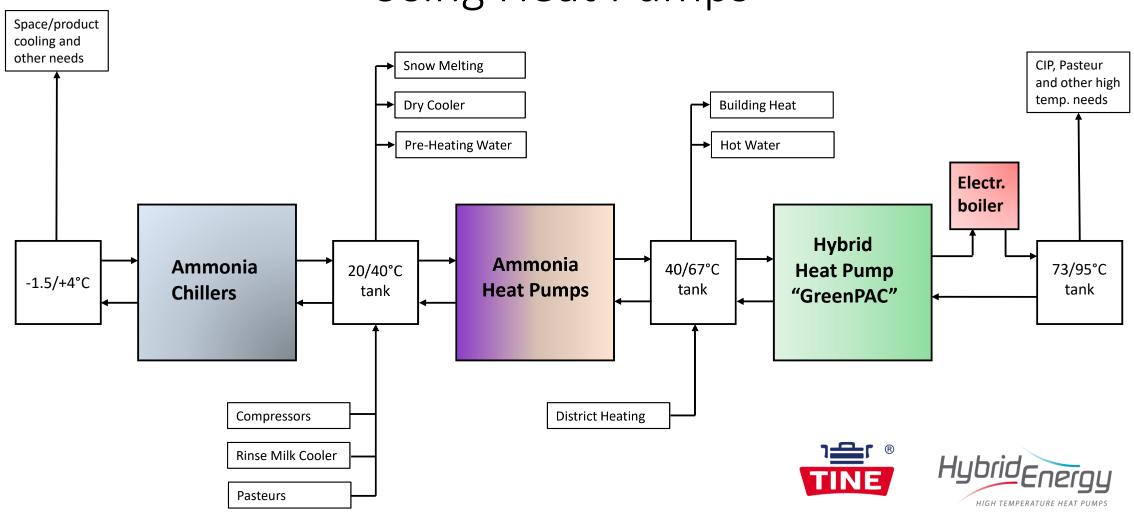
HIGH TEMPERATURE HEAT PUMPS

TINE Bergen – Integrated Energy Recovery Using Heat Pumps



TINE Bergen – Technical details

- Hybrid Heat Pump:
  940 kW, 67/60 °C, 83/95°C
- Ammonia Heat Pumps (1&2):
  1577 kW, 40/20 °C, 40/67 °C
- Chiller plant (3 Chillers):
  2400 kW, +4/-1,5°C, 20/40°C

Thermal Storage Tanks:

- 95°C: 130 m^3
- 67°C: 130 m^3
- 40°C: 130 m^3
- 20°C: 130 m^3
- -1,5°C: 60 m^3
- +0,5°C: 60 m^3

TINE Bergen – Hybrid Heat Pump Performance

• Operational hours	737 hours
• Delivered heat	319 MWh
• Mean effect	432 KW
• Power consumption	57 MWh
• Output Temp.	95-97°C
• $COP_{heating}$	5.6

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HIGH TEMPERATURE HEAT PUMPS

TINE Bergen – Key Success Factors

- Research projects like HighEff/HeatUp
- State aid from ENOVA
- Research partners like SINTEF
- Suppliers and cooperative partners like:
  - Sweco and AF Group consultants and building contractors
  - Krones AG and Milkron GmbH
Machinery installations. Process equipment and Energy central
  - Hybrid Energy AS
Delivery of Hybrid Heat pump, Heat pumps and Refrigerant plant including control systems.
  - Johnson Controls AS and Sabroe
Manufacturing of Components for Energy Central

The solutions are based on conventional knowledge, but the overall composition ensures that the project goes beyond the best available technology.

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HIGH TEMPERATURE HEAT PUMPS

Thank you for your attention!

HybridEnergy

Hydrocarbon Heat Pumps with Combined Process Cooling and Heating at 115°C

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Keywords: High temperature heat pump, Hydrocarbon, Compressor technology, Cascade, Cooling, Heating, Ice water, Diary

Extended Abstract

Introduction

Heat pumps are key technology to decarbonize industry, increase its energy efficiency and reduce operation expenses. Process heat demands up to 150 °C are estimated to 172 TWh/year on European basis (Nellissen and Wolf 2015). Process heat can be supplied utilizing process waste heat and renewable electricity applying heat pumps up to about 95 °C with common technology (TRL > 8). High temperature heat pumps (HTHP) are defined as having supply temperatures above 100 °C (Arpagaus, Bless et al. 2018). Market ready technologies are scare (Elmegaard, Zühlsdorf et al. 2017) and HTHP development trends are: i) extending the limits of heat supply temperature to higher values, ii) improving HTHP efficiency, iii) applying new environmentally friendly refrigerants and iv) increase the TRL level from 3 to 7/8, lab scale to industrial pilot scale, respectively.

The Norwegian diary producer TINE SA operates 31 dairies spread right across Norway. TINE aims to produce all dairy products with renewable energy in 2025. Waste heat recovery and increasing energy efficiency are key elements to phase out fossil fuel based thermal energy supply of 133 GWh/a of the total 503GWh/a, on 2014 basis (Lovas, Elmegaard et al. 2017). Large quantities of the heat demand is in the temperature range of 90 °C to 180 °C. The processes in a diary such as: sterilisation and pasteurisation, require product heating which is followed by product cooling. This work focuses on technology development for combined process heating and cooling. The temperature range is about 0 °C and 115 °C, at heat source and heat sink side, respectively. However, the available waste heat from dry-coolers is also considered as heat source, reducing the temperature required temperature lift to about 90 K.

The SkaleUp-project will develop and demonstrate an innovative, proven heat pump concept into a universal, modular industrial energy system.

Results and Discussion

In order to enable the high temperature lift required for the process integration a cascade-HTHP system was selected with the natural working fluids propane (R290) in the low temperature cycle (LTC) and butane (R600) in the high temperature cycle (HTC), as depict in Figure. 1. The working fluid selection is detailed described in (Bamigbetan, Eikevik et al. 2016, Bamigbetan, Eikevik et al. 2018).

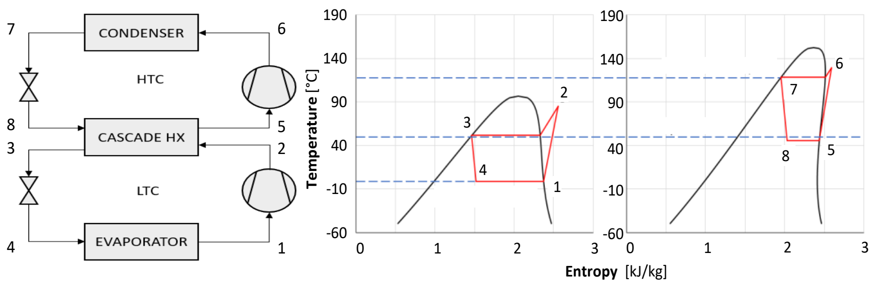

Figure 1 Simplified cycle and T-S diagram of R290/R600 cascade-HTHP

A 1:10 lab-scale prototype was built in accordance to DIN EN 378-1:2008. LTC and HTC components are "of the shelve", whereas the Butane compressor was a prototype. The system performance was evaluated in a heat source outlet (HSOin) and heat sink outlet (HSIout) temperature range of 3.7 °C to 26.9 °C and 110 °C to 117 °C, respectively. The highest temperature lift achieved was 119 K determined as difference between condensation and evaporation temperature. The use of both heat source and heat sink side requires the definition of a combined coefficient of performance $COP_{HSI+HSO}$ for system performance evaluation. The dependency of combined $COP_{HSI+HSO}$ and evaporation temperature is shown in Figure 2a. The estimated COPs of the 300 kW prototype system for the combined heating and cooling integration (Concept A) and utilisation of waste heat from dry-cooler (Concept B) are shown visualised in Figure 2b.

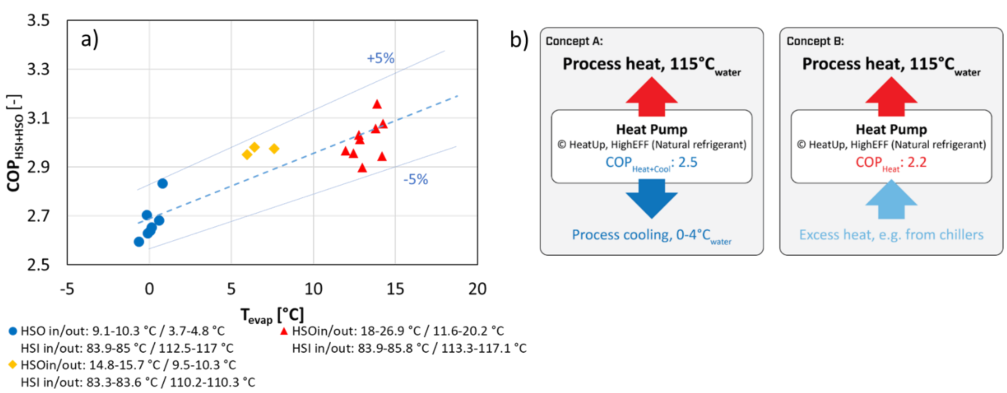

Figure 2 a) System performance expressed as combined COP ($COP_{HSI+HSO}$) in dependency of evaporation temperature (T_{evap}), b) Possible integration concepts with its estimated coefficients of performance for the 300kW prototype.

The active utilisation of heat source due to ice water production increases the system performance compared to the upgrade of waste heat from the dry-coolers, despite the increase of temperature lift. The Carnot efficiency of the system was nearly constant at 0.45 to 0.5.

In order to evaluate the energy and CO_2 saving potential of the HTHP a reference system comprising of gas fired boiler and ammonia chiller, with COP equal to 4.5, was assumed. The analysis indicates a 57% and 50% primary energy saving for Concept A and B, respectively. Reducing carbon emissions by about >90 %, considering Norwegian energy mix with 22 g_{CO2}/kWh.

Conclusion

The simultaneous needs of ice water and process heat at a diary require the integration of a highly efficient high temperature heat pump lifting 115 K, from a heat source temperature of about 0 °C to a heat sink temperature of 115 °C.

The experimentally conducted system performance evaluation (1:10 scale) underlined the possibilities of either producing ice water directly or utilizing waste heat from existing dry-coolers with sufficient high COPs of 2.5 and 2.2, for combined ice- and process hot-water production and dry cooler waste heat utilisation, respectively. Primary energy and CO_2-emission saving potential are in the order of 50 % and 97 % compared to fossil fuel-based process hot-water production. The saving will be approximately 1 GWh primary energy per year per unit, the equivalent energy consumption of 50 Norwegian households.

Acknowledgements

The authors would also like to acknowledge the support of The Research Council of Norway and the industrial partners through the grant NFR-296374 (SkaleUp), NFR-243679 (HeatUp) and NFR-257632/E20 FME-HighEFF (Centre for Environment-friendly Energy Research).

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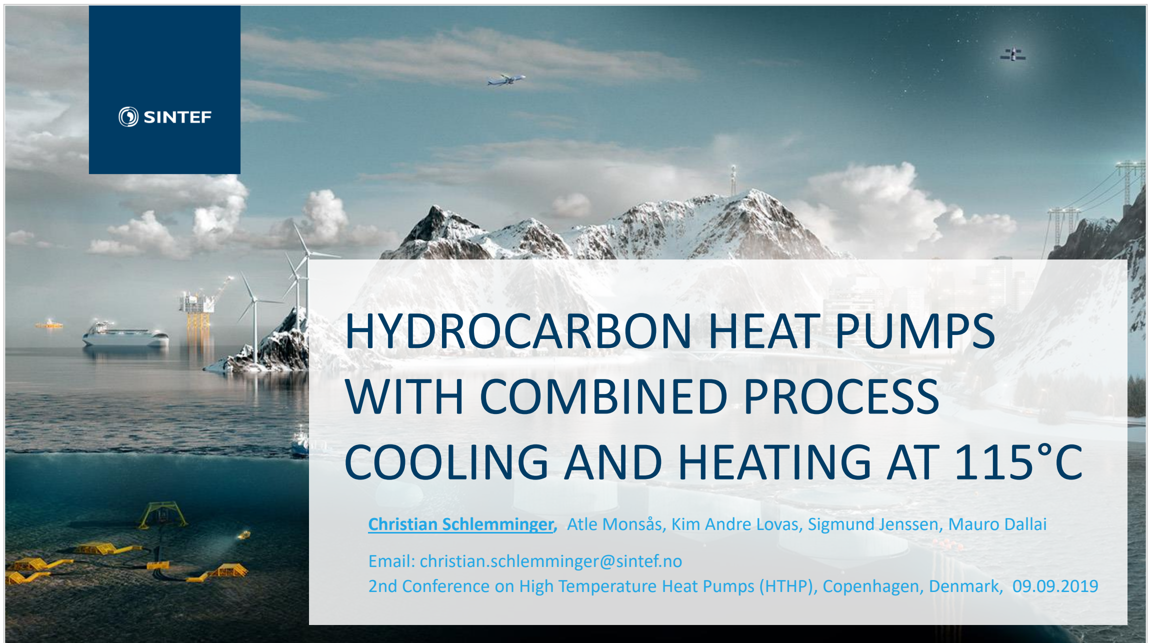
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HYDROCARBON HEAT PUMPS WITH COMBINED PROCESS COOLING AND HEATING AT 115°C
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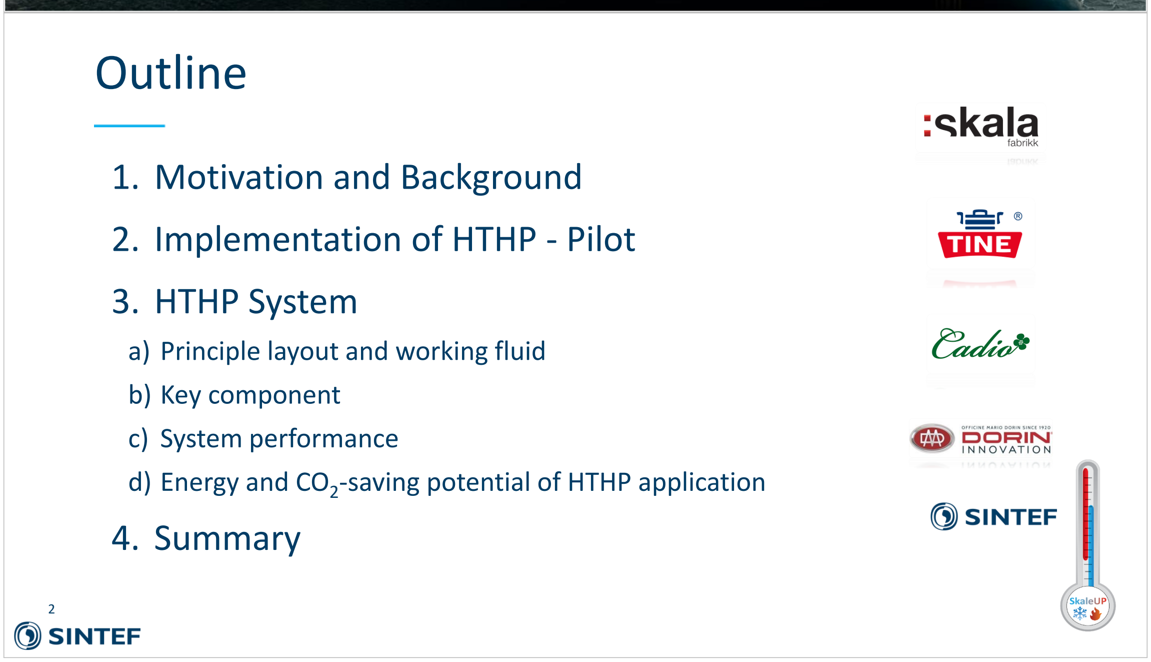
2nd Conference on High Temperature Heat Pumps (HTHP), Copenhagen, Denmark, 09.09.2019


Outline
1. Motivation and Background
2. Implementation of HTHP - Pilot
3. HTHP System
a) Principle layout and working fluid
b) Key component
c) System performance
d) Energy and CO2-saving potential of HTHP application
4. Summary
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2
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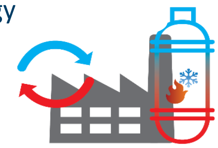
Challenges are opportunities – Low-emission concepts combined with energy-efficient solutions will strengthen the industry

- Industrial process sectors heavily relies on fossil fuel
- Climate targets are not reached due to lack of market ready technology
- Long return of investment for climate-friendly energy supply
- Low temperature excess heat is not utilized
- Awareness of HTHP-technologies little
- Demand for natural refrigerants as consequence concepts of the F-Gas regulation



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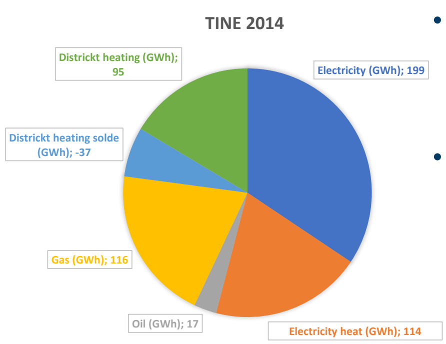
Motivation – HTHP enables industry to increase energy efficiency and reduce GHG-emissions



- **TINE aims to produce all diary products with renewable energy in 2025**
- **Situation for diary production in 2014**
  - Total energy consumption 503 GWh
  - Heating 305 GWh
  - Large demand for heating in temperature range of 90°C to 180°C

→ HTHPs and HPs can help TINE to achieve this goal

SkaleUP
Sustainable and efficient heat pump development for combined process heat and cool

4

HTHP - Development

- Utilize low temperature **excess or waste heat as valuable heat source**
- **HTHP are enabler** for:
  - **Increase energy efficiency** of processes, by utilizing waste heat and simultaneous coverage of cooling and heating demand
  - **Reduced OPEX for process plant**
  - **Reduce CO_2-emissions**
- Potential for utilization through
  - Novel cycle layouts and concepts
  - Utilization of **natural working fluids**
    → **long term sustainable**
  - Improved component design
- Pushing HTHPs from the laboratory scale (TRL4) **towards industry pilot (TRL7)**

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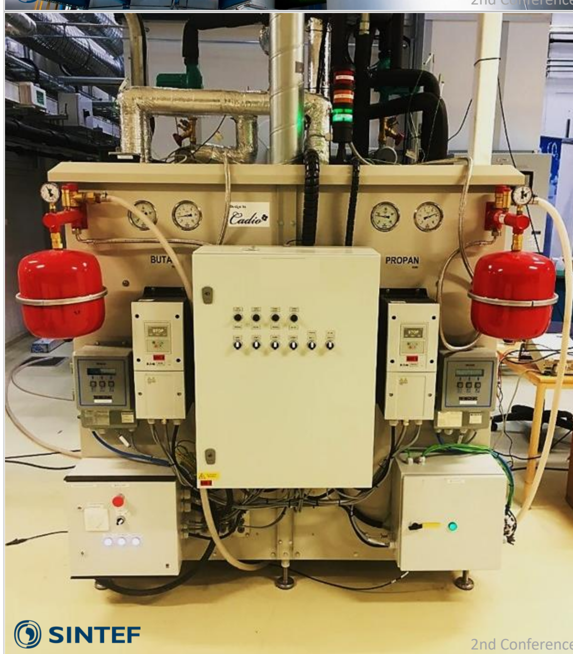

R&D challenges

- Develop a ***robust, stable*** *and* ***cost-effective heat pump*** *scale up* based on a propane-butane cascade
- **Temperature stability**, **respond** on **operational demands**
- **Optimal system implementation** with **thermal buffers/storages**
- **Response and interaction** of the installed back-up systems to ensure **safe process supply** and **continuity in production at any time.**
- **Safety in all steps**

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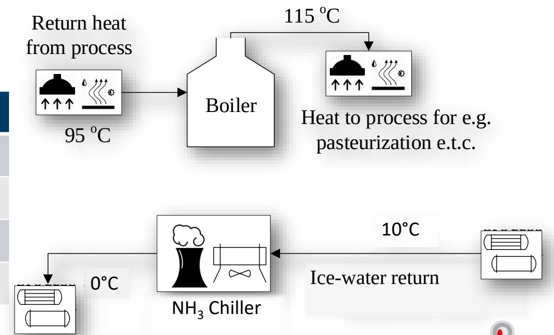
Implementation - Simultaneous utilization of heat-sink and heat-source

Example: diary

Parameter	Temperature °C
Heat-sink inlet	85°C - 95°C
Heat-sink outlet	105°C - 115°C
Ice-water return	10°C
Ice-water supply	0°C



Temperature lift: Heat sink outlet – Ice-water supply > **115 K**

Condensation temperature > 115°C → Application of High Temperature Heat Pump HTHP (>100°C)

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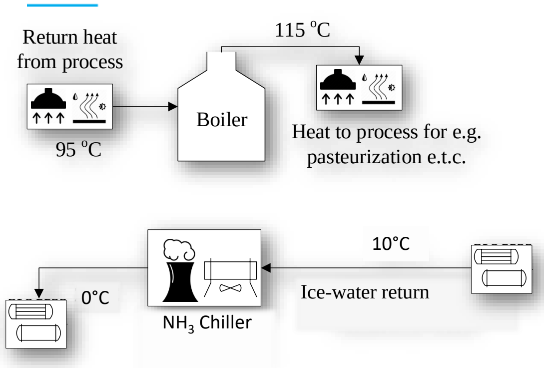
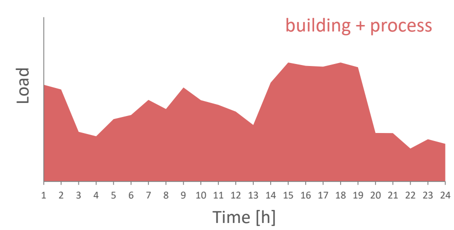
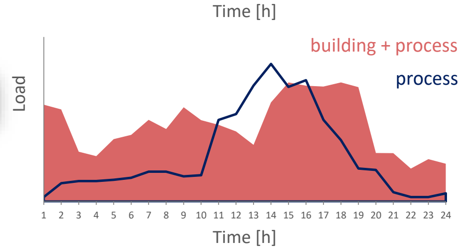
Implementation - Simultaneous utilization of heat-sink and heat-source







Heat source and heat sink are simultaneous!

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Implementation – Concept HTHP-pilot integration

Concept A:

Process heat, 115°C$_{water}$

Heat Pump
© HeatUp, HighEFF (Natural refrigerant)
$COP_{Heat+Cool}$

Process cooling, 0-4°C$_{water}$

Concept B:

Process heat, 115°C$_{water}$

Heat Pump
© HeatUp, HighEFF (Natural refrigerant)
COP_{Heat}

Excess heat, e.g. from chillers

Application area:

- Heat sink and source with moderate glide (<30K)
- HTHP with high temperature lift

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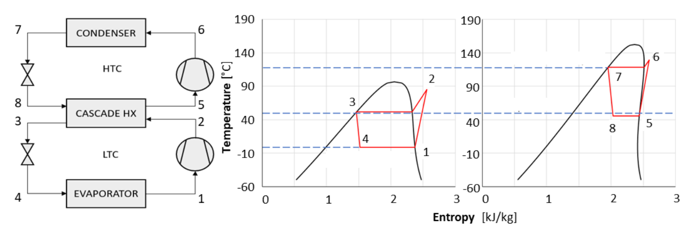
SINTEF

Application of natural refrigerants propane und butane for high temperature lift HTHP

- Why hydrocarbons?
  - Natural working fluid
  - Non poisoning (class A)
  - ODP = 0
  - Very low GWP
  - Good thermodynamic properties (20bar bei T_c=115°C)



- Challenge is the flammability (class 3)
- Requirements for safe applications are well described in standards e.g. DIN EN 378 – 1:2008 und ISO 5149 – 1:2014
- Standard components available, except butane compressor for high temperature application
- **Proper education of service personal and engineers is essential for success and** is required (EN-13313 Refrigerating systems and heat pumps - Competence of personnel)

11

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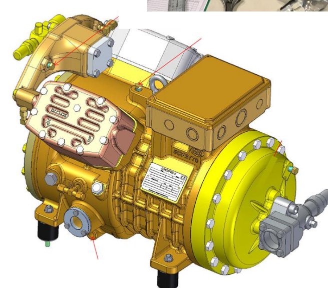
Key component - High temperature compressor

- Challenge is the heat management of the compressor:
  - Suction temperature <80°C
  - Discharge temperature <140°C
- Necessary modification of semi-hermetic compressor:
  - External discharge manifold
  - Extended measurement rang of thermal protection to 160°C
  - Effective electric motor
  - High temperature lubricant
  - Oil sump heater



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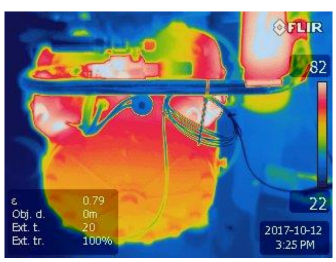
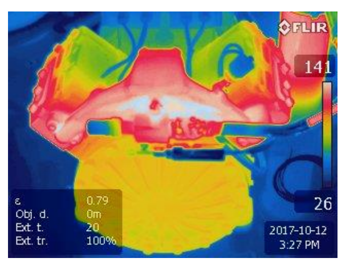
Key component - Thermal management

Propane compressor



Butane compressor prototype



- Temperature level HTC-Compressor is much higher than LTC-Compressor
- External discharge manifold reduces heat transfer to compressor body of HTC-compressor

HeatUp HighEFF

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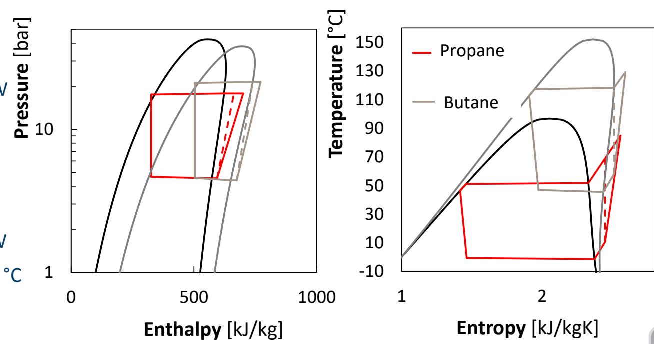
System performance - Lab scale

- Ice-water production
  - T_0= -1 °C – +1 °C
  - Evaporator capacity 8.37 kW – 11.3 kW
  - Water side T_{Hso_in}=10°C → T_{Hso_out}=4°C
- Hot-water production
  - T_C= 113 °C – +115 °C
  - Condenser capacity 19.0 kW – 27.3 kW
  - Water side T_{Hsi_in}=85 °C → T_{Hsi_out}=118 °C
- Combined Carnot-effciency
  - η_C=0,45



- R290/R600 cascade enables 119 K temperature lift at moderate working pressures 21 bar

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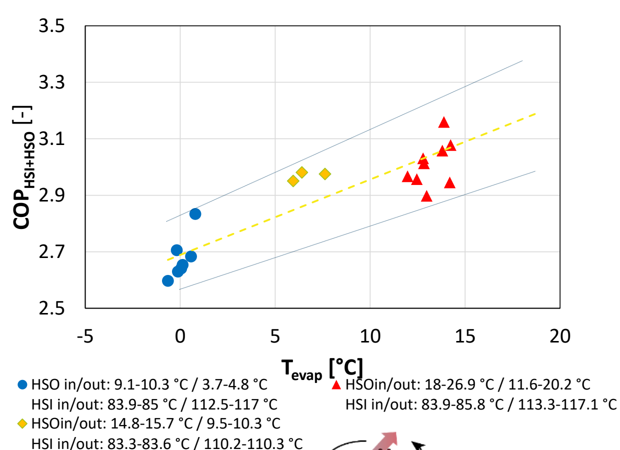
System performance - Lab scale

- Combined COP shows expected development:
  - COP ~ T_0
  - COP ~ 1/ T_C
  - COP ~ 1 / (T_C - T_0)



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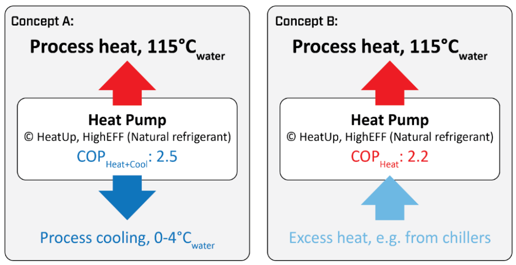
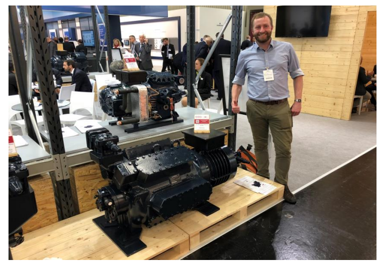
Saving potential of HTHP-pilot





	Reference case	Concept A	Concept B
hot-water production [kWh]	1	1	1
ice-water production [kWh]	0.41	0.41	0.41
power required [kWh]	0.09	0.54	0.64
gas boiler fuel [kWh]	1.18	-	-
total energy required [kWh]	1.27	0.54	0.64
total energy saving [kWh]	-	0.72	0.64
relative energy saving	-	57 %	50 %
CO_2-reduktion	-	94 %	91 %

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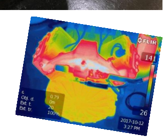
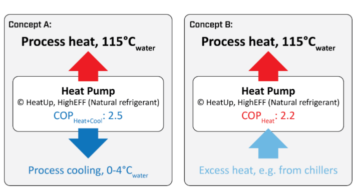
Summary



- Diary is a "perfect" match for HTHP for simultaneous ice and process hot water production (110 °C)
- Cascade system is efficient (combined COP = 2.6) for high temperature lift (119K) and T_C = 118°C
- Standard "on the shelve" components can be used beside R600 compressor
- Implementation concepts



- Saving potential for HTHP application:
  - Primary energy reduction of 57% and
  - CO_2–emission reduction of up to 94%
- Future work:
  - Development integration and test of 300kW pilot
  - Working range extension $T_0 \downarrow$; $T_c \uparrow$

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Cadio
DORIN INNOVATION
SINTEF
HighEFF
Heat Up
Thank you for your attention ☺
The authors would also like to acknowledge the support of The Research Council of Norway and the industrial partners through the grant NFR-296374 (SkaleUp), NFR-243679 (HeatUp) and NFR- 257632/E20 FME-HighEFF (Centre for Environment-friendly Energy Research).
SkaleUP
Sustainable and efficient heat pump development for combined process heat and cool
18
2nd Conference on High Temperature Heat Pumps (HTHP), Copenhagen, Denmark, 09.09.2019


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Two-phase vane compressor for supply of industrial process steam

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Keywords:
High temperature heat pump, R718, compressor technology, two-phase compression, evaporative compression

Abstract

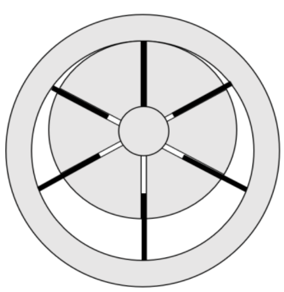
Recompression of waste steam, to supply it as heat at useable pressure and temperature is desirable in several industrial applications. However, the waste steam often contains impurities, such as liquid, air, and particles carried over from the industrial process (e.g. drying processes, frying processes). Direct compression of such steam requires a robust compressor able to handle two-phase flow, and at least where the steam is in direct contact with food or beverage products, an oil-free compressor.

Conventional rotary vane compressors are known to be tolerant to both liquids and particles in the working fluid, but has limitations when it comes to size, pressure and oil-free operation.

Tocircle Industries is developing an oil-free two-phase rotary vane compressor with high volumetric and pressure capacity. As for conventional rotary vane machines, the Tocircle compressor is a rotary positive displacement compressor in which the drive shaft directly drives a rotor eccentrically supported within a static casing. A number of vanes can slide radially in and out of the rotor, and a compression chamber is formed between the casing, the rotor and two vanes. The chamber volume then changes as the rotor rotates.

In a conventional vane compressor, radial guidance of the vanes is provided only by their contact with the casing. The resulting friction between vane tip and casing sets a limit to the size and capacity obtainable without oil lubricant in the process chamber.

The Tocircle compressor differs from a conventional rotary vane compressor in that the radial guidance of the vanes is handled in the machine centre, at a diameter small enough to use conventional bearing technology, hence avoiding contact between vane tip and casing completely, and making it possible to build larger machines without the need for oil lubrication.



Conventional rotary vane machine

Tocircle machine, Vading principle

Fig 1. Tocircles geometric principle (Vading) vs conventional rotary vane machines

Testing of the machines have taken place in Tocircles test facilities in Glomfjord, Norway, first in an open-loop compressor test rig directly compressing waste steam in 1 stage from app 3.5 to 12 bara, and then in a full closed-loop steam heat pump, compressing in 2 stages from app. 1 to 12 bara,

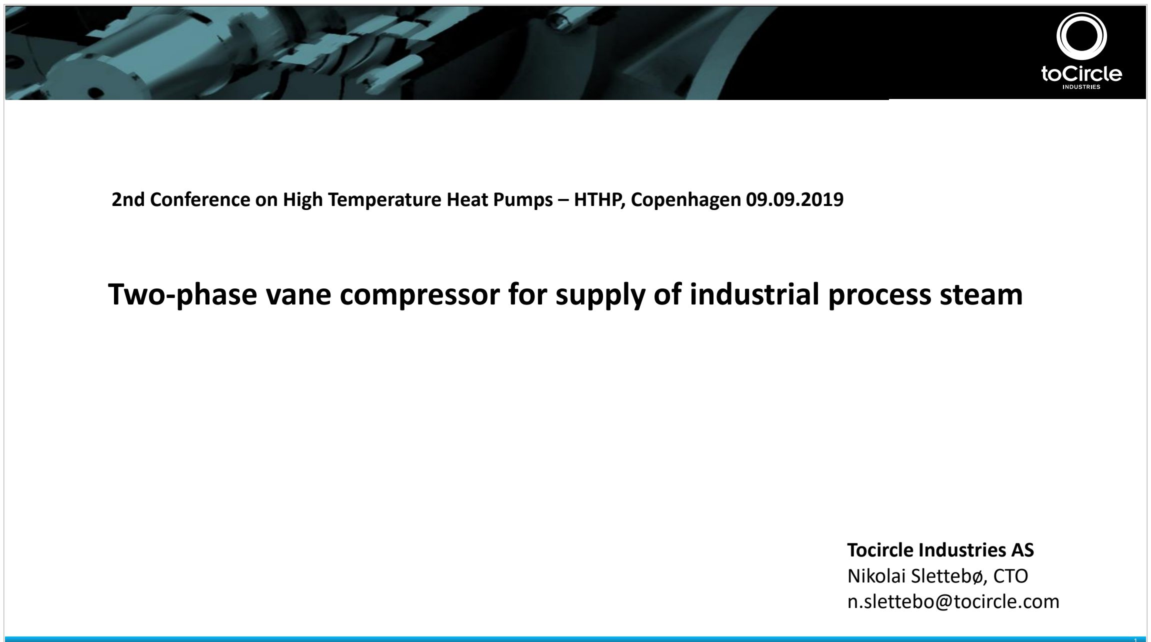
The testing confirmed that the compressors are tolerant to liquid in the working fluid, and benefit from a certain liquid content as it aids in sealing gaps between moving parts.

The ability to compress a mixture of gas and liquid makes it possible to realize so-called evaporative compression in the machines; By injecting liquefied working fluid directly into the working chamber, evaporative cooling of the compressed gas is obtained. The injected liquid will fully or partly evaporate during the compression, while the compressed gas is always kept at the saturation line. Compared to a standard heat pump with dry compression, where the working fluid is superheated through compression, evaporative compression gives a better system performance, and ensures that the working fluid temperature never exceed the condensation temperature.

The tests performed in Glomfjord further confirmed that by means of liquid injection, the compressors can deliver saturated or wet steam, with condenser temperatures exceeding 188 degC.

To fulfil requirements of oil-free operation, an important part of the compressor r&d work has been the development of seal and bearing solutions that are lubricated only by liquified process media. This has resulted in two different solutions for hydrostatic linear bearings (“Hydroslide 2” and “Hydroslide 3”) and a hydrostatically supported axial process seal. “Hydroslide 2” has been developed through 3 prototype generations and is currently being tested in the pilot heat pump in Glomfjord. The machines that have been built and tested up till now are based on the so-called “Vading principle”, after the inventor of the basic geometric principle, Kjell Vading.

Through the cooperation with SINTEF in the Free2Heat project, Tocircle are developing a new compressor concept based on a different geometry than the Vading principle. A key element in this work is the development of the “Hydroslide 3” solution. The new concept has been named “Rigid Vane” and is of interest because, compared to the Vading principle, the new geometry reduces both leakages and loads/friction, hence improving both the isentropic efficiency and reliability in the machines.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Two-phase vane compressor for supply of industrial process steam
Tocircle Industries AS
Nikolai Slettebø, CTO
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Introduction
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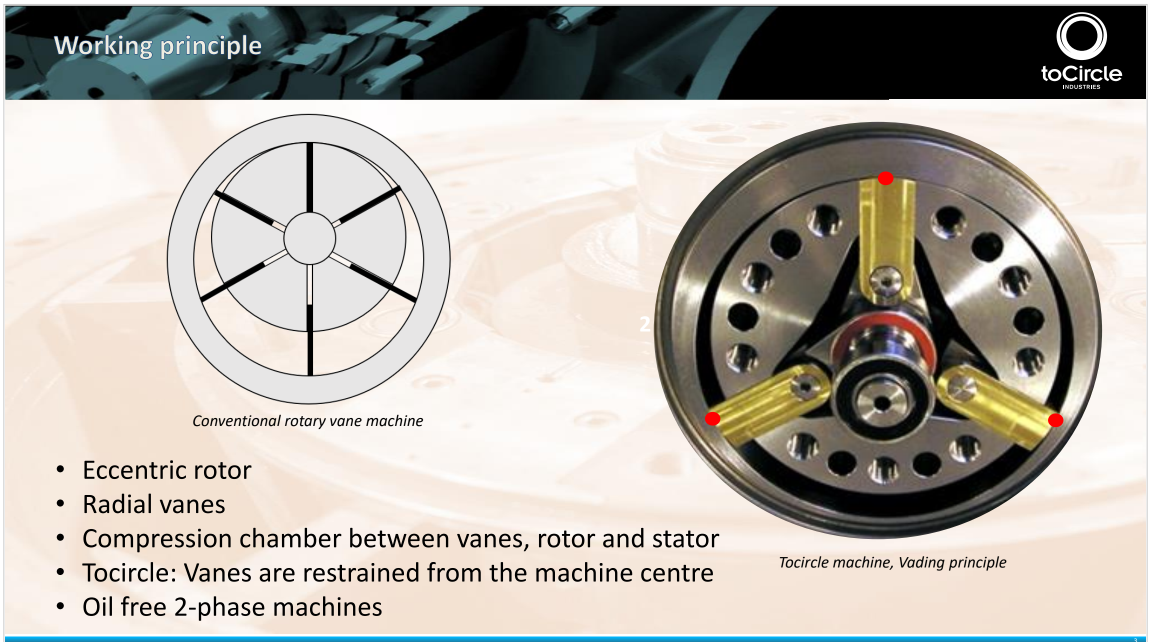
This presentation:
• Introduction
• Working principle
• Two-phase and evaporative compression
• Testing
• Compressor development
• Free2Heat


Working principle
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Conventional rotary vane machine
Tocircle machine, Vading principle
• Eccentric rotor
• Radial vanes
• Compression chamber between vanes, rotor and stator
• Tocircle: Vanes are restrained from the machine centre
• Oil free 2-phase machines
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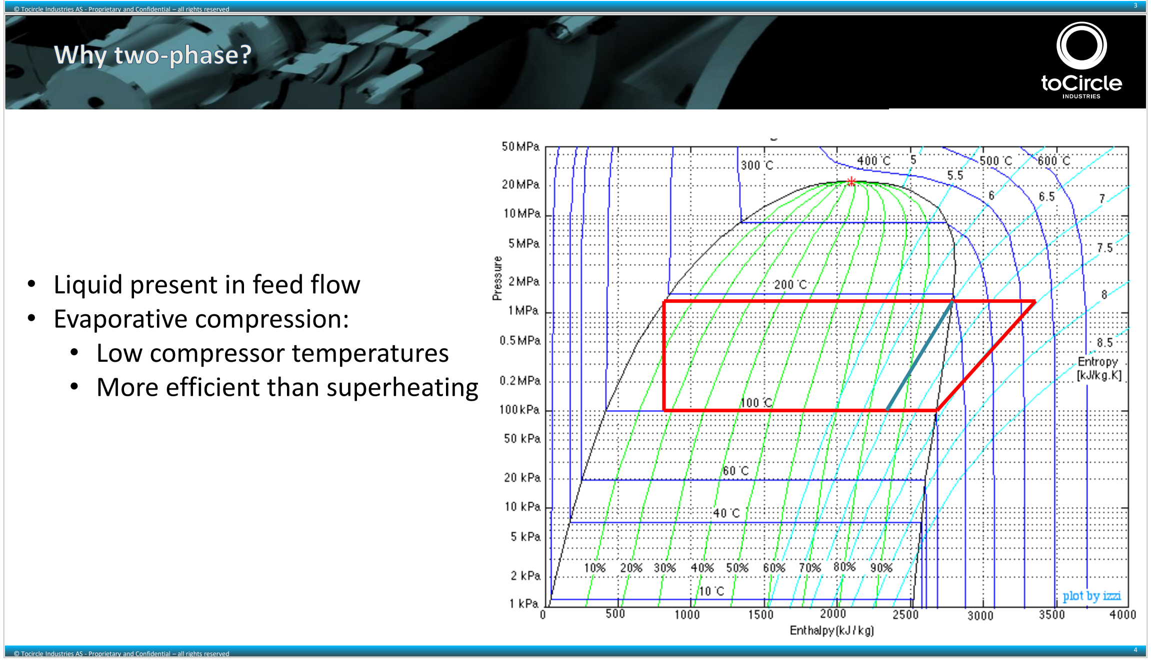
Why two-phase?
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• Liquid present in feed flow
• Evaporative compression:
• Low compressor temperatures
• More efficient than superheating
Pressure
Enthalpy (kJ / kg)
Entropy [kJ/kg.K]
50 MPa
20 MPa
10 MPa
5 MPa
2 MPa
1 MPa
0.5 MPa
0.2 MPa
100 kPa
50 kPa
20 kPa
10 kPa
5 kPa
2 kPa
1 kPa
0
500
1000
1500
2000
2500
3000
3500
4000
300 °C
400 °C
500 °C
600 °C
200 °C
100 °C
60 °C
40 °C
10 °C
10%
20%
30%
40%
50%
60%
70%
80%
90%
plot by izzi
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Compressor testing
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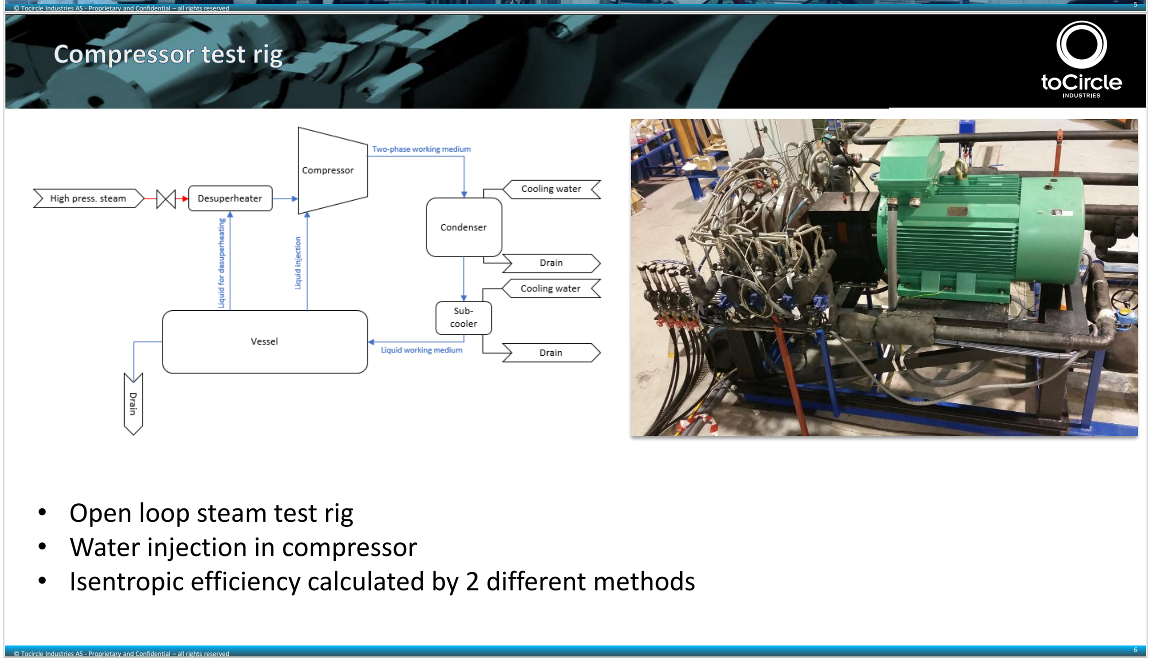
Compressor test rig
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High press. steam
Desuperheater
Compressor
Two-phase working medium
Cooling water
Condenser
Drain
Cooling water
Sub-cooler
Drain
Liquid for desuperheating
Liquid injection
Vessel
Liquid working medium
Drain
• Open loop steam test rig
• Water injection in compressor
• Isentropic efficiency calculated by 2 different methods
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Compressor test rig
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Compressor pressures and temperatures
Inlet desuperheated temp
Outlet steam temperature compressor
Inlet pressure
Outlet pressure
Temperature [°C]
Pressure [barg]
300
250
200
150
100
50
0
14
12
10
8
6
4
2
0
10:55:12
10:58:05
11:00:58
11:03:50
11:06:43
11:09:36
11:12:29
11:15:22
11:18:14
11:21:07
11:24:00
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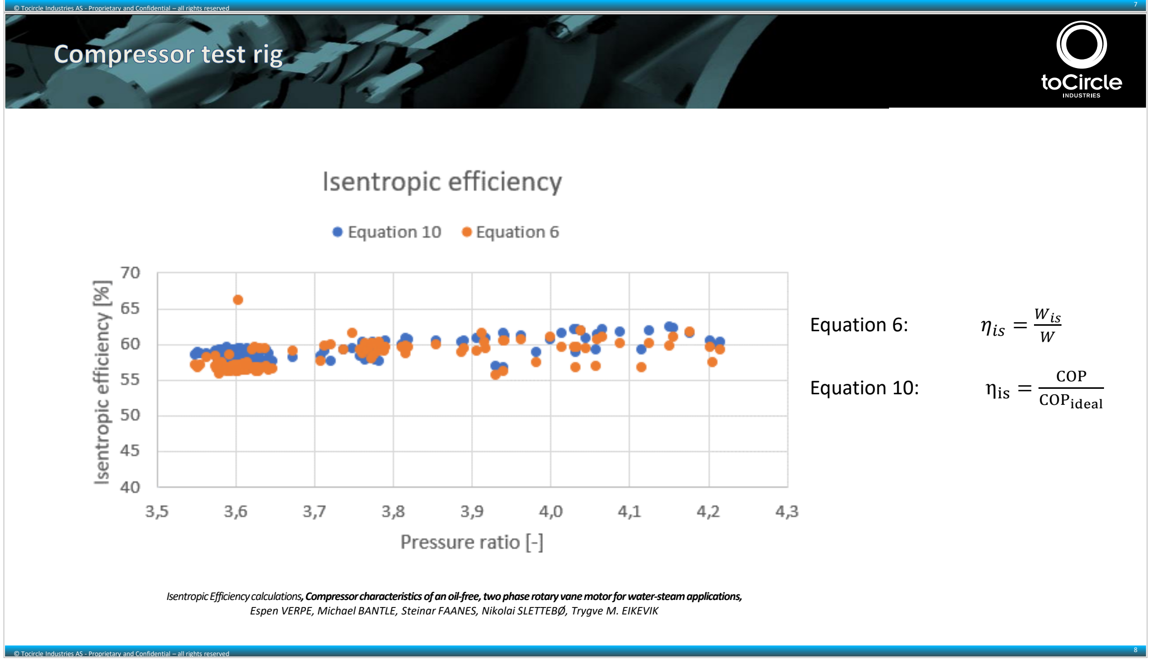
Compressor test rig
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Isentropic efficiency
Equation 10
Equation 6
Isentropic efficiency [%]
70
65
60
55
50
45
40
3,5
3,6
3,7
3,8
3,9
4,0
4,1
4,2
4,3
Pressure ratio [-]
Equation 6: $\eta_{is} = \frac{W_{is}}{W}$
Equation 10: $\eta_{is} = \frac{COP}{COP_{ideal}}$
Isentropic Efficiency calculations, Compressor characteristics of an oil-free, two phase rotary vane motor for water-steam applications,
Espen VERPE, Michael BANTLE, Steinar FAANES, Nikolai SLETTEBØ, Trygve M. EIKEVIK
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Full heat pump tests

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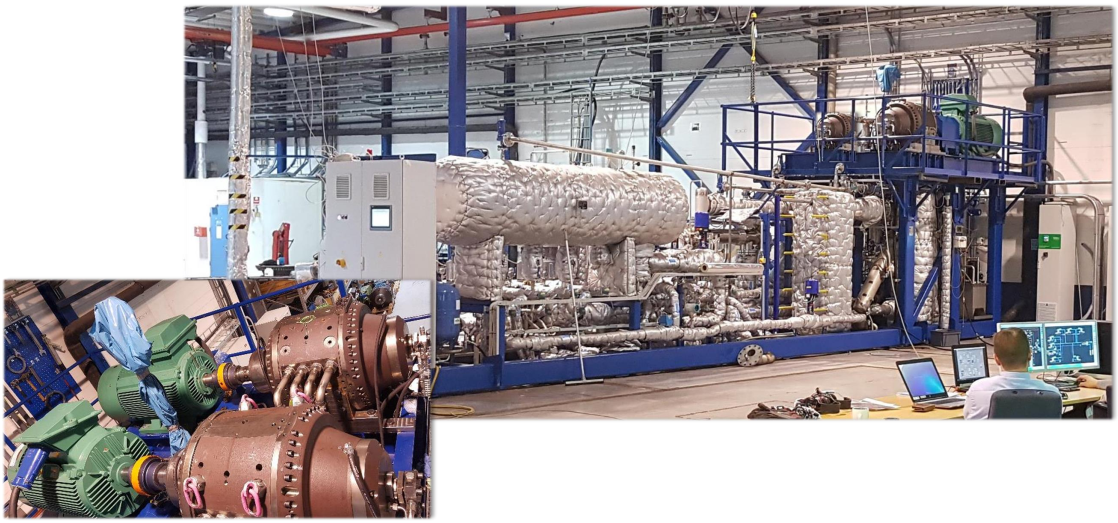

Compressor development – hydrostatic bearings

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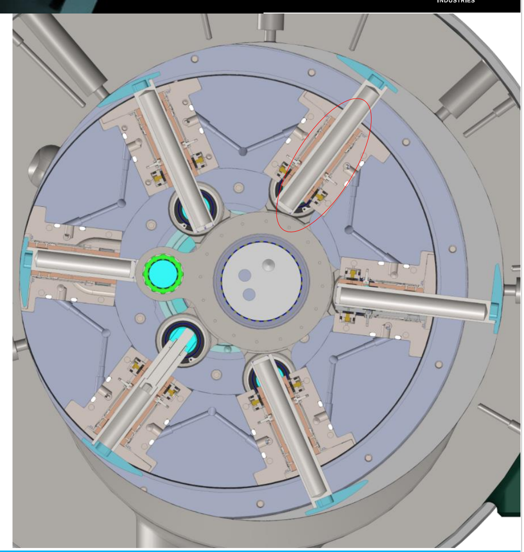
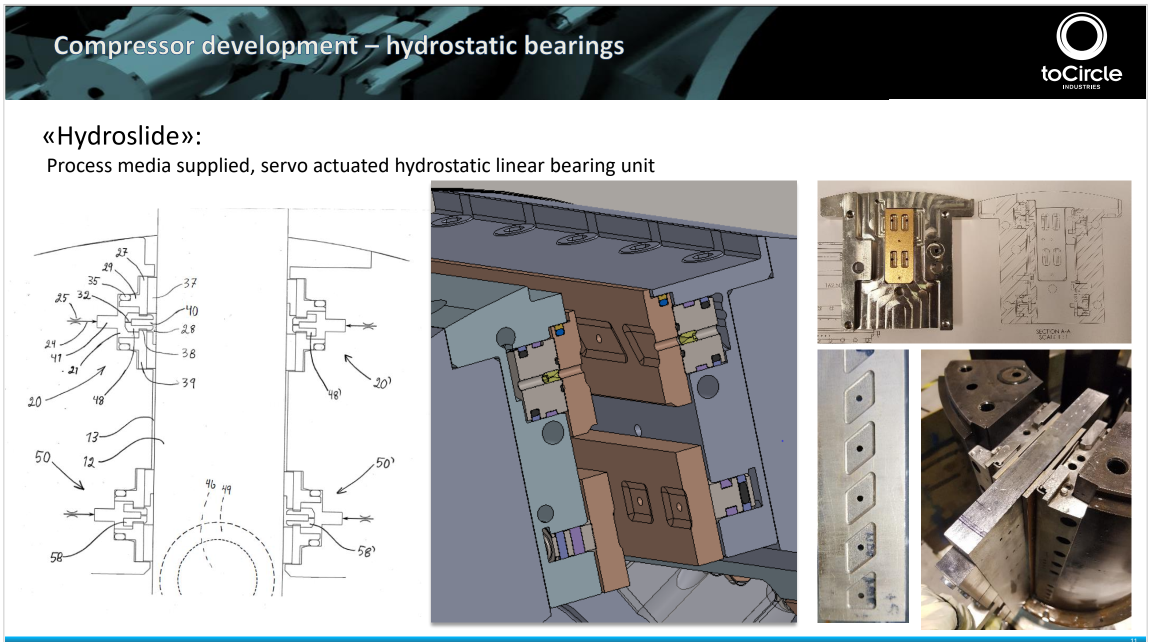
Design challenges for linear bearings:

- Pressure and diff pressure varies over a rotation
- Edge loading
- Not possible to isolate from process => no oil
- => Using liquefied process media




Compressor development – hydrostatic bearings
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«Hydroslide»:
Process media supplied, servo actuated hydrostatic linear bearing unit
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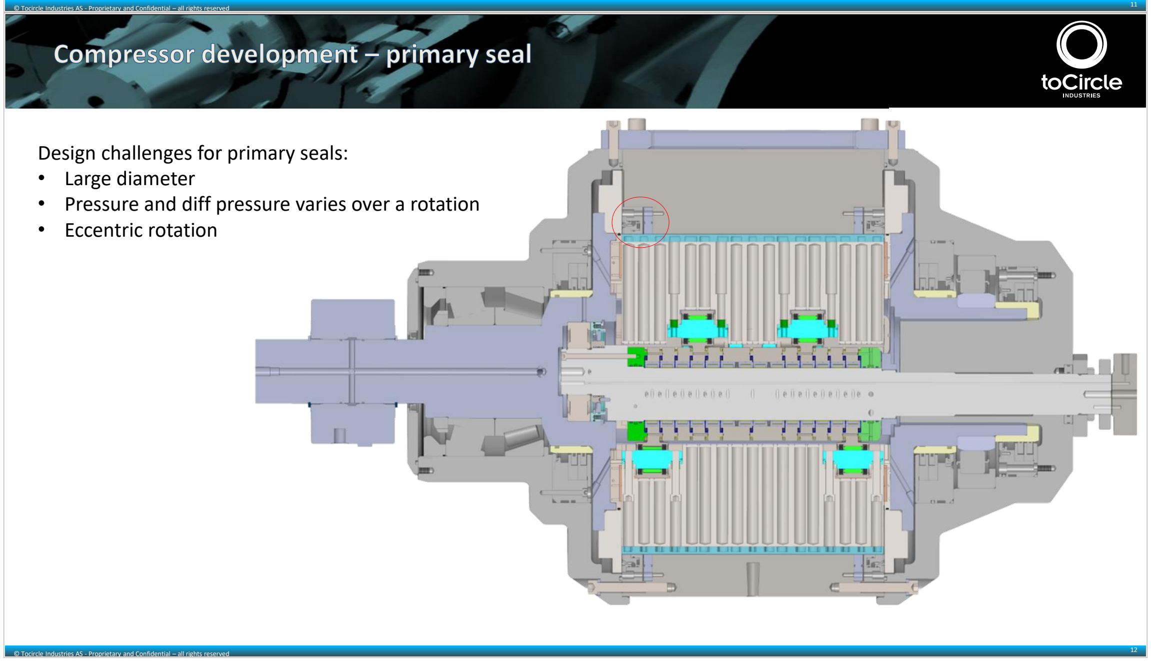
Compressor development – primary seal
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Design challenges for primary seals:
• Large diameter
• Pressure and diff pressure varies over a rotation
• Eccentric rotation
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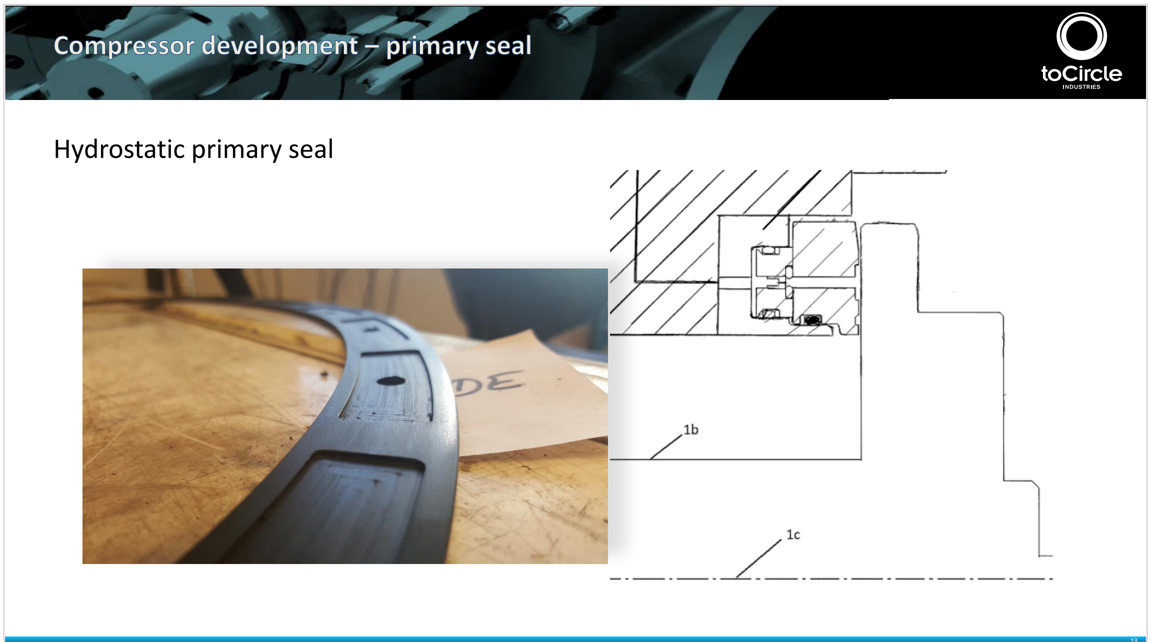
Compressor development – primary seal
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Hydrostatic primary seal
DE
1b
1c
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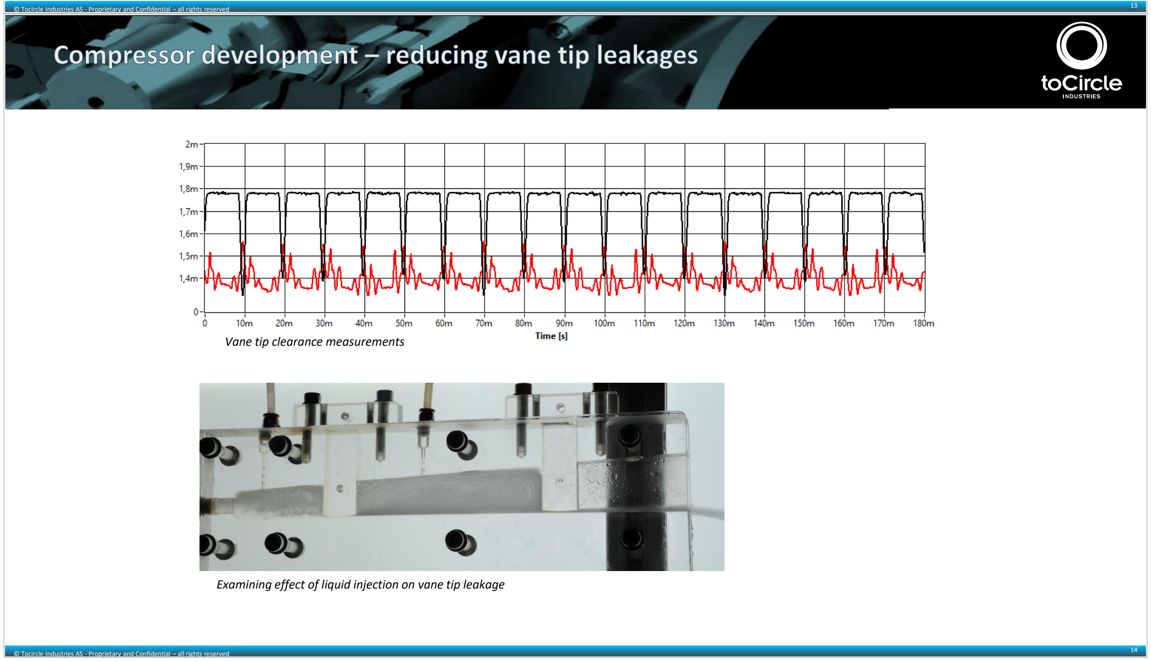
Compressor development – reducing vane tip leakages
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2m
1,9m
1,8m
1,7m
1,6m
1,5m
1,4m
0
0 10m 20m 30m 40m 50m 60m 70m 80m 90m 100m 110m 120m 130m 140m 150m 160m 170m 180m
Time [s]
Vane tip clearance measurements
Examining effect of liquid injection on vane tip leakage
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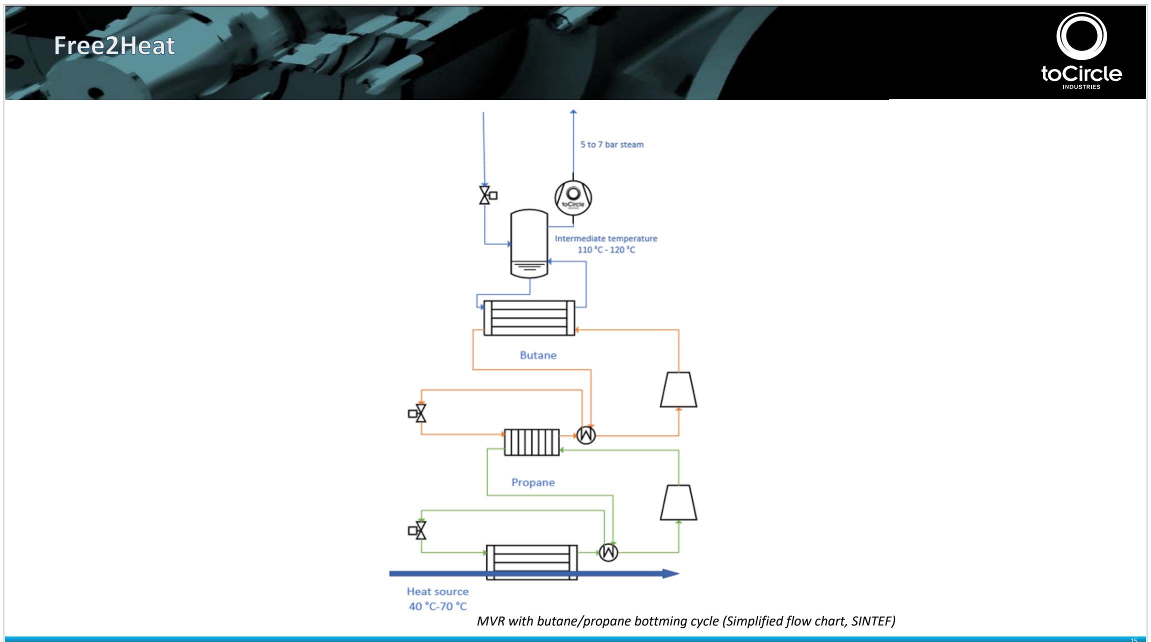
Free2Heat
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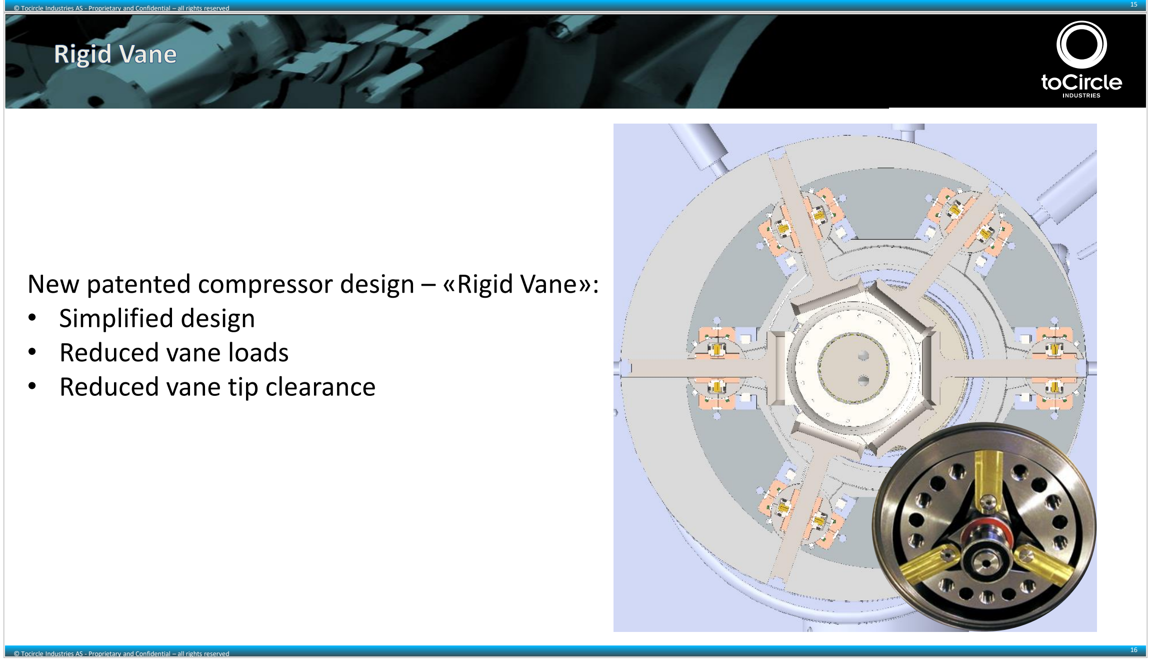
5 to 7 bar steam
Intermediate temperature
110 °C - 120 °C
Butane
Propane
Heat source
40 °C-70 °C
MVR with butane/propane bottming cycle (Simplified flow chart, SINTEF)


Rigid Vane
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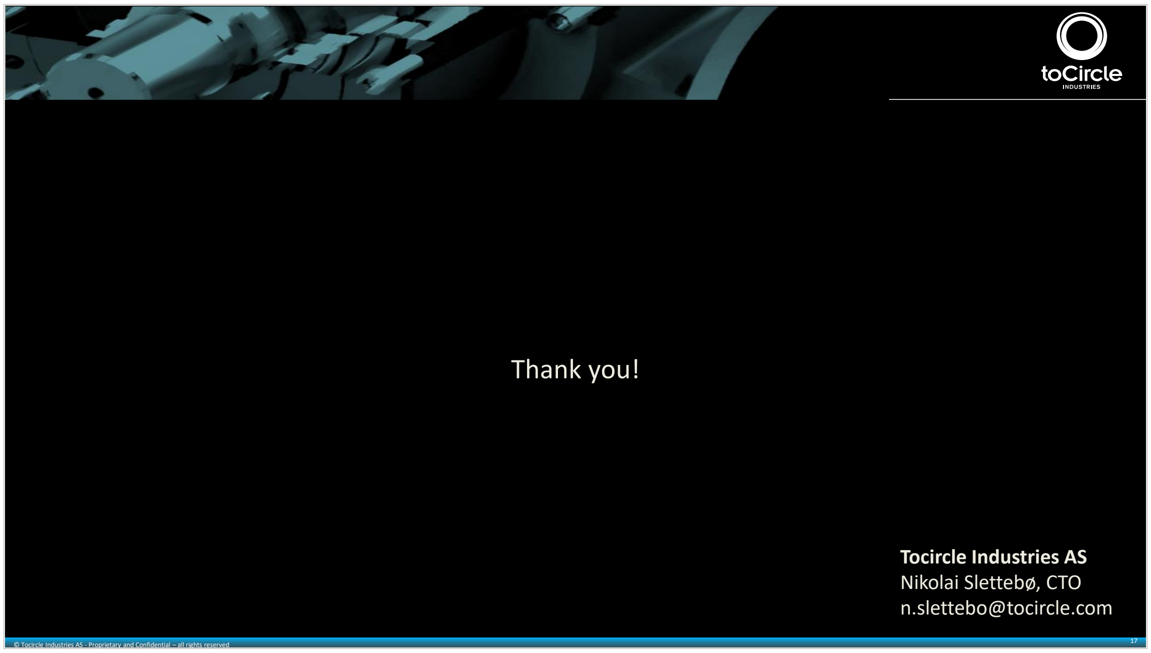
New patented compressor design – «Rigid Vane»:
• Simplified design
• Reduced vane loads
• Reduced vane tip clearance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Thank you!
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3 Current developments and trends for high-temperature heat pumps

1.1 High-temperature CO2 heat pump integration into the spray drying process, Lorenzo Bellemo, GEA

1.2 Transcritical heat pump solution for industrial dryers, Florence De Carlan, EDF

1.3 Experimental results of HFO/HFCO refrigerants in a laboratory scale HTHP with up to 150 °C supply temperature, Cordin Arpagaus, NTB Buchs

1.4 Supply of high-temperature heat and cooling with MAN ETES HP, Raymond Decorvet and Emmanuel Jacquemoud

High temperature CO_2 heat pump integration into the spray drying process

Lorenzo Bellemo[1], *Jan Gerritsen*[2], *Kenneth Hoffmann*[3]

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[3] GEA, Refrigeration Product Technology Center, EE`s-Hertogenbosch, Netherlands,
Kenneth.Hoffmann@gea.com

Keywords:
High temperature heat pump, R744, compressor technology, process integration

Introduction

Spray drying is a highly energy intensive process. Dairy spray dryers consume around 1.2 kWh for air heating per kg powder produced at temperatures above 200°C. High temperatures are conventionally generated by combusting fossil fuels, mostly natural gas. Significant reductions in gas consumption could be achieved with high temperature heat pumps. Heat pumps could also provide energy for air dehumidification in the form of chilled water and/or hot water for regenerating desiccant dehumidifiers. Dehumidification of process air maximises powder production rate, especially in humid climates. Spray dryers exhaust air containing large amounts of heat, which can be utilized by heat pumps to provide stable operating conditions independently of climatic conditions [1]. Therefore, heat pumps can provide useful heating and cooling to the spray drying process, if able to generate high (>100°C) and low (<5°C) water temperatures simultaneously, and recover exhaust heat. Trans critical CO_2 heat pumps are the only suitable one stage cycle with natural refrigerant able to generate hot water temperatures up to 135°C and cold water temperatures below 5°C with existing compression technology, as CO_2 piston compressors can rise gas pressure from a minimum of 35 bar to a maximum of 130 bar.

Methods

The trans critical CO_2 cycle has been modelled in EES for estimating the system steady state performance. Semi-hermetic GEA Bock piston compressors for heat pump applications are considered and modelled by polynomials from the manufacturer, linking CO_2 suction and discharge conditions. CO_2 gas cooling is performed in two serial stages, each stage using different water flow rates for realizing hot water temperatures by optimal CO_2-water temperature profile coupling. After CO_2 evaporation, an additional superheating step is included with heat provided from an external heat source, for obtaining the high compressor discharge temperatures. The model is used for dimensioning the heat pump components to satisfy the heating and cooling loads required by spray dryers. Loads are divided into three temperature levels: high temperature (water around 135°C from the first gas coolers), medium temperature (water around 75°C from the second gas cooler) and low temperature (water around 4°C from the evaporator). All heat pump components are commercially available, but the control system for optimal heating and cooling regulation at these three temperature levels via water circuits is not.

GEA has built a trans critical CO_2 heat pump prototype for developing such control system, as well as for demonstrating performance at high temperature operation. Two semi-hermetic 4-cylinder GEA Bock HGX 34/110-4 SH compressors have been selected, which can cover a large operating range to provide simultaneous heating and cooling. Both compressors can be frequency controlled in the range 25-70 Hz. The multi-compressor configuration has been preferred to a single compressor for facilitating start up and small load scenarios. The prototype maximum heating capacity is around 90 kW. The gas coolers are both copper-brazed plate heat exchangers, mounted in series. Danfoss controllers are used to regulate the compressors and the evaporator expansion valve. The heat pump has been installed at the GEA Test Center in Søborg (DK) and connected to an MSD™ spray dryer via three separated water circuits (high, medium and low temperature). The spray dryer requires around 1800 m^3/h process airflow for the main drying chamber, as well as airflows for other drying stages (static-fluid-bed and vibro-fluidizer). Valves and pumps in the three water circuits are controlled by a dedicated PLC, also communicating with Danfoss controllers in the prototype. CO_2, water and air temperatures, pressures and flow rates are constantly measured and logged at several points in the installation to allow a complete system analysis.

Results

Prototype tests are still at an initial stage, as the total running time is around 100 hours, and control optimization is an ongoing development. However, stable operation of the heat pump has already been achieved under different spray dryer load conditions. The compressor maximum frequency has been tested in the range 50-65 Hz with satisfactory results. The heat pump can quickly start up from cold system conditions (heat pump and spray dryer starting up together) or hot system conditions (heat pump starting up during ongoing spray dryer operation).

The longest period of stable operation tested so far has lasted 4 hours with simultaneous generation of 550 kg/h high temperature water around 130°C (approx. 30 kW), 1100 kg/h medium temperature water around 75°C (approx. 47 kW) and 7400 kg/h low temperature water around 5°C. The CO_2 compressors, running with suction conditions 36 bar and 45°C and discharge conditions 105 bar and 145°C, have performed according to manufacturer specifications, within the tolerances specified in EN12900 [2]. Under these conditions, main drying air was heated up to 115°C.

Discussion

Preliminary results are very positive, especially the stable behaviour and the achieved simultaneous generation of high and low water temperatures. System improvements are ongoing, particularly regarding heat release from the water circuits to the air streams at the medium temperature level, where water return temperatures as high as 38°C have been recorded, causing high CO_2 temperatures after gas cooling. These improvements are installation specific and do not depend on the performance of heat pump components. Furthermore, more test runs are required to develop a fully automated control system of the water circuits. Calculations on industrial size trans critical CO_2 heat pumps indicate that full use of the high and medium temperature heating loads would result in heating COP as high as 3.5 with simultaneous production of cold water that, if useful to the process, would result into a combined COP as high as 5.5.

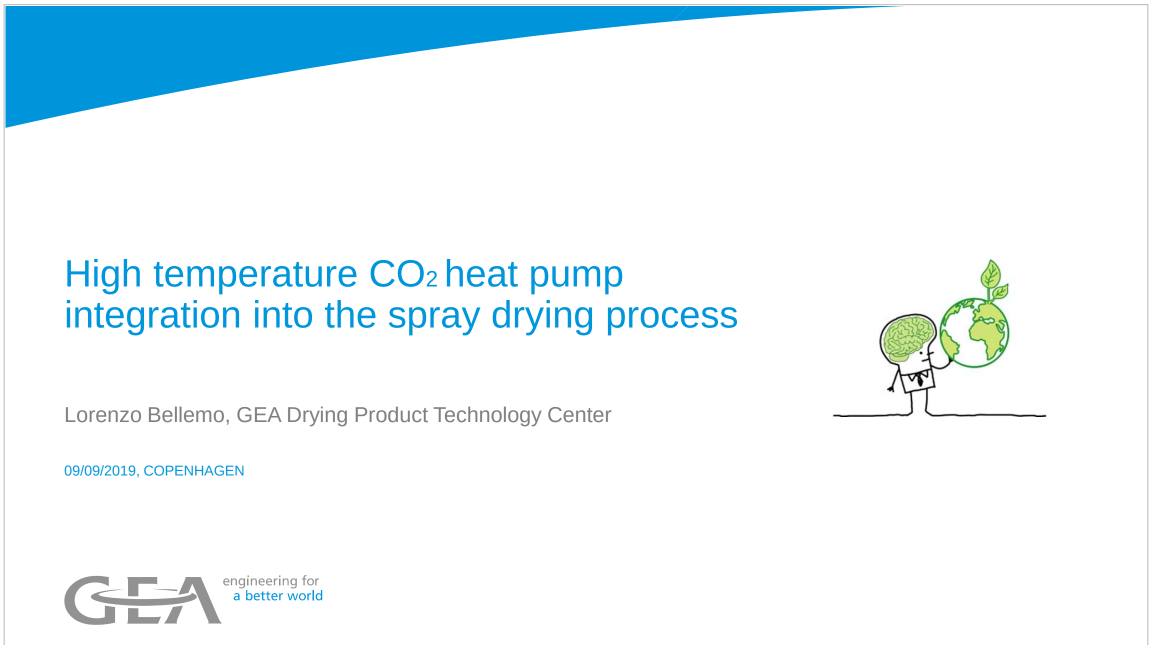
Conclusion

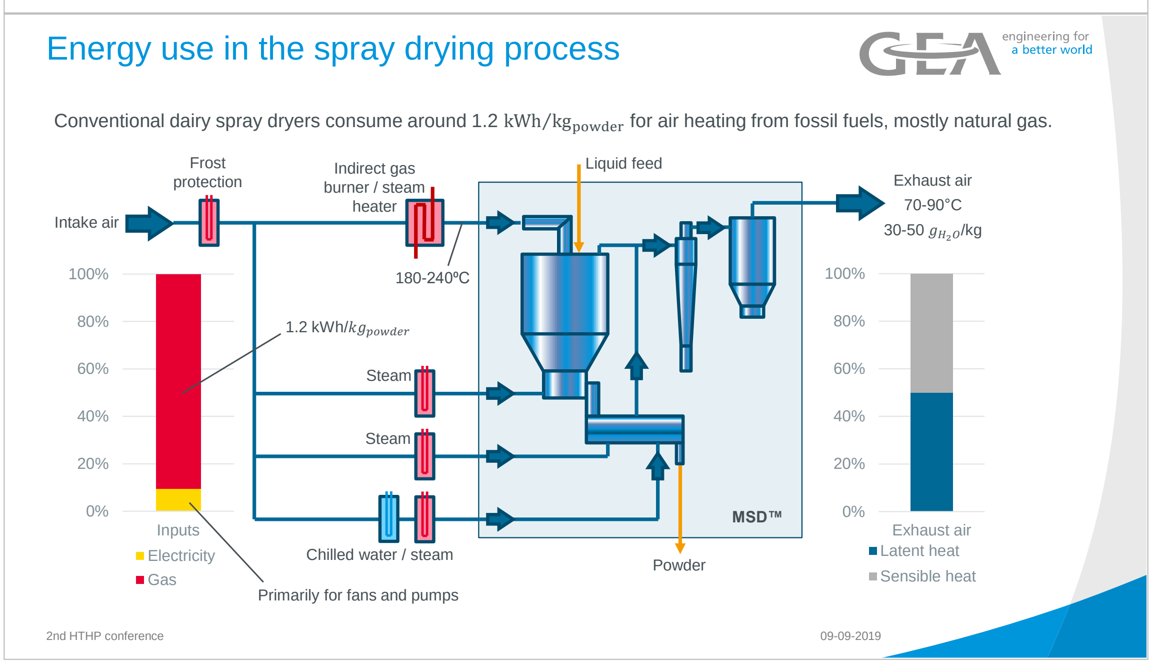
The experience acquired at the prototype installation is fundamental for integrating trans critical CO_2 heat pumps in spray drying plants. Successful installations of high temperature heat pumps depend as much on the heat pump component performance as on optimal process integration and on robust automated control systems for optimizing system performance. The high temperature trans critical CO_2 cycle is confirmed to be a promising solution to provide combined high temperature heating and cooling in spray drying plants.

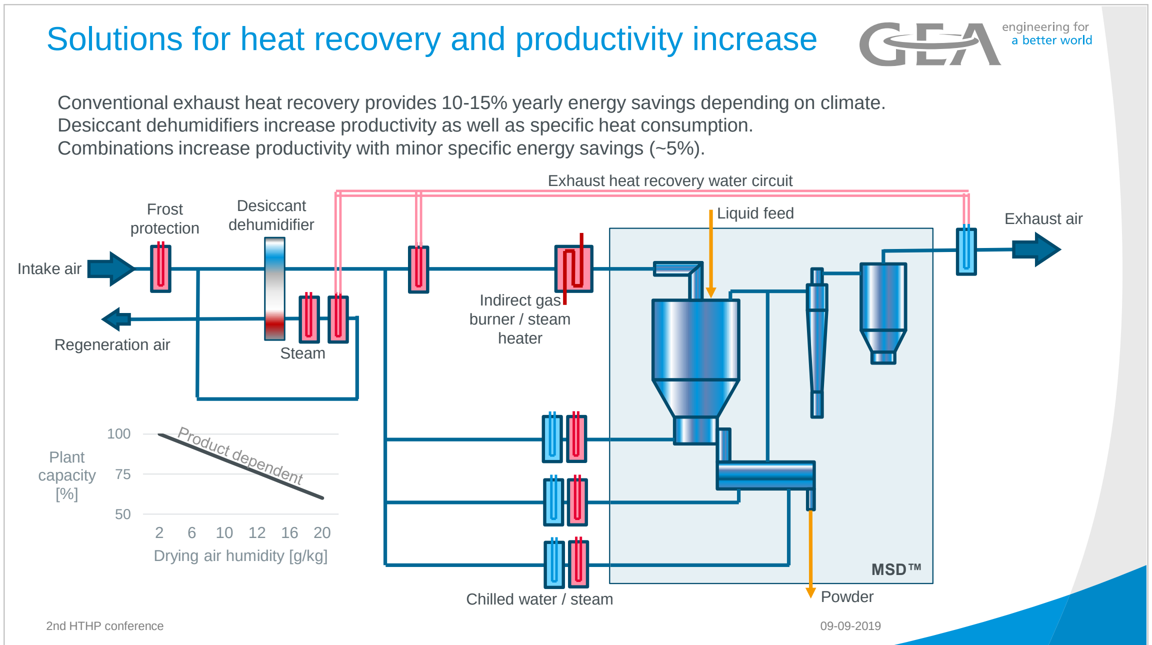
References

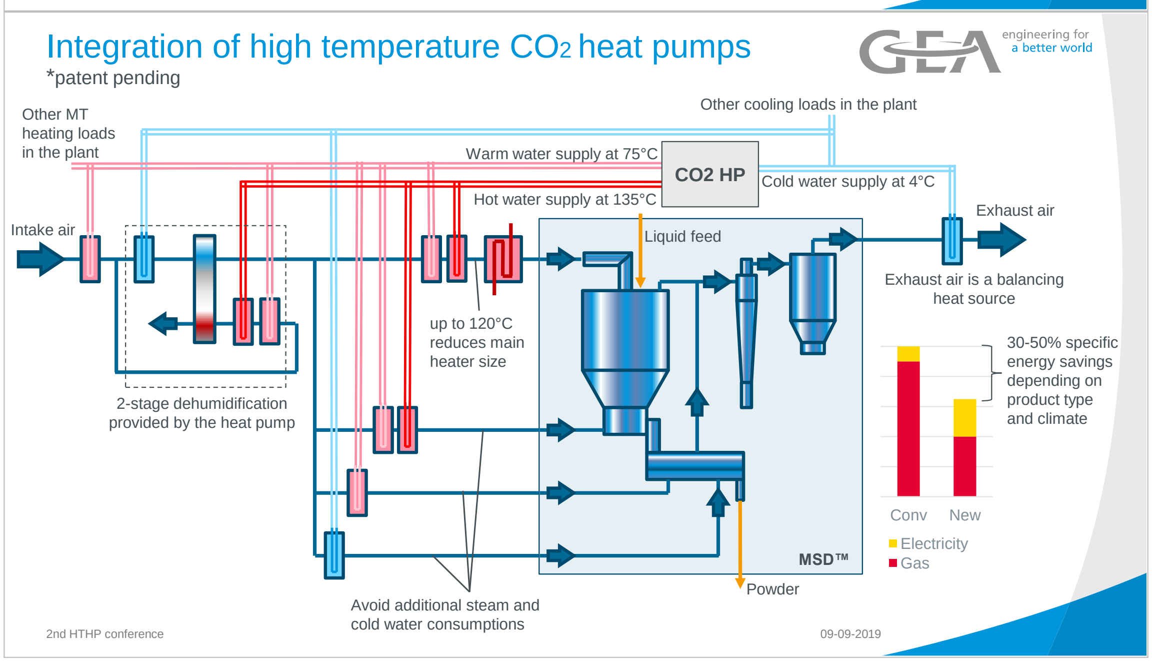
[1] Feuerriegel B., Bellemo L., “Drying system with improved energy efficiency and capacity control”, international patent application WO2018091049(A1).

[2] Standard EN 12900:2013. “Refrigerant compressors. Rating conditions, tolerances and presentation of manufacturer's performance data”.


High temperature CO2 heat pump integration into the spray drying process
Lorenzo Bellemo, GEA Drying Product Technology Center
09/09/2019, COPENHAGEN
engineering for a better world


Energy use in the spray drying process
engineering for a better world
Conventional dairy spray dryers consume around 1.2 kWh/kg$_{powder}$ for air heating from fossil fuels, mostly natural gas.
Frost protection
Intake air
Indirect gas burner / steam heater
180-240ºC
Liquid feed
Exhaust air
70-90°C
30-50 g_{H_2O}/kg
1.2 kWh/kg_{powder}
Steam
Steam
Chilled water / steam
MSD™
Powder
100%
80%
60%
40%
20%
0%
Inputs
Electricity
Gas
Primarily for fans and pumps
Exhaust air
Latent heat
Sensible heat
2nd HTHP conference
09-09-2019


Solutions for heat recovery and productivity increase
engineering for a better world
Conventional exhaust heat recovery provides 10-15% yearly energy savings depending on climate.
Desiccant dehumidifiers increase productivity as well as specific heat consumption.
Combinations increase productivity with minor specific energy savings (~5%).
Exhaust heat recovery water circuit
Frost protection
Desiccant dehumidifier
Liquid feed
Exhaust air
Intake air
Regeneration air
Steam
Indirect gas burner / steam heater
Plant capacity [%]
100
75
50
Product dependent
2 6 10 12 16 20
Drying air humidity [g/kg]
MSD™
Chilled water / steam
Powder
2nd HTHP conference
09-09-2019


Integration of high temperature CO2 heat pumps
engineering for a better world
*patent pending
Other MT heating loads in the plant
Other cooling loads in the plant
Warm water supply at 75°C
CO2 HP
Cold water supply at 4°C
Hot water supply at 135°C
Intake air
Liquid feed
Exhaust air
Exhaust air is a balancing heat source
up to 120°C reduces main heater size
30-50% specific energy savings depending on product type and climate
2-stage dehumidification provided by the heat pump
Conv
New
Electricity
Gas
MSD™
Powder
Avoid additional steam and cold water consumptions
2nd HTHP conference
09-09-2019

Overview of high temperature trans critical CO_2 cycle

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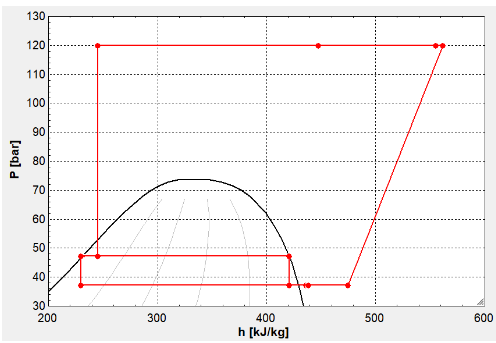
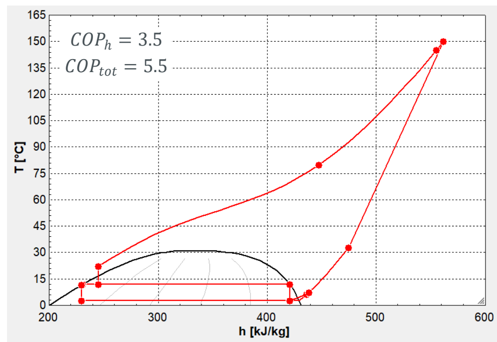
- One step compression with parallel semi-hermetic piston compressors
- Two gas coolers in series to provide high and medium temperature water streams
- DX evaporation and superheating

Cycle representation with best exploitation of high and medium temperature heating loads and low temperature cooling load.





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Trans critical CO_2 cooling

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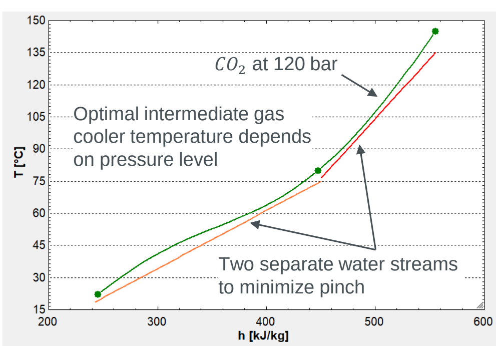
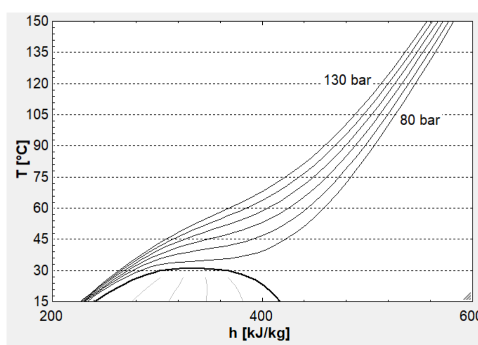
Trans critical CO_2 is cooled in two steps in series to maximize the generated hot water temperature.





CO_2 specific heat capacity is strongly dependent on temperature in the trans critical region. Its variation is lower as pressure increases, providing a more homogeneous heat release.

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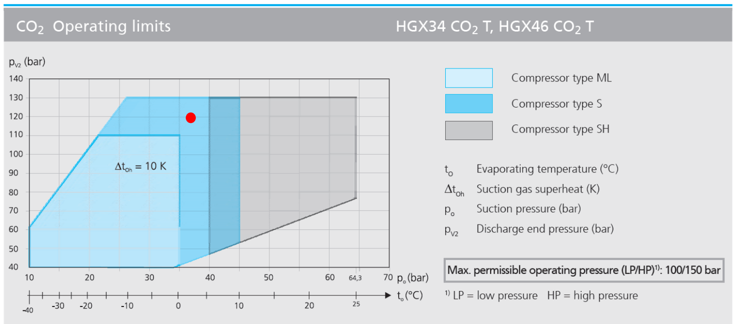
Compressor operating conditions

GEA engineering for a better world

CO_2 discharge conditions up to 130 bar and 150°C are possible within the GEA Bock piston compressor limits under continuous operation.

High CO_2 discharge temperatures are needed to generate hot water temperature. This requires a regulation of the suction temperature, based on suction and discharge pressure levels.

The compressor motor is cooled by suction gas, hence maximum allowed suction gas temperatures is around 40°C.



GEA Bock semi-hermetic piston compressor

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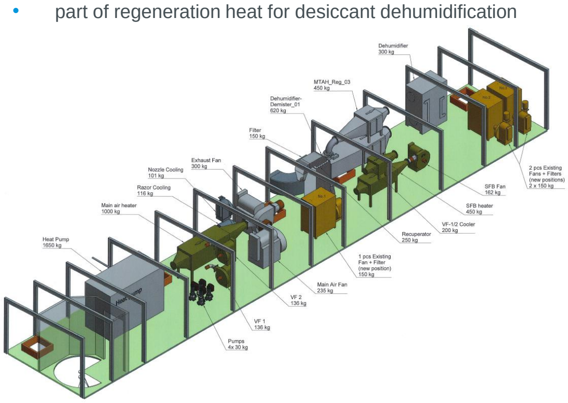
CO2 heat pump prototype at GEA Drying Test Center

GEA engineering for a better world

The heat pump prototype provides:

- total heating capacity up to 90 kW
- water temperatures up to 135°C → possibility for air heating up to 120 °C
- cold water down to 4 degC for intake air dehumidification and cooling
- part of regeneration heat for desiccant dehumidification



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CO_2 heat pump prototype at GEA Drying Test Center

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Front side **Back side**

3 pressure safety levels:

- Pressure switches
- Danfoss controller
- Blow-off pipe

2 GEA Bock HGX34/110-4 SH CO2 T compressors

CO_2 discharge up to 110 bar and 150°C from 36 bar with frequency regulation 25-70 Hz

2 smaller compressors preferred over 1 bigger for better capacity regulation

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Preliminary test results

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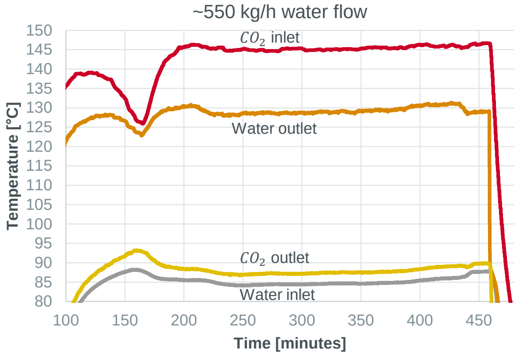
The prototype system has been operated for ~ 100 hours, including initial commissioning.

The heat pump is now able to generate high temperature water around 130°C, medium temperature water around 75°C, and cold water around 5°C with stable operation.

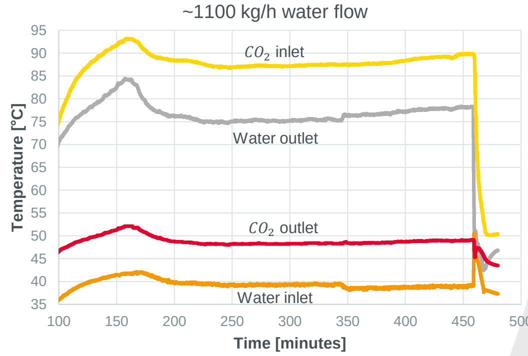
High temperature gas cooler operation

~550 kg/h water flow



Medium temperature gas cooler operation

~1100 kg/h water flow



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Preliminary test results

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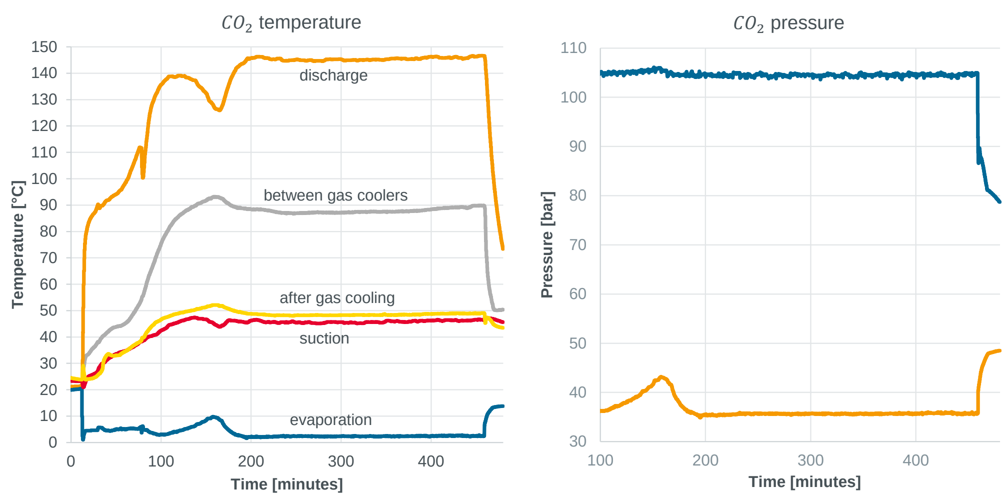

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Conclusions and next steps

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- The prototype heat pump operates satisfactorily in accordance to the design specifications
- Simultanesous generation of high (>130°C) and low (<5°C) water temperatures is proven with stable heat pump response
- Optimal process integration into spray drying plants is required to maximize plant energy savings

Next steps:

- Improvements of prototype installation with focus on heat transfer on the air side
- Development of automated control system for water circuits
- Industrial scale prototype (heating capacity > 1.5 MW)

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TRANSPAC : Transcritical Heat Pump Solution for industrial dryers

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Keywords:
High Temperature Heat Pump, Transcritical Cycle, Drying, Heat Recovery

Abstract

INTRODUCTION

Thermal losses associated with drying operations in the industry represent approximately 40 TWh in France. The energy challenge of recovering calories from air extracted from dryers is therefore very important.

A simple exchanger between the extracted moist air and the incoming dry air makes it possible to recover only a small quantity of the calories of the extracted air, between 10 and 15%.

To recover most of the energy lost, it is necessary to cool the extracted air down to a temperature level sufficiently low to condense a significant part of the water contained in the air. The calories recovered will then be at a low temperature level, generally not usable on the dryer itself. To raise this temperature level, it is necessary to use a heat pump.

The energy recovered by cooling the exhaust moist air can then be transferred to the fresh air to preheat it before entering the dryer.

In drying applications, the fresh air has to be heated a lot (eg from 60 to 120 ° C). With a conventional heat pump, this great heating need (60°C for the example) leads to a low coefficient of performance (approximately 2), which is insufficient to ensure the economic profitability of the installation.

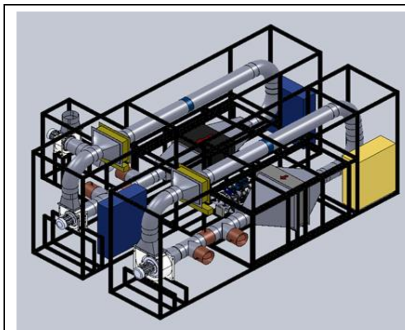
It was therefore necessary to develop an innovative heat pump to ensure the profitability of the installation.

METHODS

A model of integration optimization of high temperature heat pumps has been developed and proved that, at these temperature levels, the use of a transcritical cycle allows to double the coefficient of performance of the Heat Pump compared to a conventional cycle (Besbes et al. 2014). This work has resulted in a patent (Peureux et al.,2014).

Then, an experimental study dealing with the energy performance of a transcritical HP using the R32 as working fluid was carried out (Besbes et al., 2015).

A pilot of 30 kW thermal to ensure a heat output of 30 kW at 120 ° C was built. It consists of two closed airflow loops in which humid air circulates at 50 ° C (heat source / cold loop), and dry air at 60 ° C (heat sink / hot loop).



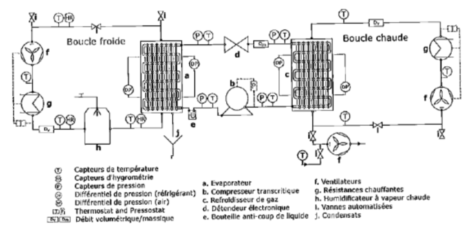


FIGURE 5.2 – Schéma du banc d'essai

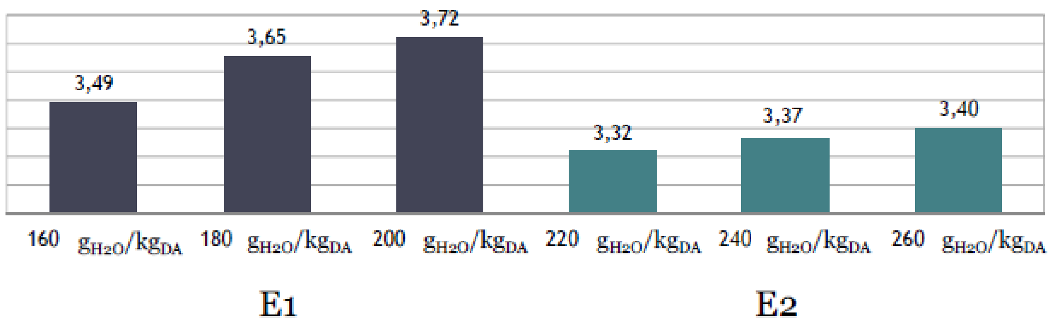
RESULTS

The tests showed a coefficient Of Performance close to 4 under the conditions of an industrial dryer.

	T moist air (°C)	RH(%)	Inlet air T (°C)	Outlet air T(°C)	Exp COP
Nominal point	**50**	**100**	**59**	**117**	**3.70**
Variation of inlet air T to be heated	_	_	**↘ 55**	116.5	**3.75**
	_	_	**↘ 50**	116.5	**4.01**
Variation of moist air humidity	_	**↘ 90**	60	116.5	**3.61**
	_	**↘ 80**	_	_	**3.58**
Variation of moist air T	**↘ 45**	90	_	107	**3.51**

Similar tests were performed with HFO 1234ze-E to investigate the feasibility of delivering hot air at 150°C. The COP of the transcritical HP was measured for several operating conditions

For the first set of operating conditions (E1), the medium at the evaporator inlet is a humid air at 82°C with different level of absolute humidities. The medium at the gas cooler inlet is an ambient air preheated from T_{amb} to 90°C to simulate a preheater. For the second set (E2), the medium at the evaporator inlet is a humid air at 82°C with different level of absolute humidities. The medium at the gas cooler inlet is an ambient air preheated from T_{amb} to 100°C to simulate a preheater. The figure below shows the measured COP.



COP values for both sets E1 and E2

These COP can be increased with the use of an internal heat exchanger. This option was simulated for the first test of the E2 set and allows a 6% COP improvement.

CONCLUSION

A transcritical HP pilot using the HFC R32 and the HFO R1234ze-E was developed and is able to supply 150°C hot air. The COP of the transcritical HP was measured for several operating conditions and the worst operating conditions lead to a COP equal to 3.32.

The challenge now is to produce an industrial demonstrator and a project supported by the French Energy Agency is in progress.

The first two years were dedicated to the studies (choice of refrigerant and oil pair, specifications, sizing and selection of components). This third year is devoted to the realization and implementation of the demonstrator on an industrial site.

The operating conditions of the industrial dryer on which the Heat Pump will be installed are the following: heating of the incoming fresh air from 90 to 150 ° C. The Heat Pump energy is taken from the exhaust humid air at 80 ° C containing about 200 g water/kg. A COP close to 4 is expected.

REFERENCES

- **Besbes** Karim, Zoughaib Assaad, de Carlan Florence, and Peureux Jean-Louis. 2014. “Exergy Based Methodology for Optimized Integration of Heat Pumps in Industrial

Processes.” International Refrigeration and Air Conditioning Conference at Purdue. http://docs.lib.purdue.edu/iracc/1435.

- **Peureux** Jean-Louis, Bourig Ali, Zoughaib Assaad, Besbes Karim. "Equipement including a heat pump for heating an external fluid with a large temperature differential." International Patent WO2014020255 (A1), 2014.
- **Besbes** Karim, Zoughaib Assaad, Florence de Carlan, and Peureux Jean-Louis. 2015. “A R-32 TRANSCRITICAL HEAT PUMP FOR HIGH TEMPERATURE INDUSTRIAL APPLICATIONS.” International Congress of Refrigeration, Yokohama.

TRANSPAC

Transcritical Heat Pump Solution for industrial dryers

Florence DE CARLAN

EDF FRANCE

2019/09/09 - 2nd Conference on High Temperature Heat Pumps - Copenhagen 1

Industrial drying, a major challenge for energy savings

- **Dryers energy consumption in France : 43 TWh/an**
  - **Paper industry : 39 %**
  - **Food industry : 23 %**
  - **Chemical industry : 13 %**
  - **Materials industry : 11 %**
- **90% of the energy consumed by the dryer is discharged into the output moist air**
- **A preheater can recover only about 10% of the energy**
- **A heat pump makes it possible to cool the moist air below its dew point and thus to recover a lot of energy**

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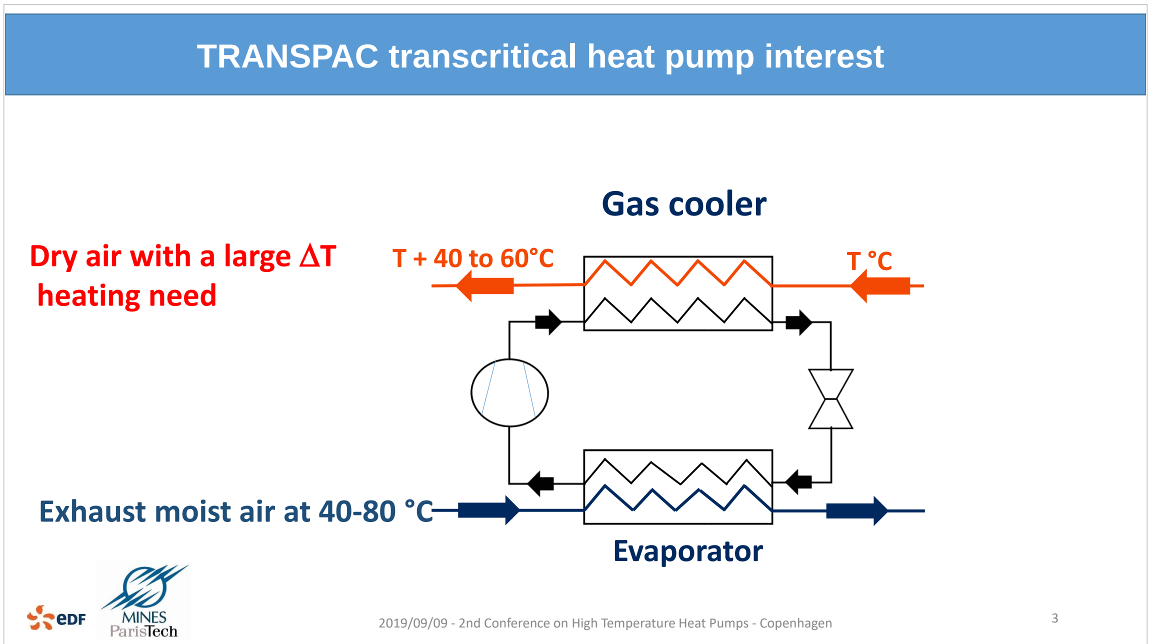
TRANSPAC transcritical heat pump interest
Gas cooler
Dry air with a large ΔT heating need
T + 40 to 60°C
T °C
Exhaust moist air at 40-80 °C
Evaporator
EDF
MINES ParisTech
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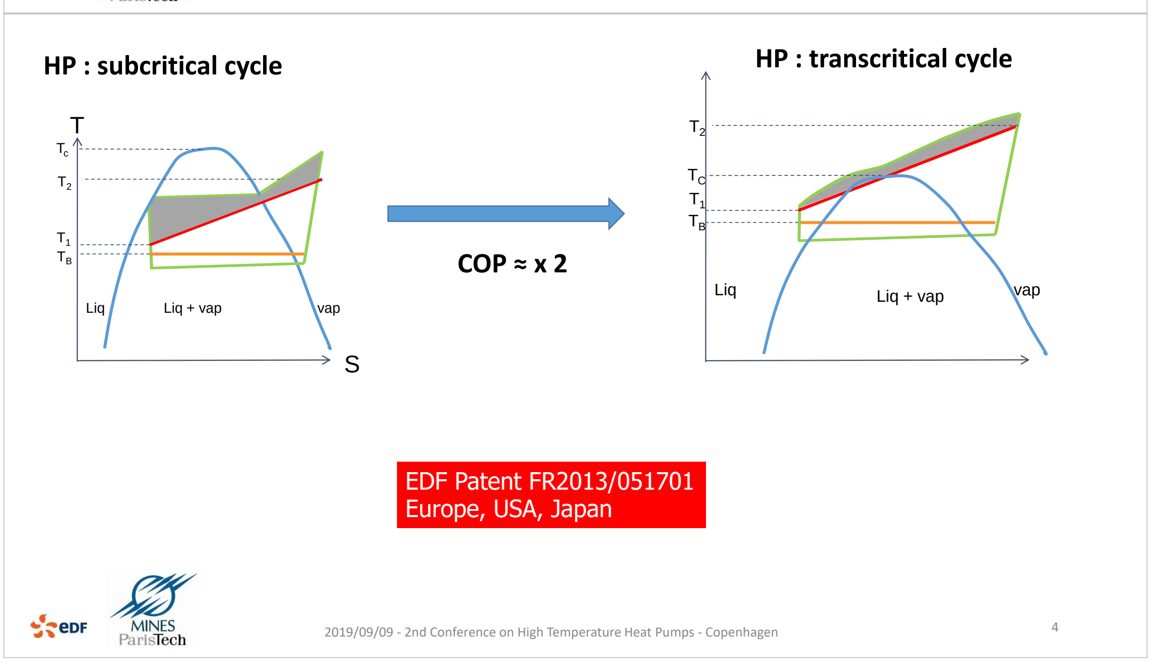
3


HP : subcritical cycle
HP : transcritical cycle
T
S
Liq
Liq + vap
vap
COP ≈ x 2
EDF Patent FR2013/051701
Europe, USA, Japan
EDF
MINES ParisTech
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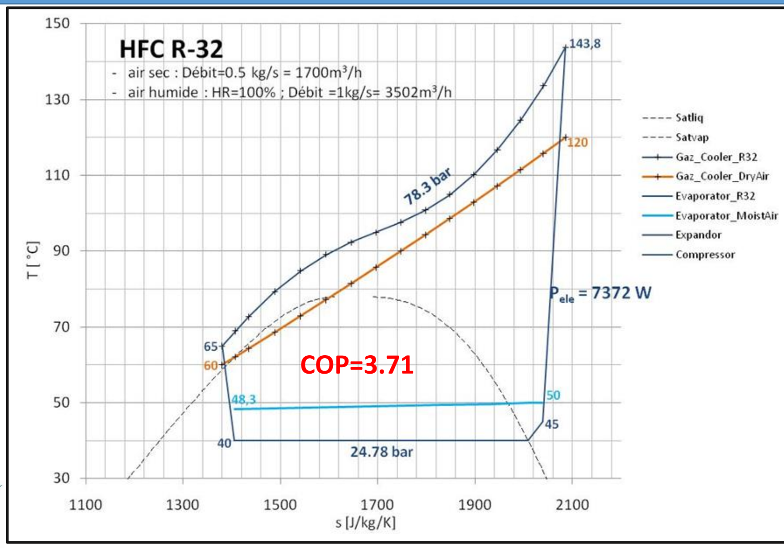
Theoretical cycle



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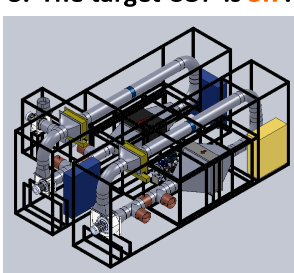
Realization of a test bench

1. **Realization of a transcritical heat pump operating with R-32 and a heat source (saturated humid air) at 50 ° C to heat air from 60 up to 120 ° C.**

2. **Gaz cooler power is 30 kW.**

3. **The target COP is 3.7.**



L x l x H = 5500 x 4100 x 2100 mm³

Test bench photograph at MinesParisTech

EDF

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MINES ParisTech

Experimental results with R32

	$T_{moist\ air}$ (°C)	RH(%)	Inlet air T (°C)	Outlet air T(°C)	Exp COP
Nominal point	**50**	**100**	**59**	**117**	**3.70**
Variation of inlet air T to be heated	_	_	**↘ 55**	116.5	**3.75**
	_	_	**↘ 50**	116.5	**4.01**
Variation of moist air humidity	_	**↘ 90**	60	116.5	**3.61**
	_	**↘ 80**	_	_	**3.58**
Variation of moist air T	**↘ 45**	90	_	107	**3.51**

The test bench proved the efficiency of the R-32 transcritical heat pump to heat air from 60 to 120 ºC with a COP close to 4 by recovering energy in moist air at 50 ºC.

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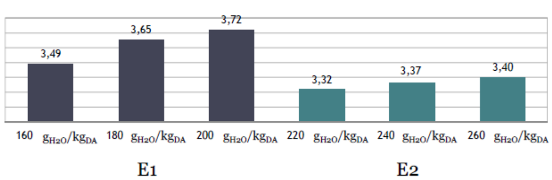
Experimental results with HFO R1234ze-E to increase the supply air T to 150°C

Operating conditions

		E1	E2
Evaporator	Humid air T	82°C	82°C
	Absolute humidity (g_{H2O}/kg_{DR})	160-180-200	220-240-260
Gas Cooler	Ambient air T	90°C	100°C
	Target T	150°C	150°C

COP



Simulation of an internal heat exchanger shows a 6% COP improvement.

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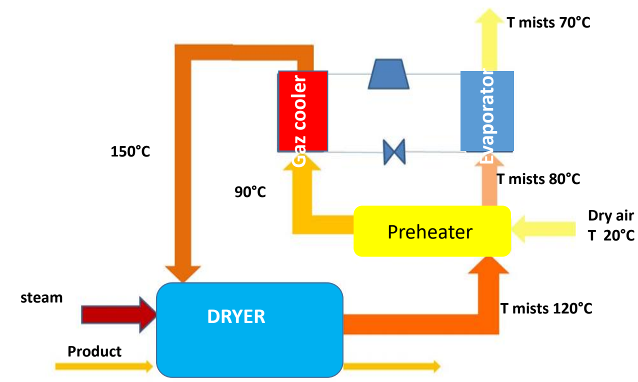
MINES ParisTech

First demonstration at scale 780 kW in paper industry, with different dryer conditions

- Project supported by th French Energy Agence
- Objective : Air heating from 90°C to 150°C (780 kW)
- Planning : installation 2020
- Gaz savings: **-7%**
- CO_2 reduction: **1400 t/an**



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Challenges achieved

- Refrigerant : performances, environmental impact, HP, density, availability, cost…
- Oil : works are in progress
- Compressor suitable for transcritical cycle conditions : HP, suction and discharge T, cost, valves adapted to the refrigerant density…

All components have been sized.
The study of integration including the civil engineering is in progress.

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Potential applications

Starch industry :
- **Air to be heated from 30°C up to 120°C**
- **Moist air at 60°C**

Dryers of wood pellets :
- **Air to be heated up to 150°C**
- **Moist air at 80°C and 120 gwater/kg**

Tiles and bricks

Nonwoven technical textile

Paper pulp dryer

Pet food dryer

Heat network to heat water from 70 to 110°C

...

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Questions ?

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Experimental results of HFO/HCFO refrigerants in a laboratory scale HTHP with up to 150 °C supply temperature

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Extended Abstract

Keywords:
High temperature heat pump, HFO, HCFO, R1336mzz(Z), R1233zd(E), R1224yd(Z), efficiency, COP

1. Introduction

The range of electrically driven high temperature heat pumps (HTHP) for industrial applications has grown steadily in recent years. At least 26 industrial HTHP products are commercially available, capable of delivering heat at sink temperatures from 90 to 160 °C [1–4]. Heat pumps of this type are available in a wide range of heat outputs (20 kW to 20 MW) and the technology will be further commercialised in the coming years to play a key role in the decarbonisation of the industrial sector. Several presentations at the industrial heat pumps sessions at the ICR 2019 conference in Montréal confirmed this trend. By switching from fossil fuels to renewable energies and increasing energy and resource efficiency, the CO_2 footprint can be significantly reduced.

The use of industrial HTHPs is particularly interesting for heat recovery applications and various industrial processes, such as drying, steam generation, sterilisation, paper production or food preparation. From a research perspective, HTHP technology is being further developed and the limits of heat supply temperatures and performance figures are further explored. Various experimental R&D projects are currently running on an international level to push HTHPs from the laboratory scale towards industry. The main research objectives are (1) extending the limits of heat source and sink temperatures to higher levels, (2) improvement of heat pump efficiency (COP) by multi-stage cycles and oil-free compressors, (3) development of temperature-resistant components, such as valves and compressors, and (4) developing and testing of new synthetic environmentally friendly refrigerants with low GWP.

The choice of the optimal refrigerant is subject of much debate. The partially fluorinated hydrocarbons (HFC) R134a, R245fa and R365mfc have a greenhouse effect (GWP of 1’300, 858 and 804 [5]) and are experiencing a phase-down (i.e. production and consumption) in most industrialised countries. In Europe, the F-Gas regulation prohibits the use and reduces the market availability of greenhouse refrigerants. Consequently, only refrigerants with a GWP < 150 may be used in new commercial heat pumps starting from 2022. In Switzerland, the legal basis for refrigerants is regulated in the Chemicals Risk Reduction Ordinance (ChemRRV) [6] and industrial heat pumps with heat source capacity >600 kW are affected by the HFC ban.

Beside natural refrigerants, such as water (R718), CO_2 (R744), ammonia (R717), butane (R600), propane (R290), and pentane (R601), the application of the 4th generation of new synthetic hydrofluoroolefin (HFO) and hydrochlorofluoroolefin (HCFO) refrigerants with low environmental impact is becoming increasingly important as drop-in replacements for HFC in future HTHPs. Even though HCFOs contain a chlorine atom in their structure and do not comply with the legal requirements of the Montreal Protocol (ODP of zero) there are national regulations, like the ChemRRV [6] that allow the use of HCFO refrigerants with an OPD < 0.0005.

At NTB Buchs a laboratory scale HTHP has been developed as part of the SCCER-EIP project [7]. The developed HTHP is single-stage, operates with a variable-speed reciprocating compressor, and contains a continuously adjustable internal heat exchanger (IHX) for superheating control. A viscous POE oil (173 mm^2/s at 40 °C) was used to achieve sufficient lubrication at high temperatures with the refrigerants. The basic functionality of the HTHP and first experimental results with R1233zd(E) and R1336mzz(Z) have already been published in previous papers by Arpagaus et al. [8–12].

This paper examines the performance of R1336mzz(Z) (Opteon™MZ from Chemours), R1233zd(E) (Solstice®zd from Honeywell), and R1224yd(Z) (AGC Chemicals) in the same laboratory HTHP in a drop-in test. A parameter study was carried out to investigate the COP as a function of the temperature lift between heat source temperatures of 30 to 80 °C and heat sink temperatures of 80 to 150 °C.

2. Investigated HFO/HCFO refrigerants

Table 1 lists the thermodynamic, environmental and safety properties of the investigated refrigerants.

Table 1: Thermophysical, environmental, and safety properties of the tested HFO/HCFO refrigerants.

Refrigerant	R1233zd(E)	R1224yd(Z)	R1336mzz(Z)	R245fa
Brand (manufacturer)	Solstice®zd (Honeywell) Forane®HTS 1233zd (ARKEMA)	AMOLEA®1224yd (AGC Chemicals) [13]	Opteon™MZ (Chemours)	Genetron®245fa (Honeywell)
Molecular formula	E-CF_3-CH=CHCl	Z-CF_3-CF=CHCl	Z-CF_3-CH=CH-CF_3	$CHF_2CH_2CF_3$
Molecular weight [kg/kmol]	130.5	148.62	164.06	134.05
Critical temperature [°C]	165.6	155.5	171.3	154.0
Critical pressure [bar]	35.7	33.4	29.0	36.5
Normal boiling point [°C]	18.0	14.6	33.4	14.9
ODP (CFC-11=1) [-]	0.00034 [5], 0.00030 [14]	0.00023 [15]	0	0
GWP (CO_2=1, 100 years) [5] [-]	1 [5], <5 [14]	0.88 [15]	2 [5]	858
Atmospheric lifetime [days]	~14 [16], 26 [5], 36 [14], 40.4 [17]	20 [15]	22 [5]	7.7 years [18]
LC50 (rat, 4 h) [ppm v/v]	120’000	>213’000	102’900	>203’000
Occupational exposure limit (OEL) [ppm v/v]	800	1’000	500	300
Safety classification (ASHRAE)	A1	A1	A1	B1
Final degradation products in the atmosphere [19]	CO_2, HF, HCl	similar structure like R1234yf with potential for degradation to TFA	CO_2, HF	CO_2, HF
TFA molar yield from degradation	~ 2% [14]		< 20 % [20]	< 10 % [18]

R1233zd(E) has a critical temperature of 165.6 °C and a critical pressure of 36.2 bar and is available as Solstice®zd from Honeywell or as Forane®HTS 1233zd from ARKEMA. Although R1233zd(E) contains a chlorine atom that potentially can participate in the catalytic destruction of the ozone layer, its atmospheric lifetime is sufficiently low (~14 days [16], 40.4 days [17]) so that the compound will not reach the stratosphere and thus not participate in ozone depletion (ODP is 0.00034 [5]). So far, there is a limited investigation on the use of R1233zd(E) for HTHP applications [21]. First experimental results could be presented at the DKV conference 2018 [10] and ICR 2019 [8] with the developed laboratory HTHP system at NTB Buchs. Compared to a basic cycle the integration of an IHX led to approx. 15% COP increase at W60/W110 conditions [8]. The maximum heat sink temperature tested was 150 °C, whereby a COP of 2.1 was achieved with a heat source of 80 °C (70 K lift). At Ulster University another HTHP test facility is being developed to test R1233zd(E) as a part of the CHESTER project [21,22]. Simulation results of Shah et al. [21] showed an up to 8% higher COP with R1233zd(E) compared to R245fa. Further investigation is ongoing to test R1233zd(E) and oil miscibility.

R1224yd(Z) is another HCFO refrigerant designed for use in heat pumps for waste heat recovery and centrifugal chillers. AGC Chemicals (Asahi Glass) markets it as Amolea™1224yd [23]. The physical properties are close to R245fa. Its critical temperature is 155.5 °C and the saturated vapour pressure slightly lower (about 13% smaller at 120 °C) [24]. With an ODP of almost zero (0.00023, atmospheric lifetime of 20 days) and a GWP of 0.88 it has a low environmental impact [15]. The toxicity is indicated with a value of 1'000 ppm of OEL was provided as a maximum value for organic compounds [25]. In addition, AMOLEA™ 1224yd is classified as A1, which indicates non-flammability and low-toxicity.

At the ICR 2019 conference, Kaida et al. [24] presented first experimental results of R1224yd(Z) in a commercial SGH165 heat pump (with economizer and IHX) developed by KOBELCO and the Japanese electric utilities. Drop-in tests at an operating point W50/W95 (45 K temperature lift) revealed a 3% higher heating capacity and 12 % higher COP compared to R245fa. The performance improvements were attributed to increased refrigerant mass flow rate, decreased viscosity, and decreased required pressure ratio (higher adiabatic compressor efficiency). The chemical stability and compatibility with PAG oil, O-rings, and motor insulation material was comparable with R245fa. Overall, R1224yd(Z) was suggested as suitable R245fa alternative for HTHPs.

R1336mzz(Z) has a higher critical temperature of 171.3 °C at a feasible pressure of 29 bar. Chemours commercialized R1336mzz(Z) under the brand Opteon™MZ. R1336mzz(Z) is safety class A1, has a GWP of 2, an ODP of 0, and an atmospheric life of only 22 days [5]. Polyolester oil (POE) is recommended as lubricant, as it is fully miscible over wide ranges of temperatures and compositions [26,27].

Apart from GWP and ODP, the degradation products of refrigerants in the atmosphere and their effects on human health and the environment are a hot topic repeatedly featured in the recent open public [28]. The atmospheric degradation of HCFCs, HCFCs and HFOs is initiated by reaction with OH radicals leading to the formation of halogenated carbonyl compounds which are further oxidised to hydrofluoric acid (HF), hydrochloric acid (HCl), formic acid (HC(O)OH), CO_2 and in some cases trifluoroacetic acid (TFA, $CF_3C(O)OH$) [14,18–20,29]. For comparison, the >C=C< double bond in HFOs reacts two orders of magnitude faster with OH radicals than R134a [19]. For example, the molar yield of R134a to decompose into TFA is 7 to 21%, while for R245fa it is <10% [18]. As a result of their long atmospheric lifetimes (13.4, 7.7 years [18]), the gaseous TFA can be widely distributed in the atmosphere, descends via rainfall to earth and accumulates in various water bodies, including rivers, streams, lakes and wetlands, as well as "terminal sinks" like salt lakes, playas and oceans [28,29]. HF and HCl neutralize quickly due to the buffer capacity of surface water [19].

HFO and HCFO decompose much faster into their final products, which means they can occur locally at the point of emission and have direct effects. The molar yields of TFA formation from the degradation depends on the refrigerant [8]. For example, the degradation of R1234yf leads to a 100% molar yield of TFA. Interestingly, the HFO and HCFO refrigerants suitable for HTHP show little or no TFA and therefore have a negligible impact on the environment, which seems to be promising. R1233zd(E) decomposes to max. 2% TFA [14], whereas the yield for R1336mzz(Z) (containing two CF_3-CH= groups) is expected to be < 20% [20]. As R1224yd(Z) has a similar molecular structure like R1234yf there is potential for degradation to TFA.

On the other hand, over 200 million tons of TFA are already naturally present in the oceans from natural sources such as undersea vents and volcanic activity. In a worst case scenario of unregulated use of HCFCs, HCFCs and HFOs by 2050, Solomon et al. [29] estimated a total additional contribution of TFA to the oceans of <7.5% of the approx. 200 ng acid equivalents/L present at the start of the millennium, which was judged to be a negligible risk on the aquatic organisms.

3. Experimental Results and Discussion

After heating up the HTHP to the desired heat source and heat sink temperatures, the water flow rates in the two hydraulic circuits were adjusted by the pumps to receive constant temperature differences of 3 K at the heat source (ΔT_{Source}) and 5 K at the heat sink (ΔT_{Sink}). Mean values of at least five minutes were used for the data analysis. The heating COP was determined from the measured heating capacity ($\dot{Q}_{Sink}$) and the electrical power consumption of the compressor (P_{el}). Table 1 and Figure 2 (A to D) summarize the results of the parameter studies with the refrigerants R1224yd(Z), R1233zd(E) and R1336mzz(Z).

Table 2: Operating conditions and performance parameters of the experimental runs with refrigerants R1224yd(Z), R1233zd(E), and R1336mzz(Z) in the laboratory HTHP (1-stage cycle with IHX).

	No.	$T_{Source,in}$ °C	$T_{Sink,out}$ °C	ΔT_{Lift} K	ΔT_{Source} K	ΔT_{Sink} K	$T_{Suction}$ °C	$T_{Discharge}$ °C	ΔT_{SC} K	ΔT_{SH} K	p_{Cond} bar	p_{Evap} bar	p_{ratio} -	P_{Comp} kW	$\dot{Q}_{Sink}$ kW	$\dot{Q}_{Source}$ kW	$\dot{Q}_{Loss}$ kW	η_{2nd} %	COP -
R1224yd(Z)	A1	30	70	40	3.0	4.9	57	101	22	5	5.7	1.5	3.8	1.2	4.0	3.5	0.6	40%	3.4
	A2	30	89	59	3.0	5.0	68	121	30	5	9.2	1.5	6.1	1.3	3.2	2.9	0.9	41%	2.5
	A3	40	70	30	3.1	5.0	60	100	17	5	5.9	2.1	2.9	1.3	5.5	5.0	0.8	38%	4.3
	A4	40	89	49	3.0	5.0	73	122	27	5	9.4	2.1	4.5	1.5	4.7	4.0	0.8	42%	3.1
	A5	40	110	70	3.0	5.0	86	143	36	5	14.3	2.1	6.9	1.7	3.7	3.2	1.2	40%	2.2
	A6	50	89	39	3.1	5.0	75	116	21	5	9.5	2.8	3.4	1.7	6.7	7.6	2.6	42%	3.9
	A7	50	109	59	3.0	5.0	90	141	31	5	14.4	2.8	5.2	2.0	5.3	4.7	1.5	41%	2.6
	A8	60	90	30	3.8	5.0	78	115	17	5	9.7	3.5	2.8	1.9	8.4	9.3	2.8	37%	4.5
	A9 Ref	**60**	**110**	**50**	**3.2**	**5.0**	**92**	**142**	**26**	**5**	**14.7**	**3.7**	**4.0**	**2.3**	**7.5**	**7.8**	**2.6**	**42%**	**3.2**
	A10	60	130	70	3.1	5.2	107	161	34	5	21.5	3.8	5.7	2.7	5.9	5.9	2.7	38%	2.2
	A11	70	100	30	4.4	4.9	88	126	17	5	12.2	4.5	2.7	2.3	9.8	10.6	3.1	35%	4.3
	A12	70	110	40	4.2	5.0	95	137	21	5	14.9	4.7	3.2	2.6	9.3	10.1	3.4	38%	3.6
	A13	70	130	60	3.5	5.0	109	162	29	5	21.8	4.9	4.4	3.1	8.0	8.6	3.7	38%	2.6
	A14	70	139	69	3.1	5.0	115	175	31	5	25.2	5.0	5.1	3.3	7.3	7.3	3.3	37%	2.2
	A15	79	120	40	4.8	5.1	104	148	20	5	18.0	5.9	3.1	3.1	10.9	11.6	3.7	36%	3.5
	A16	80	130	50	4.4	4.9	111	161	24	5	22.0	6.1	3.6	3.5	10.0	10.7	4.1	36%	2.9
	A17	80	140	60	3.6	5.0	120	172	21	5	25.4	6.3	4.0	3.8	8.3	8.6	4.1	32%	2.2
	A18	80	150	70	3.1	5.1	130	185	9	5	28.7	6.4	4.5	4.1	5.7	5.0	3.3	23%	1.4
R1223zd(E)	B1	30	69	39	2.9	5.1	56	94	23	5	5.1	1.3	3.9	1.1	3.5	3.3	0.9	37%	3.3
	B2	30	89	59	2.9	5.0	64	118	29	5	8.3	1.3	6.3	1.2	2.8	2.5	0.9	38%	2.3
	B3	40	70	30	3.0	5.1	59	101	18	5	5.2	1.8	2.9	1.2	4.8	4.2	0.5	36%	4.1
	B4	40	89	49	3.0	5.0	74	128	27	5	8.3	1.8	4.6	1.4	4.3	3.6	0.6	43%	3.1
	B5	40	109	69	3.0	5.0	84	143	35	5	12.9	1.8	7.2	1.5	3.3	2.9	1.1	39%	2.2
	B6	50	91	41	3.1	5.0	76	119	23	5	8.7	2.4	3.6	1.6	5.8	5.5	1.4	41%	3.7
	B7	50	109	59	3.0	4.9	90	143	32	5	12.9	2.4	5.3	1.8	4.9	4.3	1.3	41%	2.7
	B8	60	89	29	3.4	5.0	78	115	17	5	8.5	3.2	2.6	1.7	7.8	8.4	2.2	37%	4.7
	B9 Ref	**60**	**111**	**51**	**3.0**	**5.0**	**93**	**141**	**27**	**5**	**13.5**	**3.3**	**4.1**	**2.1**	**6.5**	**6.7**	**2.3**	**41%**	**3.1**
	B10	60	130	70	3.0	5.0	107	167	36	5	19.3	3.3	5.9	2.4	5.5	4.9	1.9	39%	2.3
	B11	70	110	40	3.8	5.0	95	139	22	5	13.4	4.2	3.2	2.3	8.6	9.2	2.9	39%	3.7
	B12	70	130	60	3.4	6.1	110	159	30	5	19.6	4.3	4.6	2.8	7.3	7.4	2.9	39%	2.6
	B13	78	110	32	4.5	5.0	97	141	18	5	13.5	4.9	2.8	2.5	10.3	10.8	2.9	36%	4.2
	B14	80	130	50	3.0	5.0	112	161	25	5	19.7	5.6	3.5	3.2	9.9	10.3	3.6	38%	3.1
	B15	80	149	69	3.3	5.1	126	185	31	5	27.5	5.7	4.8	3.7	7.6	7.9	3.9	34%	2.1
R1336mzz(Z)	C1	30	69	39	3.0	5.0	50	91	20	5	3.2	0.7	4.5	0.8	1.9	1.8	0.6	29%	2.5
	C2	40	70	30	3.0	5.0	58	94	18	5	3.2	1.0	3.1	0.9	3.0	2.6	0.5	30%	3.5
	C3	40	90	50	3.1	5.1	68	113	26	5	5.5	1.0	5.4	0.9	2.4	2.2	0.7	35%	2.6
	C4	40	108	68	3.1	4.8	75	126	30	5	8.5	1.0	8.3	1.0	1.9	1.7	0.7	34%	1.9
	C5	50	90	40	3.0	5.0	75	110	23	5	5.6	1.5	3.8	1.1	3.7	3.4	0.8	36%	3.3
	C6	60	90	30	3.0	5.0	78	108	17	5	5.7	2.0	2.8	1.2	5.2	5.1	1.1	34%	4.2
	C7 Ref	**60**	**110**	**50**	**3.0**	**4.9**	**91**	**128**	**28**	**5**	**9.0**	**2.0**	**4.4**	**1.5**	**4.3**	**4.1**	**1.3**	**38%**	**3.0**
	C8	60	129	69	3.1	5.0	105	148	38	5	13.5	2.0	6.7	1.6	3.3	3.2	1.5	36%	2.1
	C9	70	110	40	3.0	5.0	94	127	22	5	9.1	2.8	3.3	1.7	6.0	6.4	2.0	38%	3.6
	C10	80	111	31	3.7	5.1	98	128	18	5	9.4	3.5	2.7	1.8	7.8	8.6	2.6	34%	4.3
	C11	80	130	50	3.0	5.1	111	149	27	5	14.0	3.7	3.8	2.2	6.7	7.3	2.9	38%	3.0
	C12	80	150	70	3.0	5.1	125	172	36	5	20.4	3.7	5.5	2.5	4.6	4.9	2.8	30%	1.8

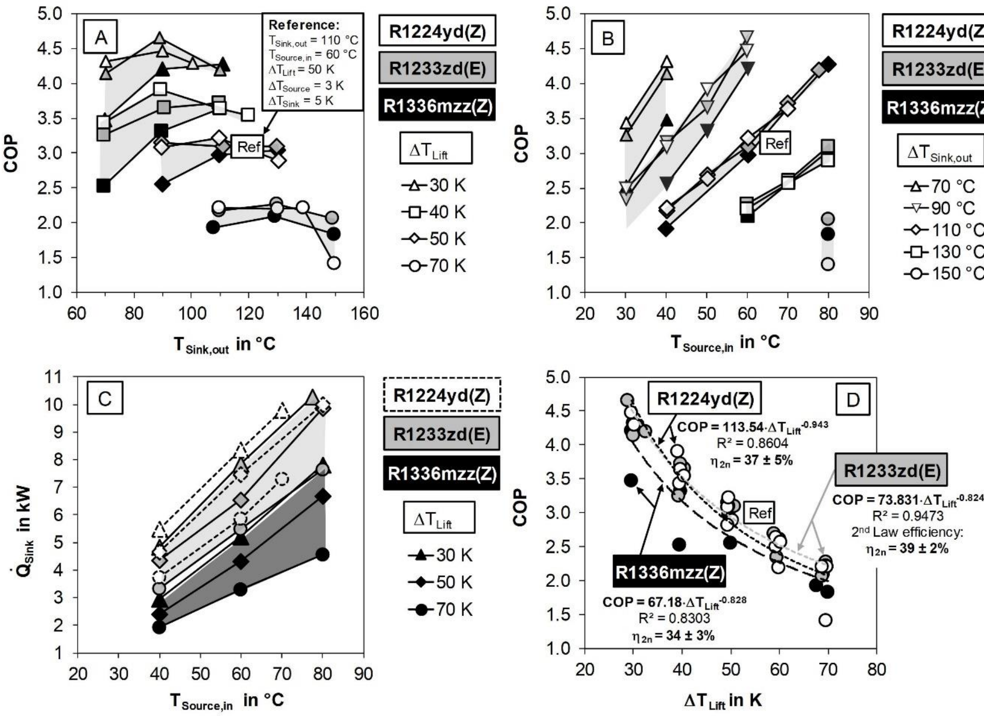

Figure 1: Experimental results of the investigated laboratory HTHP with the refrigerants R1224yd(Z), R1233zd(E), and R1336mzz(Z). (A) COP as a function of the heat sink temperature at different temperature lifts, (B) COP as a function of the heat source at different heat sink temperatures, (C) heating capacity as a function of the heat source temperature at 30, 50 and 70 K temperature lift, and (D) COP fit curves of measurement data with 2nd Law efficiencies.

Figure 1 (A) shows the COP of the HTHP as a function of the heat sink outlet temperature ($T_{Sink,out}$) and various temperature lifts (ΔT_{Lift}). At the reference point conditions (Ref) W60/W110, a heating COP of 3.2, 3.1, and 3.0 was achieved for R1224yd(Z), R1233zd(E), and R1336mzz(Z), respectively.

Up to about 110 °C, R1224yd(Z) and R1233zd(E) delivered a slightly higher COP than R1336mzz(Z), which is attributed to the higher heating capacities (see Figure 1, C) and the smaller relative heat losses at the same temperature conditions. Heat losses of about 21 ± 7% were estimated from an energy balance with the main origin at the compressor. At the higher temperatures, the deviations between the measured COPs were within the measurement uncertainty of about ± 0.2 COP. In this study, the maximal tested heat sink temperature was 150 °C. R1336mzz(Z) achieves potentially higher condensing temperatures due to the higher critical temperature of 171.3°C compared to 166.5 °C of R1233zd(E) and 155.5 °C of R1224yd(Z). In addition, an increase of the temperature glide on the heat sink from 5 to 30 K further increased the COP by 15% [8], which is advantageous in processes with low return temperatures. A larger temperature glide improved the heat transfer in the condenser.

Figure 1 (B) shows the COP as a function of the inlet temperature of the heat source. The increase in efficiency with higher source temperature is evident. The COP data of R1224yd(Z) were comparable to R1233zd(E) except for W80/W150 where the COP decreased due to the narrowing of the two-phase region (in the p-h diagram) near the critical temperature of 155.5 °C.

Figure 1 (C) shows the heating capacity ($\dot{Q}_{Sink}$) as a function of the heat source inlet temperature ($T_{Source,in}$) at constant temperature lifts (ΔT_{Lift}). Overall, the heating capacity of R1336mzz(Z) was about 46 to 76% lower than that of R1233zd(E). This is due to the lower volumetric heating capacity (VHC). A compressor with a larger swept volume would be required to achieve similar heating capacities as R1233zd(E) and R1224yd(Z). R1233zd(E) provided a heating capacity on 5.8 kW at Ref and approx. 10 kW at W80/W110, which corresponded to the capacity limit of the laboratory system. With R1336mzz(Z) a maximum heating capacity of 7.8 kW could be achieved (W80/W111). The drop-in test showed that the heat capacity of R1224yd(Z) was on average 9% higher than that of R1233zd(E). This is consistent with simulation studies by Arpagaus et al. [30] presented at ICR 2019, where VHC values of 1'600 kJ/m^3, 2'412 kJ/m^3 and 2'639 kJ/m^3 for R1336mzz(Z), R1233zd(E) and R1224yd(Z) were calculated in a single-stage cycle with IHX at W60/W110. A compromise between COP and VHC needs to be found depending on the refrigerant.

Figure 1 (D) shows the COP of the measured experimental data as a function of the respective temperature lift. As expected, the COP values decreased with ΔT_{Lift} and followed a fit curve with an average Carnot efficiency (2nd Law efficiency) of 39% for R1233zd(E), 37% for R1224yd(Z), and 34% for R1336mzz(Z). These values are comparable with the results in another HTHP laboratory setup of Helminger et al. [31] using R1336mzz(Z), but lower than with the commercial HeatBooster technology from Viking Heating Engines AS, which achieves approx. 41% at 20 kW [8,32].

Temperature-resistant compressors and stable lubricating oils are decisive components for the further development and commercialization of HTHPs. The measured suction gas temperature ($T_{Suction}$) in the laboratory HTHP exceeded the motor limit temperature of approx. 110 °C at a heat sink outlet temperature of about 130 °C and higher. However, short-term experiments over several minutes at 150 °C were still possible to run.

Finally yet importantly, the acid number (neutralization number) of the POE oils was measured by manual colorimetric titration (in mgKOH/g oil according to DIN 51558-1) as a measure of oil degradation. The POE oils were analysed after about 100 operating hours in the HTHP after each refrigerant test campaign. Fresh oil was also measured for comparison. Visual inspections revealed a slightly yellowish colour of the oils after operation in the HTHP. Overall, hardly any oil degradation was detected. The neutralisation number for fresh POE oil was 0.04, for R1233zd(E) 0.06, for R1336mzz(Z) 0.05 and for R122yd(Z) 0.25, thus significantly below the 0.5 warning value assumed by the oil supplier FUCHS for HTHP applications. Long-term tests were not the aim of this study.

4. Conclusion

R1336mzz(Z), R1233zd(E) and R1224yd(Z) have been successfully tested in a single stage HTHP with IHX cycle and up to 10 kW heating capacity on a laboratory scale. The operation of the heat pump was demonstrated at 30 to 80 °C heat source and 70 to 150 °C heat sink temperatures (30 to 70 K temperature lifts), for a possible application of waste heat recovery, steam generation or drying. At operating point W60/W110 COPs of 3.2, 3.1 and 3.0 for R1224yd(Z), R1233zd(E) and R1336mzz(Z) were measured. Up to about 110 °C, R1224yd(Z) and R1233zd(E) had a slightly higher COP than R1336mzz(Z) due to higher heating capacities and lower relative heat losses at the same temperature conditions. Due to higher critical temperatures, R1233zd(E) and R1336mzz(Z) were more efficient than R1224yd(Z) at 150 °C heat sink temperature. Otherwise, the differences in COP were within the measurement uncertainty of ± 0.2 COP. The implementation of an IHX increased the COP significantly (approx. 15% for R1223zd(E)) compared to a basic cycle. A further COP increase of approx. 15% was achieved by a higher temperature glide on the heat sink side from 5 to 30 K, which increased subcooling. The very low GWP, the non-flammability, and the negligible environmental impact (i.e. low TFA formation during atmospheric degradation) indicate a high potential for future use as refrigerant in HTHP applications and retrofit systems. The developed HTHP enables the testing of further alternative HFO and HCFO refrigerants with stabilising additives, HFCs like R245fa or R365mfc for direct comparison or other oils (e.g. POE, PAG) in the future. Further efficiency gains could be achieved by reducing heat losses at high temperatures through better insulation of heat pump components and piping.

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Experimental results of HFO/HCFO refrigerants in a laboratory scale HTHP with up to 150 °C supply temperature

C. Arpagaus, S.S. Bertsch

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2nd Conference on High Temperature Heat Pumps (HTHP)
Copenhagen, Denmark
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Content

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- Introduction to high temperature heat pumps (HTHP)
- Suitable HFOs and HCFOs for HTHPs
- System design of the laboratory scale HTHP at NTB Buchs
- Experimental results with R1336mzz(Z), R1233zd(E) and R1224yd(Z)
- Conclusions

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Introduction to high temperature heat pumps (HTHP)

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Publications

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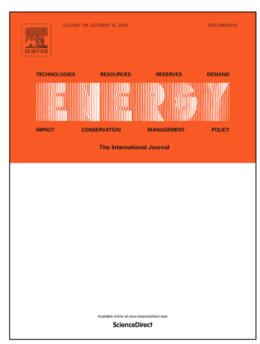
Review Papers

Arpagaus C., Bless F., Schiffmann J., Bertsch S.S.: **Multi-temperature heat pumps: A literature review**, International Journal of Refrigeration, 2016, 69, 437–465.

Arpagaus C., Bless F., Uhlmann M., Schiffmann J., Bertsch S.S.: **Review - High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials**, Energy, 2018, 152, 985-1010



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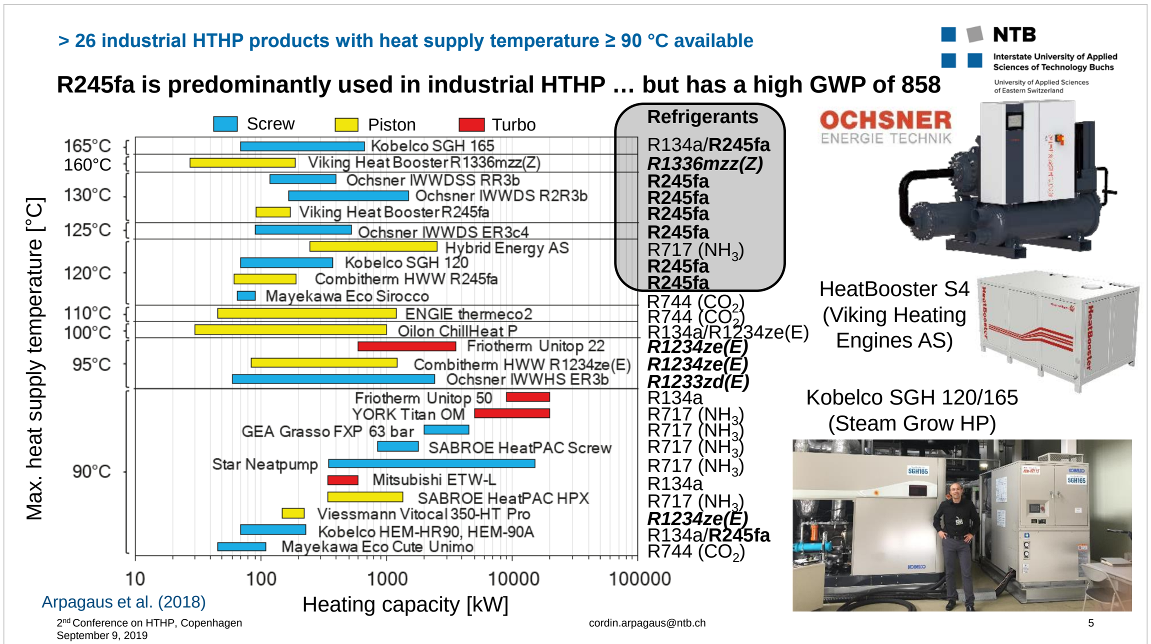
> 26 industrial HTHP products with heat supply temperature ≥ 90 °C available
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R245fa is predominantly used in industrial HTHP … but has a high GWP of 858
Screw
Piston
Turbo
Refrigerants
165°C
160°C
130°C
125°C
120°C
110°C
100°C
95°C
90°C
Max. heat supply temperature [°C]
Kobelco SGH 165
Viking Heat Booster R1336mzz(Z)
Ochsner IWWDSS RR3b
Ochsner IWWDS R2R3b
Viking Heat Booster R245fa
Ochsner IWWDS ER3c4
Hybrid Energy AS
Kobelco SGH 120
Combitherm HWW R245fa
Mayekawa Eco Sirocco
ENGIE thermeco2
Oilon ChillHeat P
Friotherm Unitop 22
Combitherm HWW R1234ze(E)
Ochsner IWWHS ER3b
Friotherm Unitop 50
YORK Titan OM
GEA Grasso FXP 63 bar
SABROE HeatPAC Screw
Star Neatpump
Mitsubishi ETW-L
SABROE HeatPAC HPX
Viessmann Vitocal 350-HT Pro
Kobelco HEM-HR90, HEM-90A
Mayekawa Eco Cute Unimo
R134a/R245fa
R1336mzz(Z)
R245fa
R245fa
R245fa
R245fa
R717 (NH3)
R245fa
R245fa
R744 (CO2)
R744 (CO2)
R134a/R1234ze(E)
R1234ze(E)
R1234ze(E)
R1233zd(E)
R134a
R717 (NH3)
R717 (NH3)
R717 (NH3)
R717 (NH3)
R134a
R717 (NH3)
R1234ze(E)
R134a/R245fa
R744 (CO2)
10
100
1000
10000
100000
Heating capacity [kW]
Arpagaus et al. (2018)
OCHSNER
ENERGIE TECHNIK
HeatBooster S4
(Viking Heating
Engines AS)
Kobelco SGH 120/165
(Steam Grow HP)
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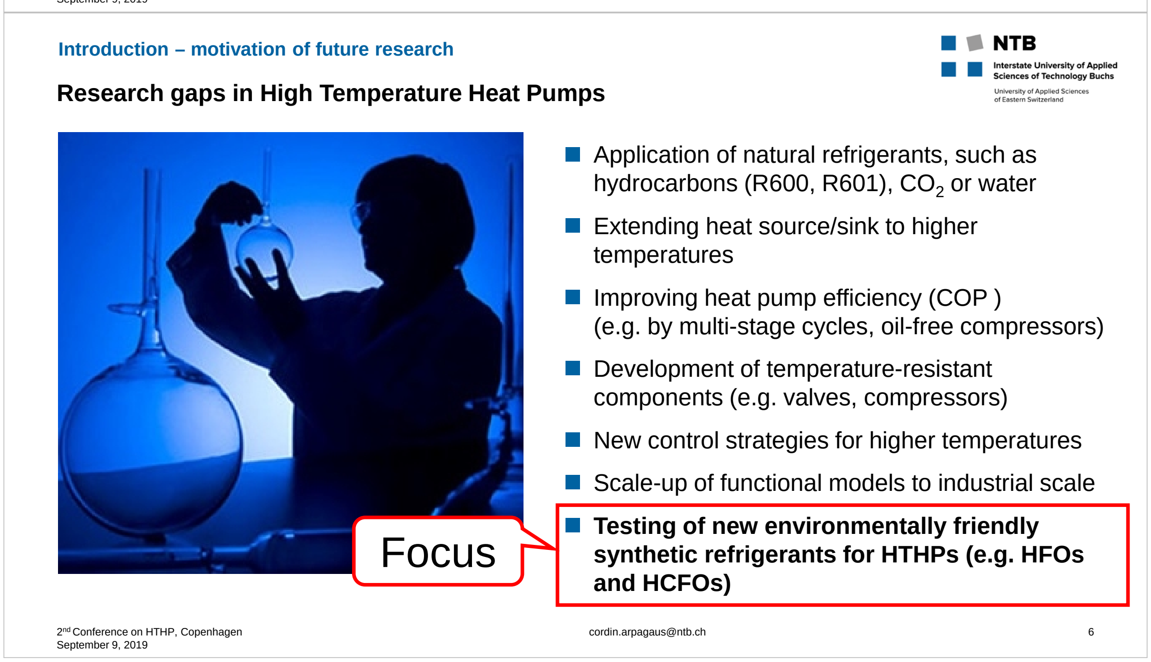
Introduction – motivation of future research
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Research gaps in High Temperature Heat Pumps
Application of natural refrigerants, such as hydrocarbons (R600, R601), CO2 or water
Extending heat source/sink to higher temperatures
Improving heat pump efficiency (COP)
(e.g. by multi-stage cycles, oil-free compressors)
Development of temperature-resistant components (e.g. valves, compressors)
New control strategies for higher temperatures
Scale-up of functional models to industrial scale
Testing of new environmentally friendly synthetic refrigerants for HTHPs (e.g. HFOs and HCFOs)
Focus
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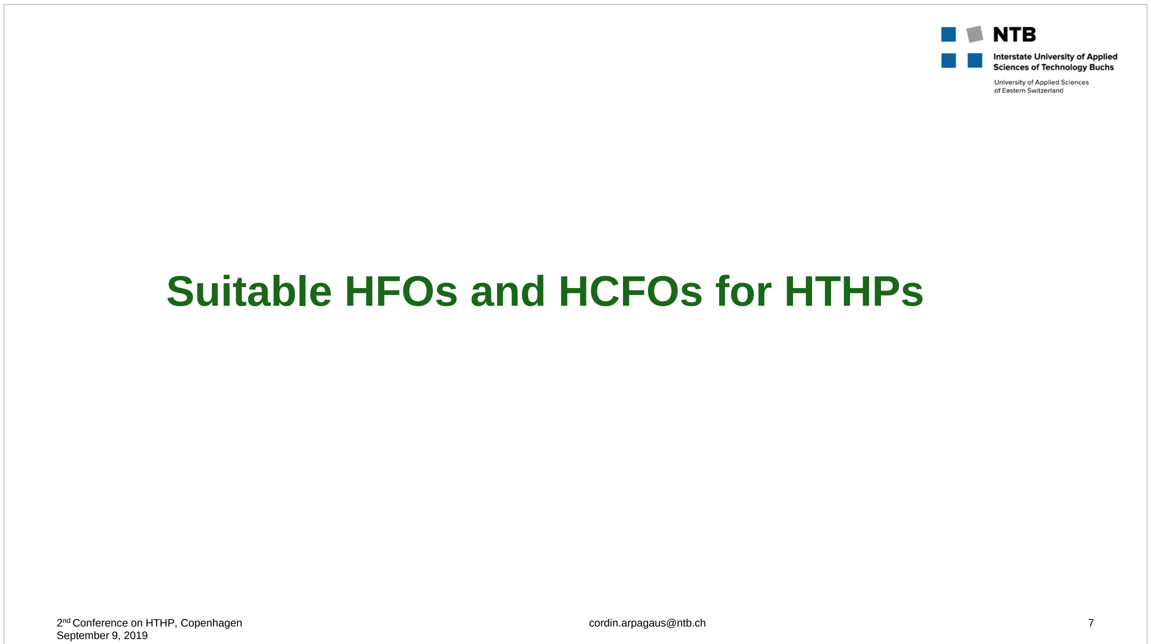
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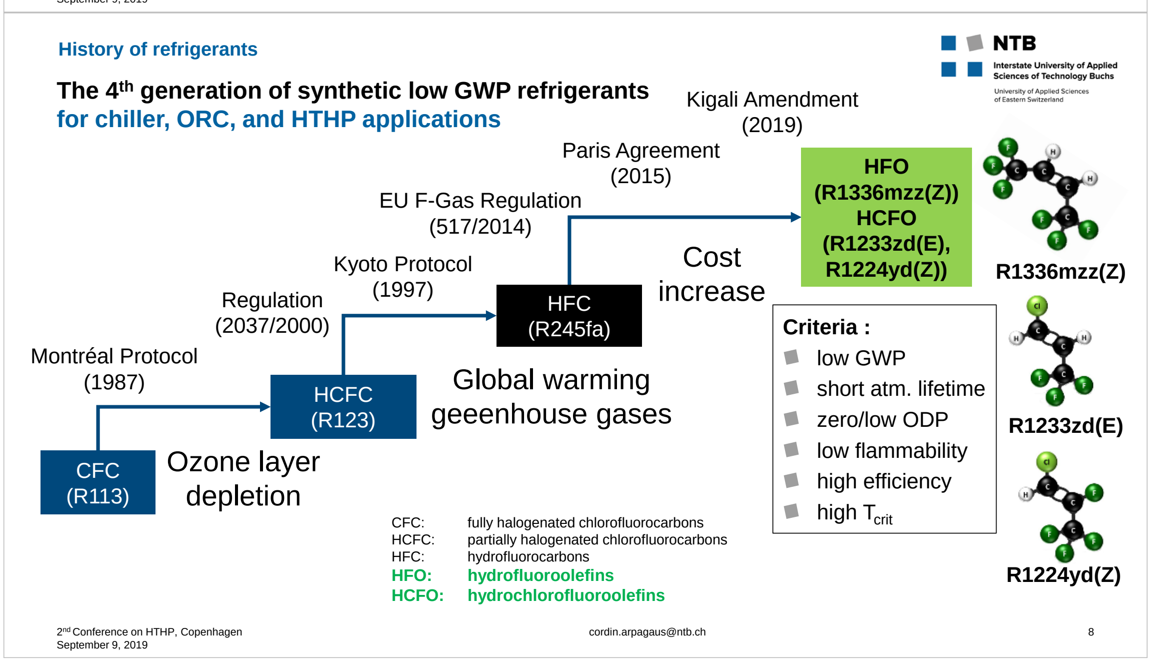
History of refrigerants
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The 4th generation of synthetic low GWP refrigerants
for chiller, ORC, and HTHP applications
Kigali Amendment (2019)
Paris Agreement (2015)
EU F-Gas Regulation (517/2014)
Kyoto Protocol (1997)
Regulation (2037/2000)
Montréal Protocol (1987)
CFC (R113)
Ozone layer depletion
HCFC (R123)
Global warming geeenhouse gases
HFC (R245fa)
Cost increase
HFO (R1336mzz(Z)) HCFO (R1233zd(E), R1224yd(Z))
Criteria :
low GWP
short atm. lifetime
zero/low ODP
low flammability
high efficiency
high T crit
R1336mzz(Z)
R1233zd(E)
R1224yd(Z)
CFC: fully halogenated chlorofluorocarbons
HCFC: partially halogenated chlorofluorocarbons
HFC: hydrofluorocarbons
HFO: hydrofluoroolefins
HCFO: hydrochlorofluoroolefins
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Refrigerant properties

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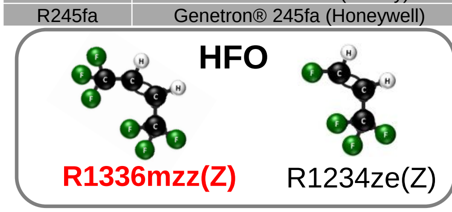
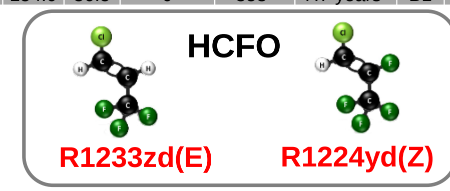
Properties of suitable HFO and HCFO refrigerants for HTHPs

Tested in this study

Refrigerant	Brand (manufacturer)	T_{crit} [°C]	p_{crit} [bar]	ODP [-]	GWP_{100} [-]	Lifetime [days]	SG	NBP [°C]
R1336mzz(Z)	**Opteon™ MZ (Chemours)**	**171.3**	**29.0**	**0**	**2[a]**	**22[a]**	**A1**	**33.4**
R1234ze(Z)[b]	Not yet available	150.1	35.3	0	<1[a]	10[a], 18[b]	A2L	9.8
R1233zd(E)	**Solstice®zd (Honeywell)** Forane®HTS 1233zd (ARKEMA)	**165.6**	**35.7**	**0.00034[d], 0.00030[e]**	**1[a], <5[e]**	**~14[f],26[a], 36[e], 40.4[d]**	**A1**	**18.0**
R1224yd(Z)	**AMOLEA®1224yd (AGC Chemicals)**	**155.5**	**33.3**	**0.00023[c]**	**0.88[c]**	**20[c]**	**A1**	**14.0**
R365mfc	Solkane®365mfc (Solvay)	186.9	32.7	0	804[a]	8.7 years[a]	A2	40.2
R245fa	Genetron® 245fa (Honeywell)	154.0	36.5	0	858[a]	7.7 years[a]	B1	14.9





References:
T_{crit} and p_{crit} (EES F-Chart Software, V10.643, 2019), ODP basis R11=1.0 (UNEP, 2017), GWP_{100} (100-year time horizon, CO_2=1.0), SG: Safety group classification (ASHRAE 34, 2016), [a]Myhre et al. (2013, IPCC 5th assessment report), [b]Fukuda et al. (2014), [c]Tokuhashi et al. (2018), [d]Patten and Wuebbles (2010), [e]Sulbaek Andersen et al. (2018) (3D global model), [f]Andersen et al. (2015)

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Refrigerant properties

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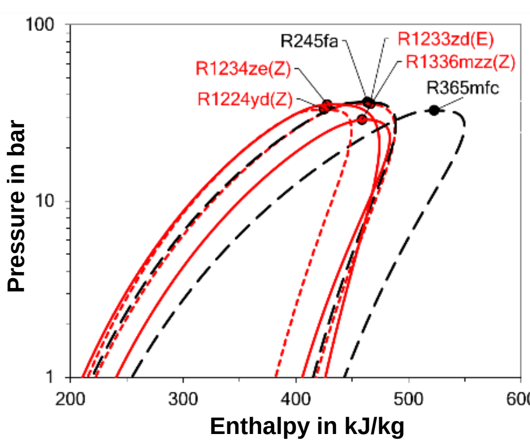
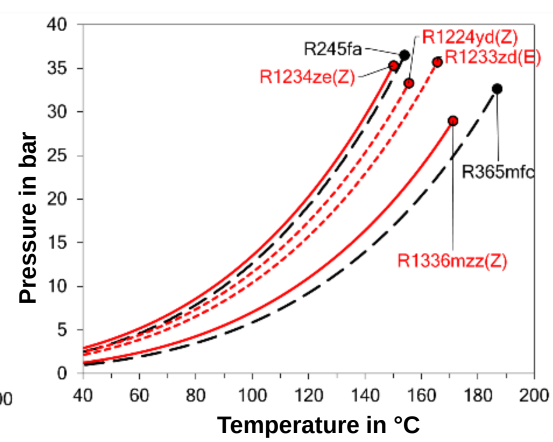
T-s, p-T diagrams of selected HFO and HCFO refrigerants





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Refrigerant efficiency simulation

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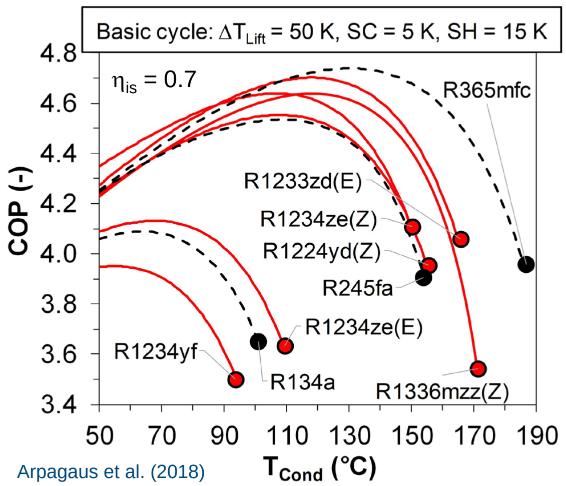
Simulated COP of selected HFO and HCFO refrigerants in a basic HP cycle



Arpagaus et al. (2018)

- Simulated COP rise to an optimum and decrease with the narrowing of the 2-phase region up to T_{crit}
- Optimal COP at about 30 K below the critical temperature
- R365mfc offers highest COP, followed by R1233zd(E) and R1336mzz(Z)
- R1234ze(Z) and R1224yd(Z) comparable to R245fa
- R1234yf and R1234ze(E) similar to R134a

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Discussion on TFA from HFCs and HFOs – environmental impact

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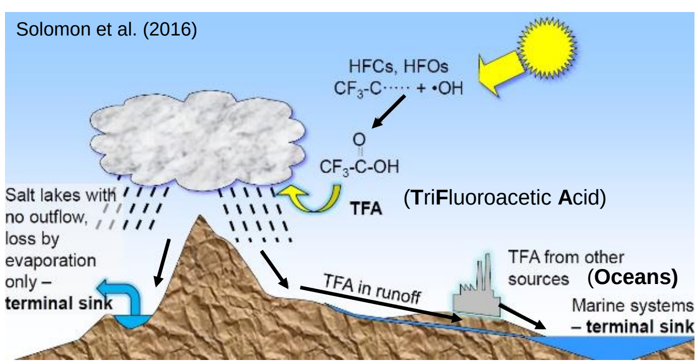
Environmental fate of TFA (trifluoroacetic acid, $CF_3C(O)OH$)



- **268 million tons TFA are present in the oceans, i.e. non-anthropogenic**
- 200 ng/L average TFA concentration in oceans (Frank et al., 2002)

Upper range scenario:

- **Total TFA yield from 1990 up to 2050 = 20.625 million tons TFA** (Solomon et al., 2016)

: 1.34 x 10^{21} L oceans water

Total additional contribution of TFA form HFCs and HFOs to the oceans is 15.3 ng/L* or <7.5% of the naturally present TFA

Negligible risk for aquatic organisms and human health

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Environmental impact of HFO and HCFO refrigerants suitable for HTHPs

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FHO Fachhochschule Ostschweiz

Atmospheric degradation products of HFOs and molar yields of TFA formation (Trifluoroacetic acid, $CF_3C(O)OH$)

	Refrigerant	Formula	Final degradation products	Molar yields of TFA $CF_3C(O)OH$
HFO	R1234yf	CF_3-CF=CH_2	$CF_3C(O)OH$, CO_2, HF	100%
	R1234ze(E)	E-CF_3-CH=CHF	CO_2, HC(O)OH, HF	<10%, 0%
	R1336mzz(Z)	**Z-CF_3-CH=CHCF_3**	**CO_2, HF**	**<20%[a]**
HCFO	**R1233zd(E)**	**E-CF_3-CH=CHCl**	**CO_2, HF, HCl**	**~ 2%[b]**
	R1224yd(Z)	**Z-CF_3-CF=CHCl**	***similar structure like R1234yf degrading to $CF_3C(O)F$ and hydrolyzing to TFA***	
HFC	R365mfc	CF_3-CH_2-CF_2-CH_3	CO_2, HF	<10%
	R245fa	CHF_2-CH_2-CF_3	CO_2, HF	<10%

Products:

$CF_3C(O)OH$	trifluoroacetic acid (TFA)
HC(O)OH	formic acid
CO_2	carbon dioxide
HCl	hydrochloric acid
HF	hydrofluoric acid

References:
Norwegian Environment Agency (2017), WMO (2018), Wallington et al. (2014), Juhasz & Kontomaris (2018), EFCTC (2019), [a]**Henne et. al. (2012),** [b]**Sulbaek Andersen et al. (2008, 2012, 2018)**, Inoue et al. (2008), ECETOC (2004), Chen et al. (1997)

TFA formation yield depends on HFO refrigerant

Risk of TFA formation for R1336mzz(Z) and R1233zd(E) is considered to be close to negligible

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Objectives

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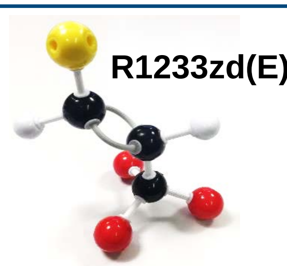
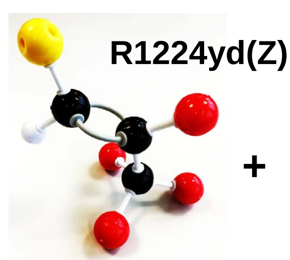
Goals of this study

Objectives:

- **Performance evaluation** of **R1336mzz(Z)** (Opteon™MZ, Chemours), **R1233zd(E)** (Solstice®zd, Honeywell), and **R1224yd(Z)** (AMOLEA®1224yd, AGC Chemicals) in a laboratory HTHP (**drop-in test**).





+ POE oil (SE 170)

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System design of the laboratory scale HTHP at NTB Buchs

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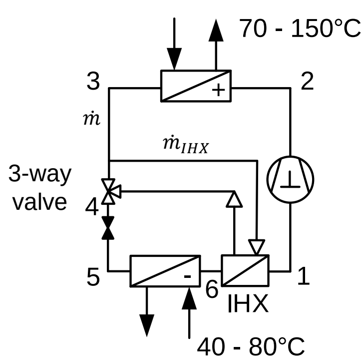
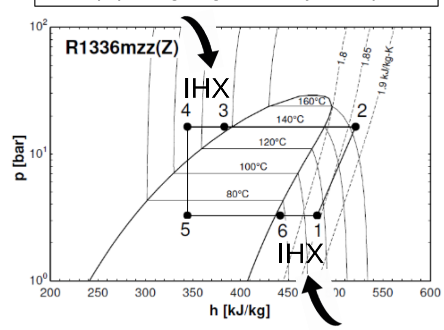
System design – laboratory scale HTHP at NTB Buchs

1-stage cycle with internal heat exchanger (IHX) and adjustable 3-way valve



$IHX: \dot{m}_{IHX}/\dot{m} = 0\% \rightarrow 100\%$
(Opening degree 3-way-valve)



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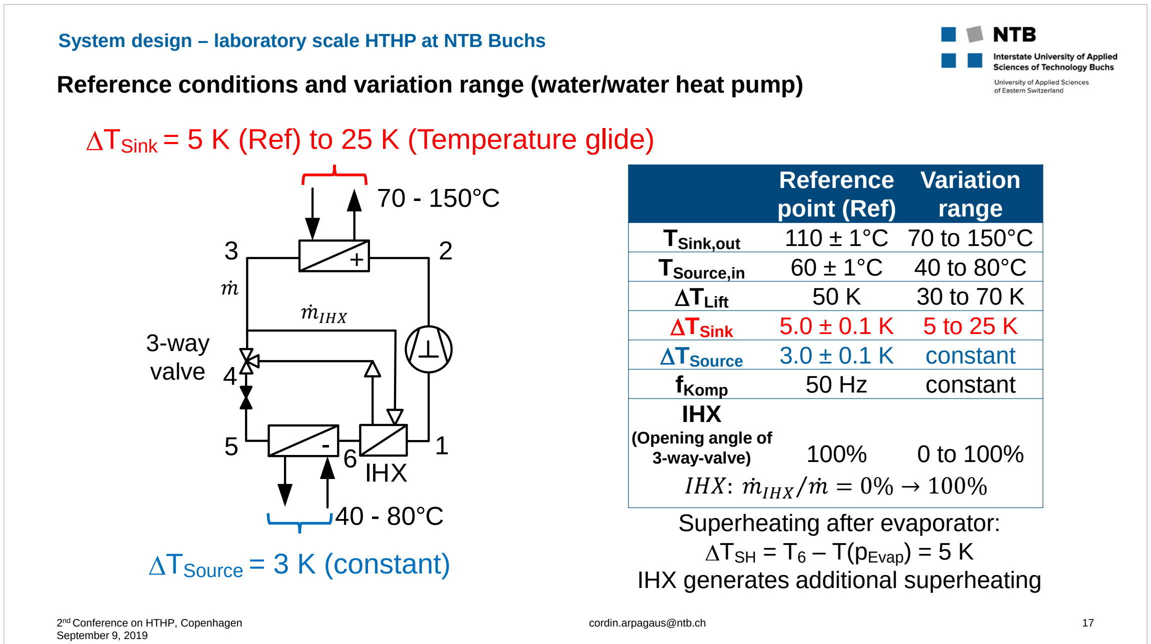
System design – laboratory scale HTHP at NTB Buchs
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Reference conditions and variation range (water/water heat pump)
ΔT Sink = 5 K (Ref) to 25 K (Temperature glide)
70 - 150°C
3
2
ṁ
ṁ IHX
3-way valve
4
5
6
1
IHX
40 - 80°C
ΔT Source = 3 K (constant)
Reference point (Ref) | Variation range
T Sink,out | 110 ± 1°C | 70 to 150°C
T Source,in | 60 ± 1°C | 40 to 80°C
ΔT Lift | 50 K | 30 to 70 K
ΔT Sink | 5.0 ± 0.1 K | 5 to 25 K
ΔT Source | 3.0 ± 0.1 K | constant
f Komp | 50 Hz | constant
IHX (Opening angle of 3-way-valve) | 100% | 0 to 100%
IHX: ṁ IHX / ṁ = 0% → 100%
Superheating after evaporator:
ΔT SH = T6 – T(p Evap) = 5 K
IHX generates additional superheating
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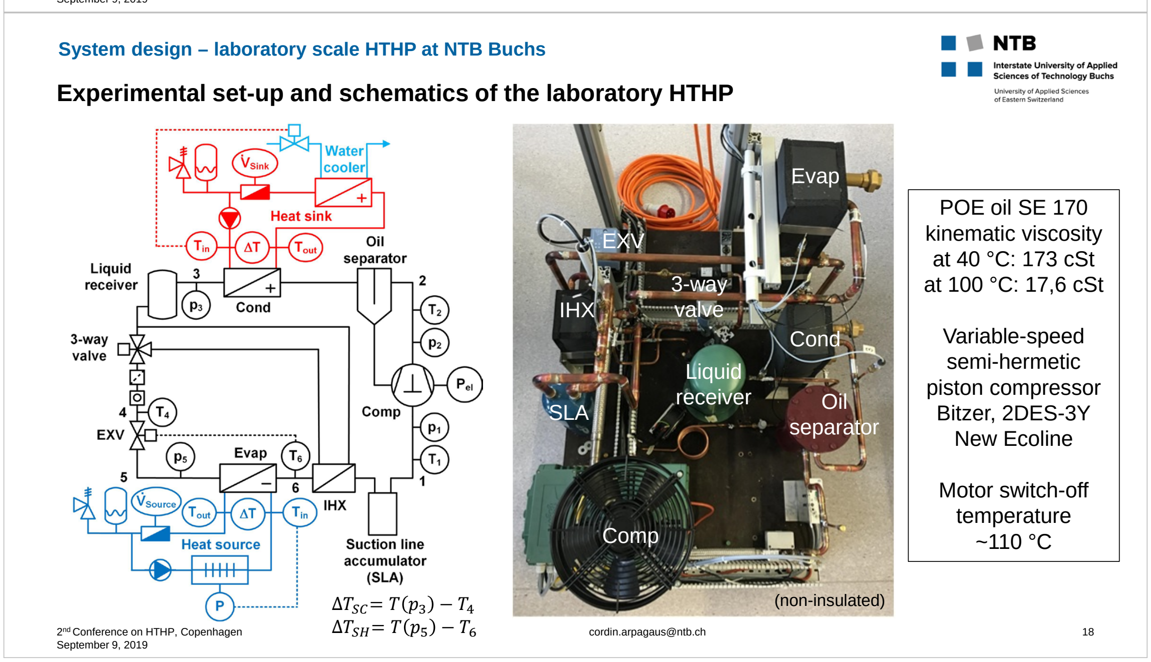
System design – laboratory scale HTHP at NTB Buchs
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Experimental set-up and schematics of the laboratory HTHP
Water cooler
V Sink
Heat sink
T in
ΔT
T out
Oil separator
Liquid receiver
3
2
Cond
p3
T2
p2
3-way valve
P el
Comp
4
T4
EXV
p1
p5
Evap
T6
T1
5
6
1
IHX
V Source
T out
ΔT
T in
Heat source
P
Suction line accumulator (SLA)
ΔT SC = T(p3) – T4
ΔT SH = T(p5) – T6
Evap
EXV
3-way valve
IHX
Cond
Liquid receiver
Oil separator
SLA
Comp
(non-insulated)
POE oil SE 170
kinematic viscosity
at 40 °C: 173 cSt
at 100 °C: 17,6 cSt
Variable-speed semi-hermetic piston compressor Bitzer, 2DES-3Y New Ecoline
Motor switch-off temperature ~110 °C
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System design – laboratory scale HTHP at NTB Buchs

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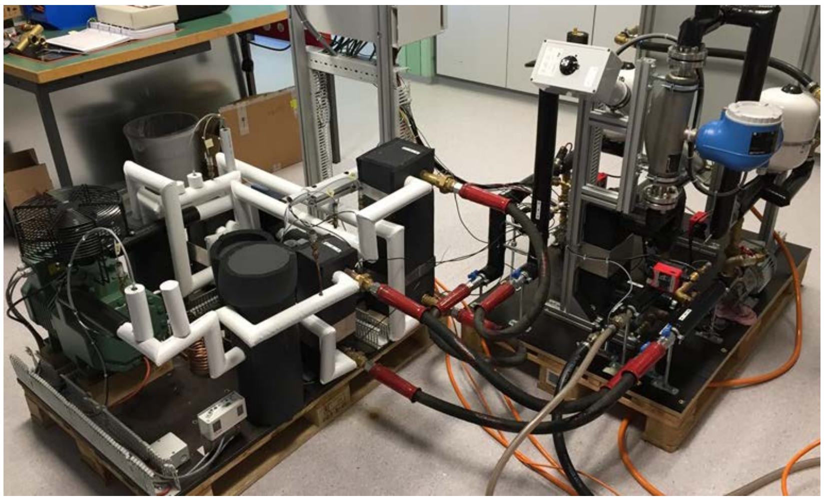
Laboratory HTHP with hydraulic loops for heat source and sink



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System design – laboratory scale HTHP at NTB Buchs

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Sensors and measurement uncertainties

Measured parameters		Sensor type	Uncertainities
Pressures	$p_{1...6}$	Piezoelectric, 0 to 50 bar, max. 120°C	max. 1.5% of full scale reading
Temperatures	$T_{1...6}$	Thermocouples, type K, class 1	± 1.5 K
Heat sink temp difference	ΔT_{Sink}	Thermocouples, type K, class 1	± 0.1 K
Compressor power	P_{Comp}	Power transmitter, 0 to 15 kW	0.2 % of measuring range + 0.1 % measured value
Heat sink mass flow (water)	$\dot{m}_{H_2O}$	Coriolis, 0 to 1'300 kg/h, max. 180 °C	± 0.05 %

$$COP = \frac{\dot{Q}_{Sink}}{P_{Comp}} = \frac{\dot{m}_{H_2O} \cdot c_{p,H_2O}(T) \cdot \Delta T_{Sink}}{P_{Comp}}$$

$$COP_{Carnot} = \frac{T_{Sink,out}}{T_{Sink,out} - T_{Source,in}}$$

2nd Law efficiency:

$$\eta_{2nd} = \frac{COP_H}{COP_{Carnot}}$$

Error propagation according to RSS method (Root Sum Squares):

$$\Delta COP = \sqrt{\left(\frac{\partial COP}{\partial \dot{m}_{H_2O}} \cdot \Delta \dot{m}_{H_2O}\right)^2 + \left(\underbrace{\frac{\partial COP}{\partial c_{p,H_2O}(T)} \cdot \Delta c_{p,H_2O}(T)}_{\sim 0}\right)^2 + \left(\frac{\partial COP}{\partial \Delta T_{Sink}} \cdot \Delta(\Delta T_{Sink})\right)^2 + \left(\frac{\partial COP}{\partial P_{Comp}} \cdot \Delta P_{Comp}\right)^2}$$

Average uncertainty	R1336mzz(Z)	R1233zd(E)
ΔCOP	± 0.21 (4.2%)	± 0.21 (4.1%)
$\Delta \dot{Q}_{Sink}$	± 0.14 kW (3.7%)	± 0.22 kW (3.8%)
ΔP_{Comp}	± 0.031 kW (2.6%)	± 0.032 kW (1.7%)

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Test procedure

ΔT_{Sink}	5.0 ± 0.1 K
ΔT_{Source}	3.0 ± 0.1 K

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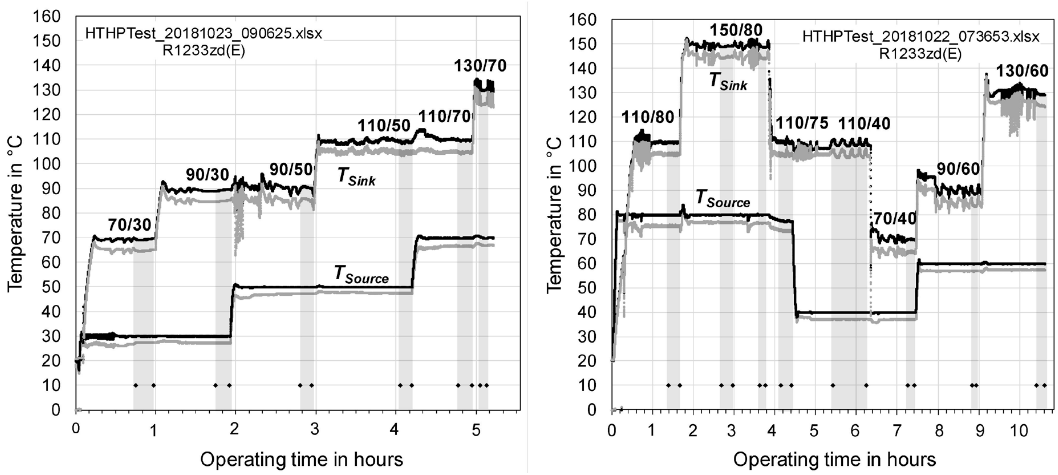
Temperature profiles of experimental runs (at least 5 min stable conditions)



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Experimental results with R1336mzz(Z), R1233zd(E) and R1224yd(Z)

Experimental results – influence of insulation

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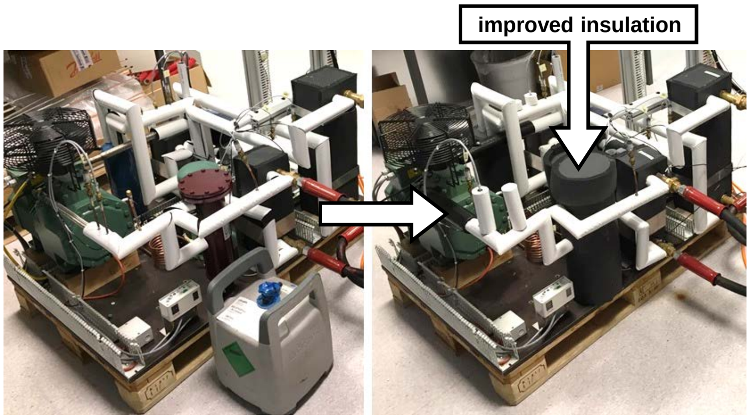
COP improvement with better insulation of oil separator, liquid receiver, and suction line accumulator with Armaflex®HT insulation



Temperature resistance: up to 150°C
Thermal conductivity (0°C): 0.038 W/m

COP improvement Δ with better insulation

$T_{Source,in}$ / $T_{Sink,out}$ (ΔT_{Lift})	COP (before[1)])	COP (after[2)])	Δ
40/90 (50)	2.58	3.14	+22%
60/110 (50)	2.78	3.09	+11%
80/130 (50)	2.67	3.10	+16%

(1-stage cycle with 100% IHX)

[1)] Arpagaus et al. (2018), 17th Int. Refrig. Air Cond. Conf., Purdue, July 9-12, 2018.
[2)] Arpagaus et al. (2018), DKV-Tagung 2018, Aachen, November 21-23, 2018.

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Experimental results – influence of insulation

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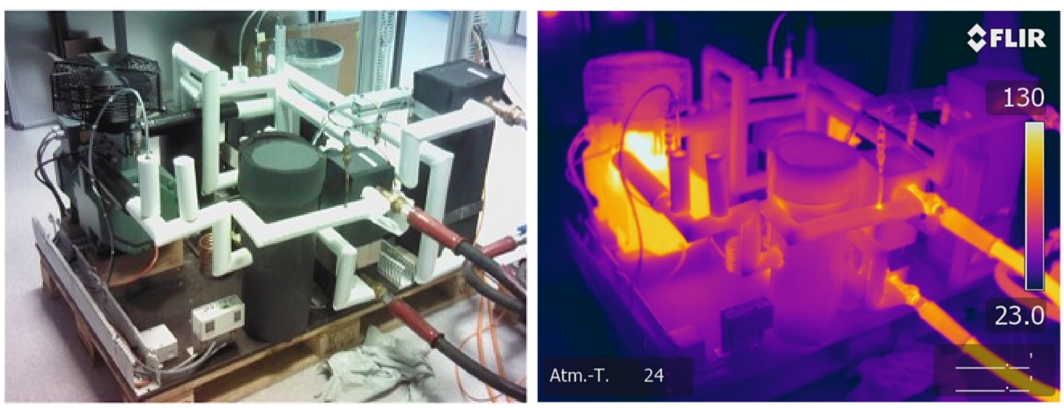
Infrared camera image for hot spot identification



- Heat losses of about 21 ± 7 % estimated from energy balance (major heat losses at the compressor)
- There is still potential for optimization in insulation and possibilities for increasing efficiency

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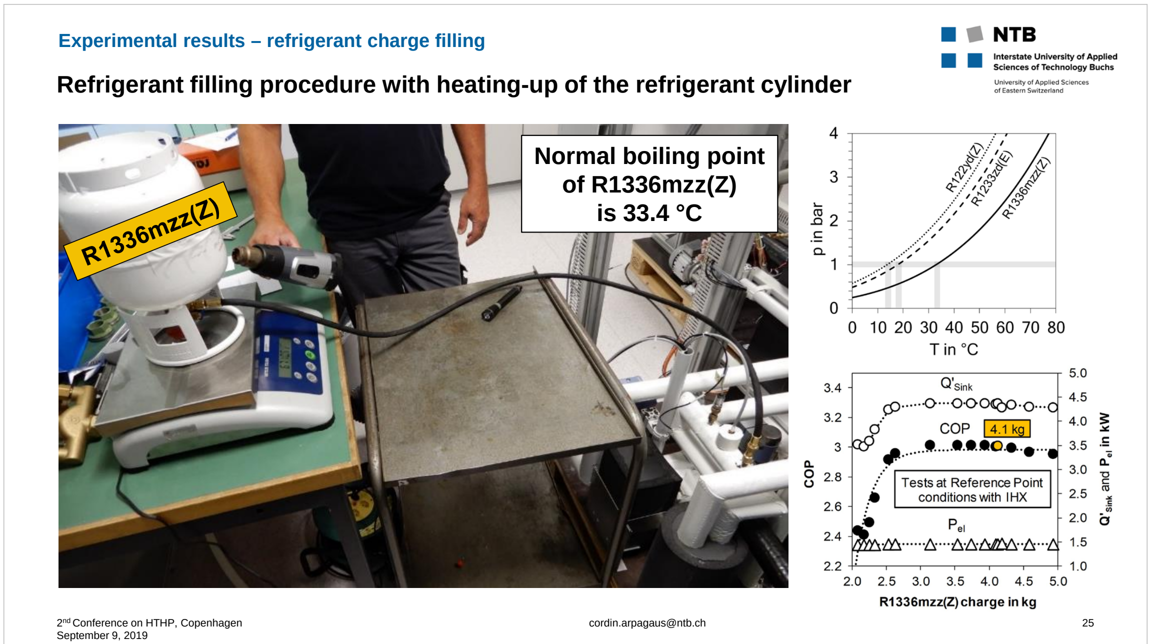
Experimental results – refrigerant charge filling
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Refrigerant filling procedure with heating-up of the refrigerant cylinder
R1336mzz(Z)
Normal boiling point of R1336mzz(Z) is 33.4 °C
p in bar
T in °C
R1224yd(Z)
R1233zd(E)
R1336mzz(Z)
Q'Sink
COP
4.1 kg
Tests at Reference Point conditions with IHX
Pel
COP
Q'Sink and Pel in kW
R1336mzz(Z) charge in kg
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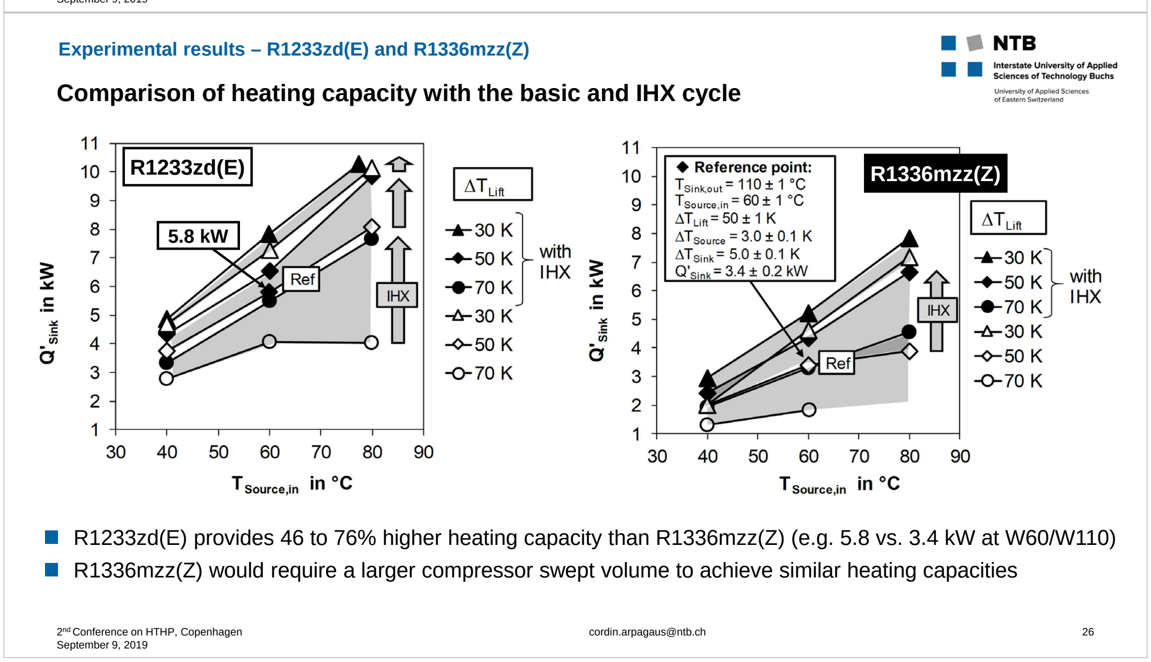
Experimental results – R1233zd(E) and R1336mzz(Z)
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Comparison of heating capacity with the basic and IHX cycle
R1233zd(E)
5.8 kW
Ref
IHX
ΔTLift
30 K
50 K
70 K
with IHX
Q'Sink in kW
TSource,in in °C
Reference point:
TSink,out = 110 ± 1 °C
TSource,in = 60 ± 1 °C
ΔTLift = 50 ± 1 K
ΔTSource = 3.0 ± 0.1 K
ΔTSink = 5.0 ± 0.1 K
Q'Sink = 3.4 ± 0.2 kW
R1336mzz(Z)
R1233zd(E) provides 46 to 76% higher heating capacity than R1336mzz(Z) (e.g. 5.8 vs. 3.4 kW at W60/W110)
R1336mzz(Z) would require a larger compressor swept volume to achieve similar heating capacities
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Experimental results – R1233zd(E) and R1336mzz(Z)

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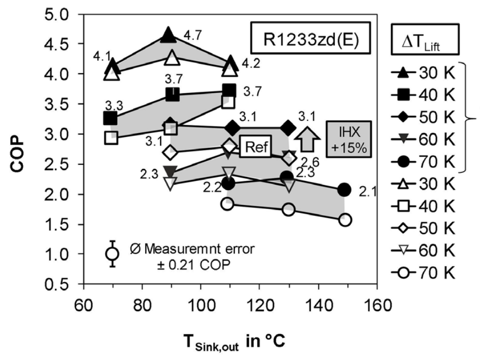
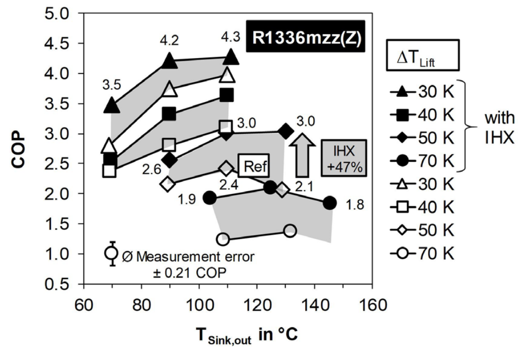
Operating maps of efficiency with basic and IHX cycle

COP vs. $T_{Sink,out}$ at different temperature lifts ($\Delta T_{Lift} = T_{Sink,out} - T_{Source,in}$) from 30 to 70 K





- As expected, COP increase with smaller ΔT_{Lift} and higher $T_{Sink,out}$ according to Carnot relationship
- IHX integration provides significant COP increase (superheat achieved in IHX, higher evaporation temperature)
- Max. heat sink temperature tested was 150 °C with a COP of 2.1 for R1233zd(E) and 1.8 for R1336mzz(Z)

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Experimental results – R1224yd(Z)

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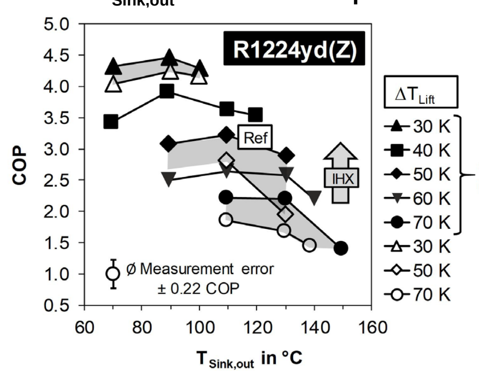
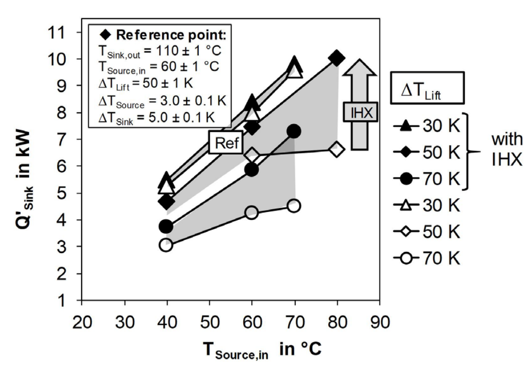
Operating maps of efficiency with basic and IHX cycle

COP vs. $T_{Sink,out}$ at different temperature lifts ($\Delta T_{Lift} = T_{Sink,out} - T_{Source,in}$) from 30 to 70 K





- IHX increases the COP about 14% compared to a basic cycle at Ref conditions
- COP of 3.2 reached at W60/W110 (50 K temperature lift, with IHX)
- 10 kW heating capacity reached at W80/W130 with IHX cycle

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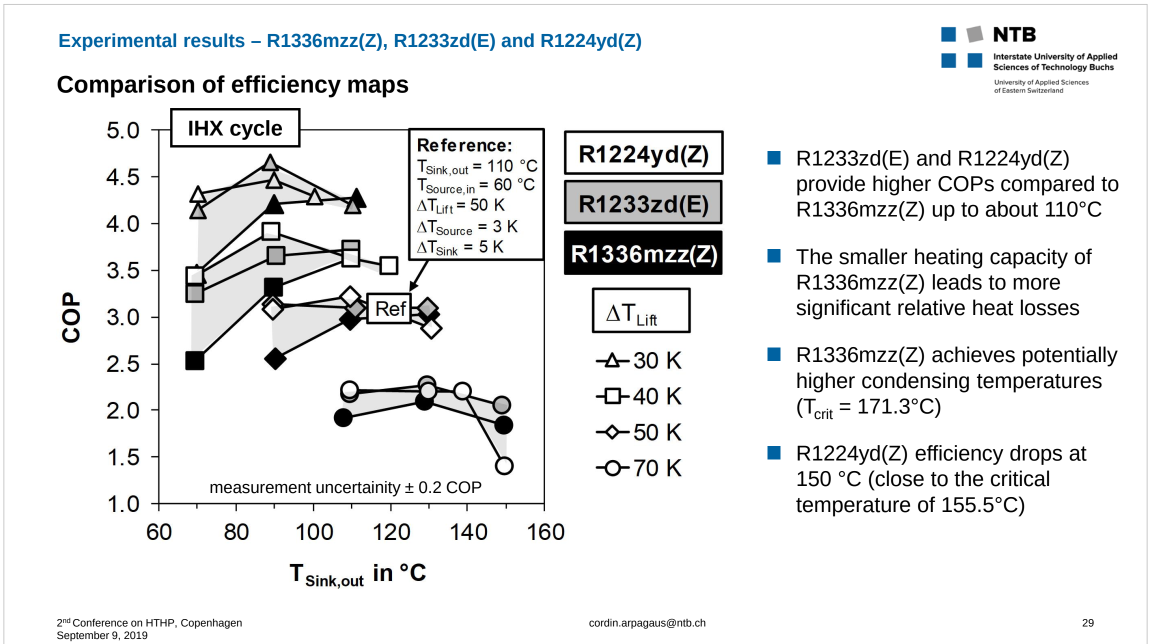
Experimental results – R1336mzz(Z), R1233zd(E) and R1224yd(Z)
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Comparison of efficiency maps
IHX cycle
Reference:
T Sink,out = 110 °C
T Source,in = 60 °C
ΔT Lift = 50 K
ΔT Source = 3 K
ΔT Sink = 5 K
Ref
measurement uncertainity ± 0.2 COP
COP
T Sink,out in °C
R1224yd(Z)
R1233zd(E)
R1336mzz(Z)
ΔT Lift
30 K
40 K
50 K
70 K
R1233zd(E) and R1224yd(Z) provide higher COPs compared to R1336mzz(Z) up to about 110°C
The smaller heating capacity of R1336mzz(Z) leads to more significant relative heat losses
R1336mzz(Z) achieves potentially higher condensing temperatures (T crit = 171.3°C)
R1224yd(Z) efficiency drops at 150 °C (close to the critical temperature of 155.5°C)
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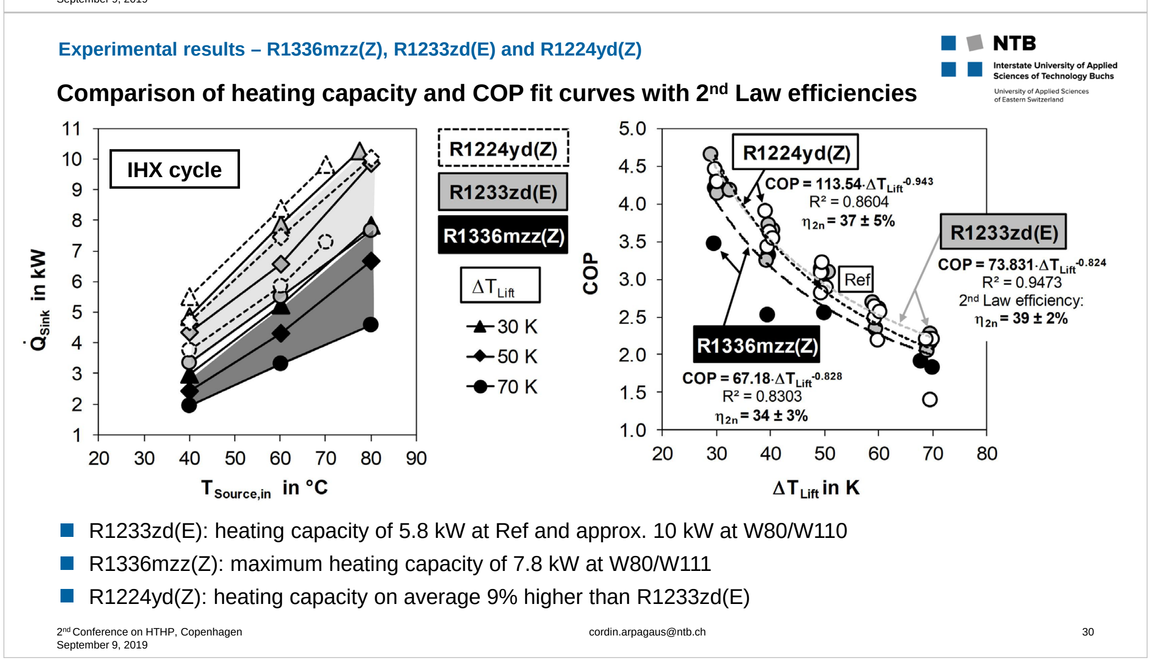
Experimental results – R1336mzz(Z), R1233zd(E) and R1224yd(Z)
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Comparison of heating capacity and COP fit curves with 2nd Law efficiencies
IHX cycle
Q Sink in kW
T Source,in in °C
R1224yd(Z)
R1233zd(E)
R1336mzz(Z)
ΔT Lift
30 K
50 K
70 K
COP
COP = 113.54·ΔT Lift^-0.943
R² = 0.8604
η2n = 37 ± 5%
Ref
COP = 73.831·ΔT Lift^-0.824
R² = 0.9473
2nd Law efficiency:
η2n = 39 ± 2%
COP = 67.18·ΔT Lift^-0.828
R² = 0.8303
η2n = 34 ± 3%
ΔT Lift in K
R1233zd(E): heating capacity of 5.8 kW at Ref and approx. 10 kW at W80/W110
R1336mzz(Z): maximum heating capacity of 7.8 kW at W80/W111
R1224yd(Z): heating capacity on average 9% higher than R1233zd(E)
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Experimental results – R1336mzz(Z)

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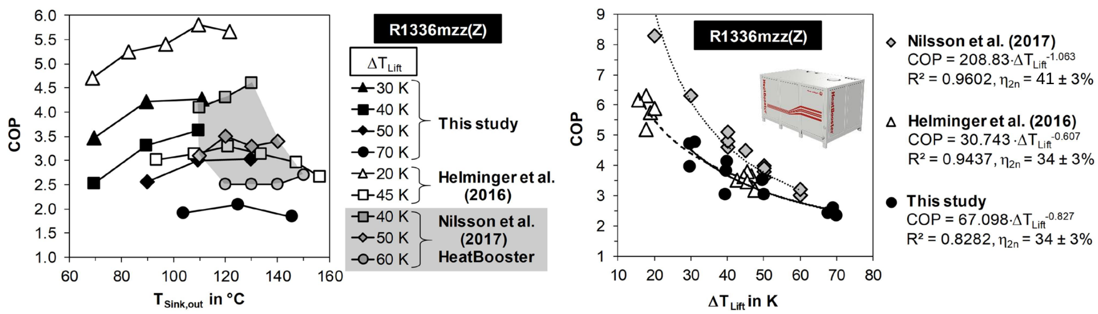
Comparison of COP data from Helminger et al. (2016) and Nilsson et al. (2017)



- Helminger et al. (2016):
  - 156.3°C and COP of almost 2.7 at 45 K ΔT_{Lift}
  - 2nd Law efficiency of 34 ± 3% similar to this study
- Nilsson et al. (2017):
  - Commercial HeatBooster technology achieves higher COP and 2nd Law efficiency of 41 ± 3 %
  - COP of 2.5 at 60 K ΔT_{lift}
  - With larger heating power the relative heat losses are getting smaller

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Experimental results – R1336mzz(Z)

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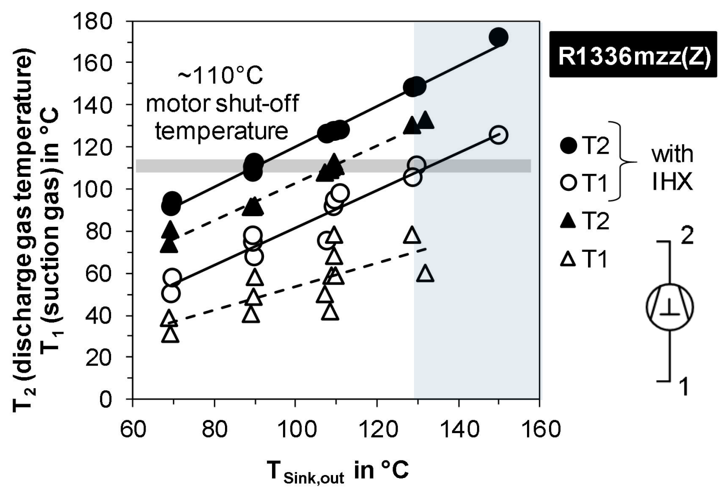
Discharge and suction gas temperatures for tests with R1336mzz(Z)



Basic cycle:

- Suction temperature (Δ T1) well below the motor shut-off temperature of approx. 110 °C (grey line)

Cycle with IHX:

- Suction temperature (O T1) exceeded the motor limit temperature at a heat sink outlet temperature of about 130 °C and higher.

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Oil degradation measurement - neutralization number (acid number)

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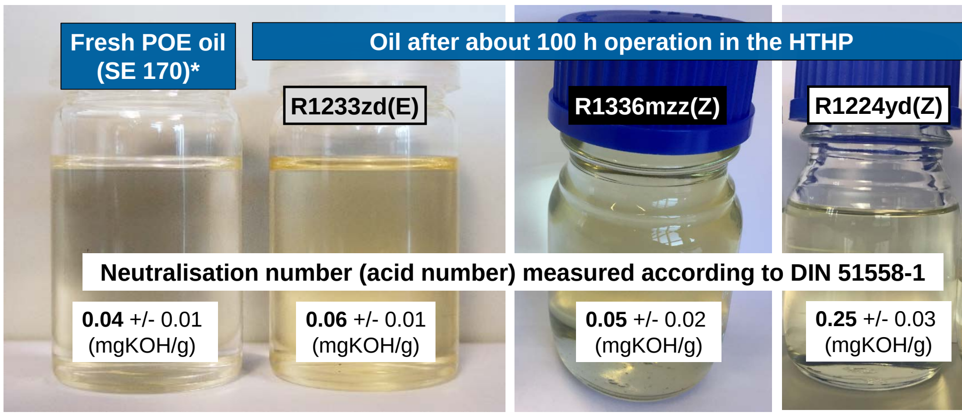
Negligible oil degradation after about 100 h operation in the HTHP



< 0.5 warning value

* 0.03 mgKOH/g according to product information from FUCHS

FUCHS RENISO TRITON SE 170 – synthetic oil based on polyolester (POE) suitable for HFO refrigerants (complete miscibility with R1233zd(E) and R1336mzz(Z) between +100°C and -40°C)

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Experimental results of HFO/HCFO refrigerants in a HTHP with up to 150 °C supply temperature

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Conclusions

- **R1336mzz(Z), R1233zd(E) and R1224yd(Z) successfully tested in single -stage lab-scale HTHP with IHX cycle and up to 10 kW**
- **Operation of demonstrated** at **30 to 80°C heat source** and **70 to 150°C heat sink temperatures** (30 to 70 K temperature lifts) for possible application of **waste heat recovery, steam generation or drying**
- At **W60/W110 COPs of 3.2, 3.1 and 3.0** for R1224yd(Z), R1233zd(E) and R1336mzz(Z) were measured
- **Up to about 110 °C, R1224yd(Z) and R1233zd(E) slightly higher COP than R1336mzz(Z)** due to higher heating capacities and lower relative heat losses at the same temperature conditions
- **At 150 °C R1233zd(E) and R1336mzz(Z) more efficient** than R1224yd(Z) due to higher critical tempeatures
- **Integration of an IHX increased COP** (+15 to 47%) and heating capacity **significantly**
- **Negligible oil degradation after about 100 h operation in HTHP** (acid numbers < 0.5 mgKOH/g level)
- **Very low GWP, non-flammability, and negligible environmental impact** (low TFA formation during atmospheric degradation) indicate a **high potential for future use as refrigerant in HTHP** applications and retrofit systems

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Thanks

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Thank you for your attention

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Supply of high Temperature Heat and Cooling with MAN ETES HP

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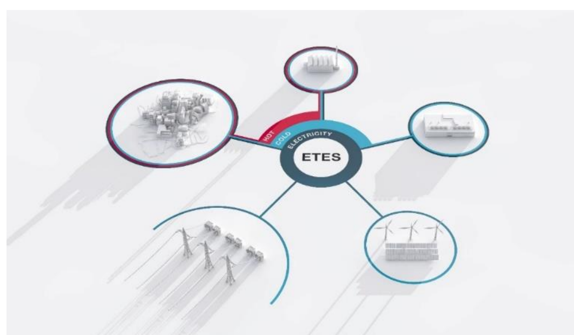
Keywords:

High temperature heat pump, R744, CO2, compressor technology, Heat, Cooling, Storage, Re-electrification

Abstract

There is a compelling need to increase utility-scale energy supply capacity in response to the dramatic growth in intermittent renewable generating capacity like wind and solar. Now though, new technologies have emerged that offers bulk power supply and storage at scale in hundreds of MWh. Perhaps even more significantly, Electro-Thermal Energy Storage (ETES) connects heating, cooling and electricity storage together. As a result, the system can meet multiple energy storage and supply needs simultaneously and thus help to achieve the climate goals until 2030.

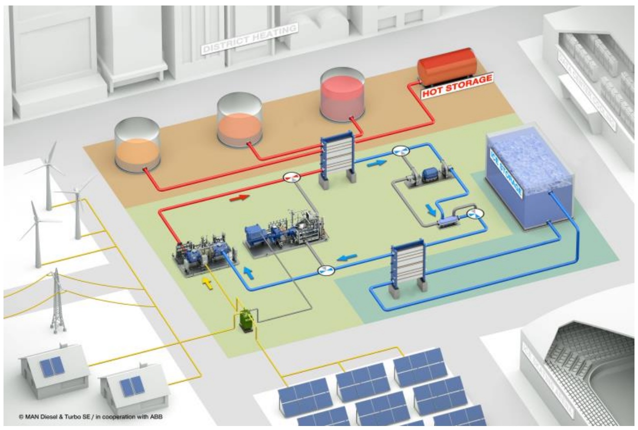


Based on a novel and reversible thermodynamic cycle, ETES is a scalable and efficient technology that supports sector coupling between the distinct energy needs of heating, cooling and electricity. With ETES, heating needed for food processing and district heating can meet cooling for applications like data centres, warehousing and large commercial buildings, as well as electricity storage capabilities to support grid balancing and renewable energy optimisation – all in a single system.

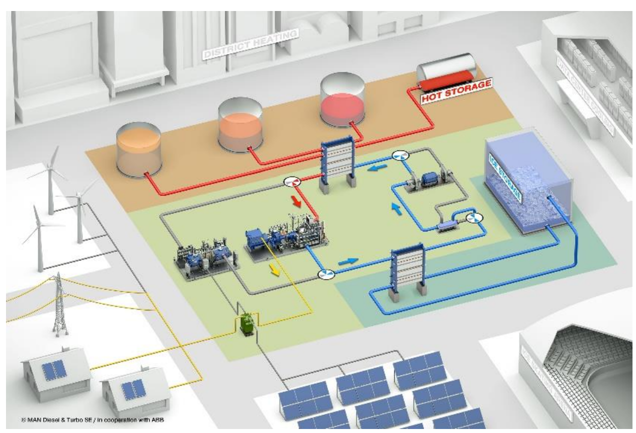
By allowing industrial, commercial and domestic sectors to combine their needs, MAN's ETES offers a comprehensive and efficient solution to a host of energy system challenges while keeping capital and operational expenditures to a minimum.

Currently the only available solution capable of using, storing and distributing heat, cold and electricity simultaneously, the patented trigeneration energy-management system is based on the use of CO_2

(R744) as working fluid. At its core ETES allows the conversion of electrical energy into thermal energy in the form of hot water and ice and vice versa. The energy is supplied directly to consumers or stored in a series of thermally insulated water tanks, making the system low risk and very robust with high resilience. Similar to a domestic refrigeration unit, in ETES the closed CO_2 cycle sees the working fluid compressed or expanded through turbomachinery to store or extract energy. Depending on specific demands, energy stored as either heat or cold may be directly distributed or efficiently reconverted back to electrical energy as required.



During the charging or better called heat pump cycle, electrical energy from any source - such as renewable energy - is used to power a MAN HOFIM™ turbocompressor. The CO_2 working fluid is compressed to supercritical conditions at 140 bar approximately and from 120°C typically up to 150°C by a single turbocompressor oil-free unit. Passing through a heat exchanger, heat from the compressed CO_2 is transferred to the hot storage tanks. There may be as many as four such tanks, for example three at atmospheric and one pressurized, depending on the temperature level demand and final application. Downstream the heat exchanger, the cooler but still pressurized CO_2 then passes into a turbo-expander where the CO2 in liquid phase drops in pressure, thus energy won back for own plant usage, and CO2 chilled to sub-zero temperature. At this stage a second set of heat exchangers (used as evaporators) chills the cold storage tank to produce ice.



In the reverse process, gaseous cold side CO_2 passes through the same set of heat exchangers (used as condensators). CO2 is fully condensed to liquid while the temperature on the cold tank side is increased and ice melt. The now liquid CO_2 is pressurized through a pump to supercritical still cold conditions and circulates through the hot side heat exchangers. The heat from the hot storage tanks is transferred back to the working fluid. Now the heated and pressurised CO_2 passes through an expansion turbine where a coupled generator is used to produce electricity as required.

Using hot water in simple insulated tanks require minimal insulation, analogous for the ice, along with moderate operating pressure standard turbomachinery equipment means that the system has a low environmental impact on one side and is reliant on well-proven and extensively deployed systems on the other side.

Among the core components of the ETES system is MAN's hermetically sealed HOFIM™ turbocompressor. Built for rugged extremes and used, for example, in subsea compression station applications, these units are multi-stage radial compressors. With casings designed for 220 bar, HOFIM™ compressors feature a 7-axes active magnetic bearing system and are arranged in a single shaft configuration together with a high-speed electric motor. The turbocompressor has no oil or sealing systems which reduces complexity of auxiliaries. Compared with traditional compressor designs HOFIM™ designs have a 60% smaller footprint and 30% less mass. The HOFIM™ compressor family, designed for subsea applications where reliability and service longevity are paramount, is currently available in a power range of 4 to 15 MWe input power.

Given that HOFIM™ compressors typically serve the hydrocarbon industry and challenging materials such as refined products, the use of inert, non-corrosive and non-abrasive CO_2 within ETES keeps operations and maintenance costs low. The compressors run in a pristine atmosphere with minimal contamination and major service intervals are estimated at 10 years or more. In addition, the peak process conditions for high temperature Heat Pump applications at roughly 180 bar and 150°C are well within the compressor's performance capabilities.

The modular and easy scalable ETES heat pump system is the main sub-system of the ETES complete process, namely the charging or better called heat pump cycle of the overall process. It is composed by the so-called "baseline" configuration comprising the core CO2 cycle and key components (turbomachinery and heat exchanger mainly). The baseline design allows a broad flexibility in the power size and temperature level, basically around 0°C on the cold sink side to max. 150°C on the hot sink side.

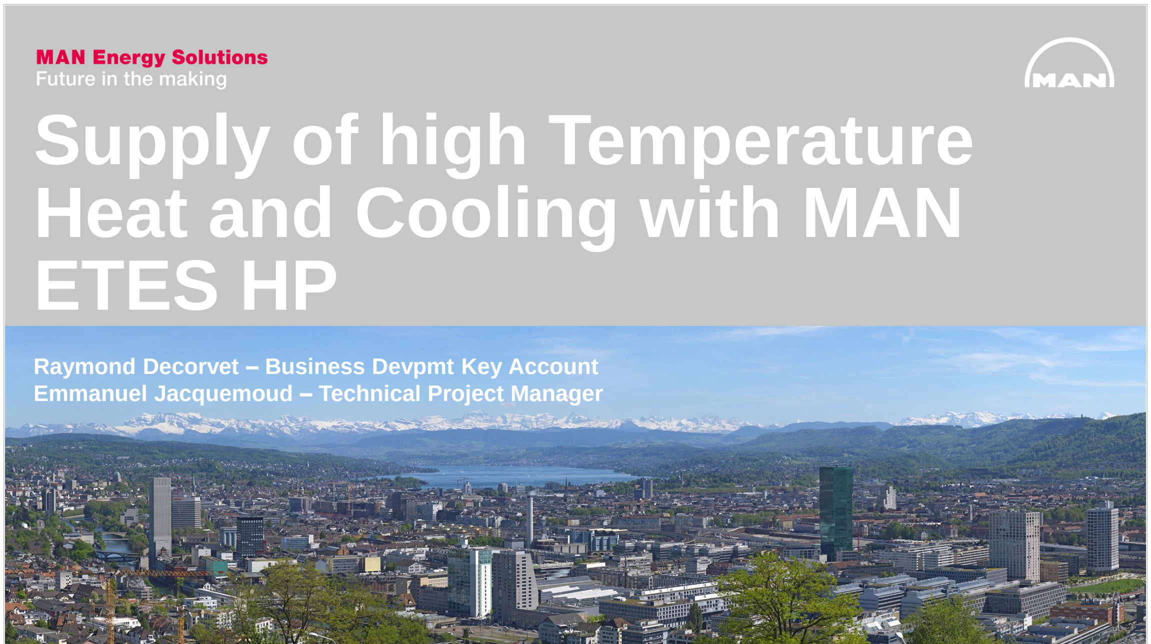
The optional integration of storage reservoirs, i.e. sensible hot water tanks on the hot side as well as latent cold water/ice storage on the cold sink side increases the design flexibility of the system with fully modular and customizable capacities for both hot and cold storage reservoirs. The operation of such a system can therefore play with flexible charging times of the system and the related power supply in hot and cold heat export on the demand side.

As a modular and easily scalable energy supply and management solution for mid- to large-scale thermal and electrical consumers, one of the primary markets for this technology are the industrial and municipal consumers.

A unique feature of the system are the multiple tanks operating at different temperatures. Not only does this maximise the cycle efficiency, it is also well suited to many process industries. With hot side temperatures ranging from say 40°C to 150°C and cold down to 0°C, a huge range of applications are possible: district heating, sterilisation, pasteurization on the hot demand side, comfort cooling of large buildings, food process industries to cooling supply for the hoggish energy consuming data centres for instance, among plenty of other possible applications.

Although significant progress has been made in greening the global electricity supply system, reaching our pressing carbon emissions goals requires the same measures of success to be achieved in the heating, cooling and transport sectors too. Heating and cooling alone account for around half of all global energy consumption and it is a sector where renewable energies have so far failed to make a significant impact. Scalable technologies such as MAN ETES, based on well-proven, long-lived and reliable industrial equipment, represents a realistic opportunity to change that fact.

Ideally suited to sector coupling, MAN ETES brings together the energy supply sector to deliver different consumer demands while increasing energy efficiency, maximising the renewable energy contribution and providing grid stability functionality. It achieves the goal of increasing the renewable energy contribution to the heating, cooling and power sectors in a way that is not viable when the energy system is considered as discrete individual silos. Changing how we think about energy coupling means that meeting all the diverse heating, cooling and electricity demands of a large modern city and using only variable renewables is now possible with technologies like MAN ETES.


MAN Energy Solutions
Future in the making
Supply of high Temperature Heat and Cooling with MAN ETES HP
Raymond Decorvet – Business Devpmt Key Account
Emmanuel Jacquemoud – Technical Project Manager


MAN Energy Solutions - Our Vision
We are investing all of our energy into creating solutions for sustainable prosperity. That's why we have changed our name from MAN Diesel & Turbo to MAN Energy Solutions.
We are a preferred employer, and the partner of choice for our customers in the marine, energy, and industrial sectors.
Building on our unique range of capabilities, we create pioneering solutions to master the business, technical, and operational challenges of decarbonization.
We enable customers to achieve sustainable value creation in the transition towards a carbon neutral future.
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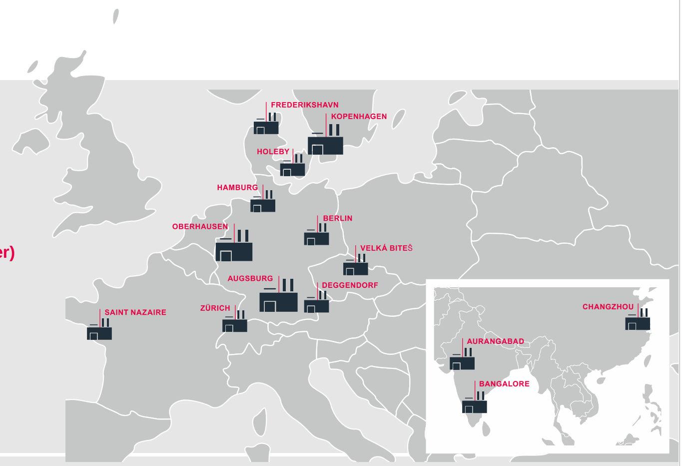
MAN Energy Solutions @ a Glance

Key Figures

11	Production sites in Europe
3	Production sites in Asia
30	Licensees in 7 countries (two- and 4-stroke, turbocharger)
+14'000	Employees worldwide
3.1bn €	Revenue 2018
HQ	Augsburg / Germany



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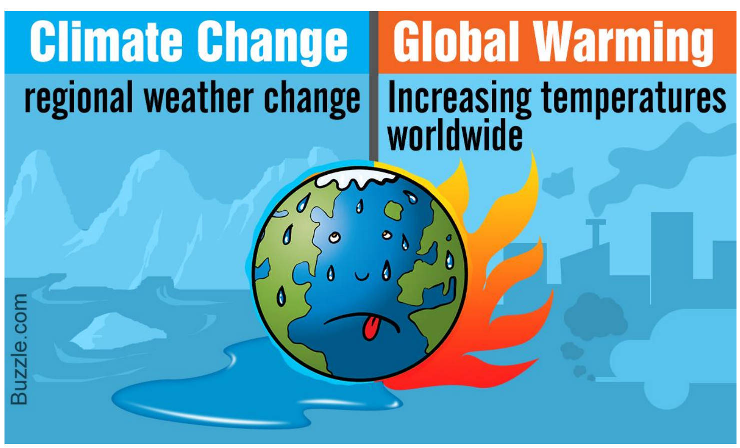
Decarbonization is the only option !

Climate Change & Global Warming is real !



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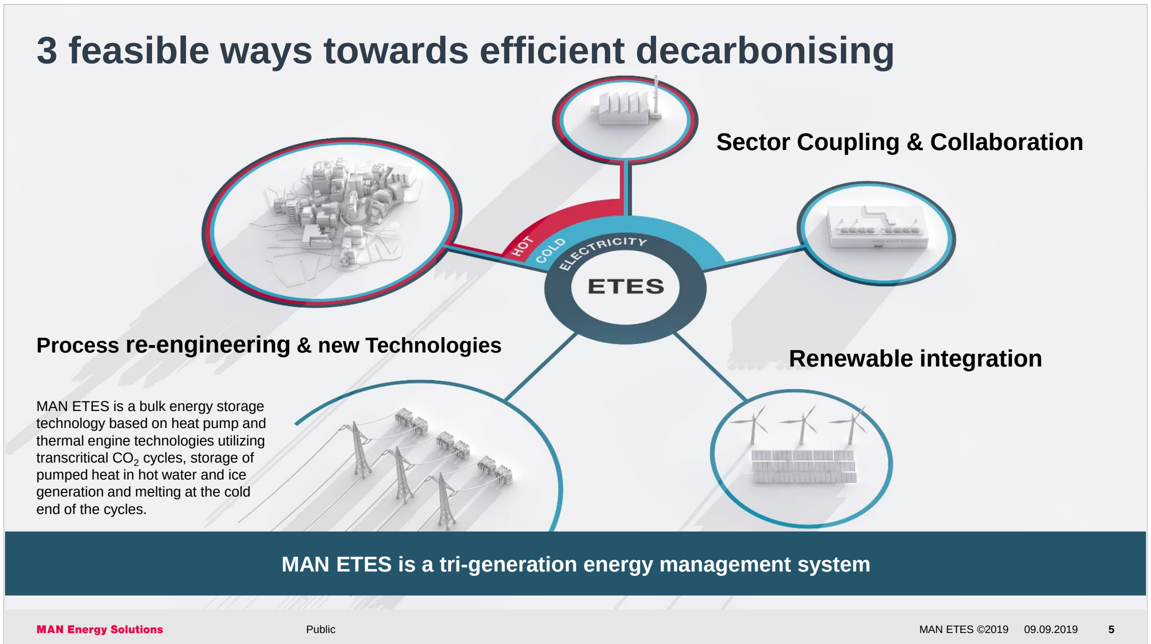
3 feasible ways towards efficient decarbonising
Sector Coupling & Collaboration
HOT
COLD
ELECTRICITY
ETES
Process re-engineering & new Technologies
Renewable integration
MAN ETES is a bulk energy storage technology based on heat pump and thermal engine technologies utilizing transcritical CO_2 cycles, storage of pumped heat in hot water and ice generation and melting at the cold end of the cycles.
MAN ETES is a tri-generation energy management system
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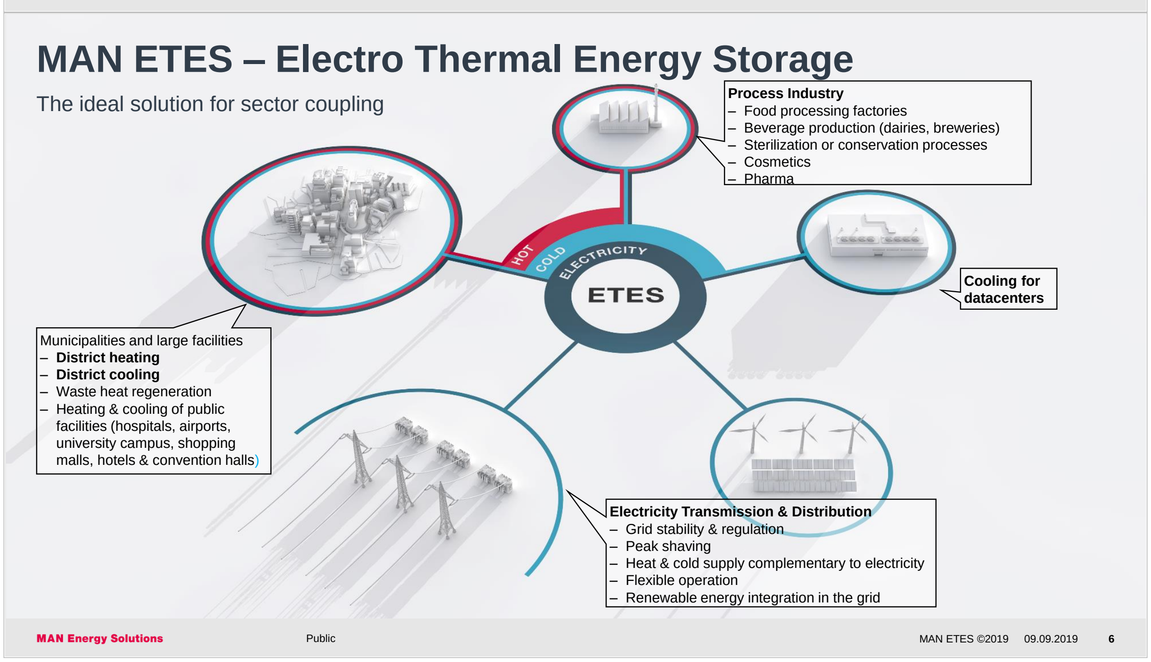
MAN ETES – Electro Thermal Energy Storage
The ideal solution for sector coupling
Process Industry
– Food processing factories
– Beverage production (dairies, breweries)
– Sterilization or conservation processes
– Cosmetics
– Pharma
HOT
COLD
ELECTRICITY
ETES
Cooling for datacenters
Municipalities and large facilities
– District heating
– District cooling
– Waste heat regeneration
– Heating & cooling of public facilities (hospitals, airports, university campus, shopping malls, hotels & convention halls)
Electricity Transmission & Distribution
– Grid stability & regulation
– Peak shaving
– Heat & cold supply complementary to electricity
– Flexible operation
– Renewable energy integration in the grid
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MAN ETES at a glance

ETES USPs for HTHP:

- Temperature ranges 0°- 150°C (single-stage compression)
- Volumes: hundreds of MWhours per day of heat & cold **and** electricity
- Heat & cold storage (short-term or mid-term)
- Storage medium is water
- >25 years lifetime – no efficiency loss
- Natural refrigerant CO_2 (R744)

Applications
Grid peak shaving
Grid balancing
Process industry
District heating
District cooling
Sector coupling

Supply
Heat
Cold
Electricity
Energy Storage

System
Proven technology
Scalability
Flexibility

Environment
Decarbonization
CO_2 reduction
Sustainability
Renewables integration

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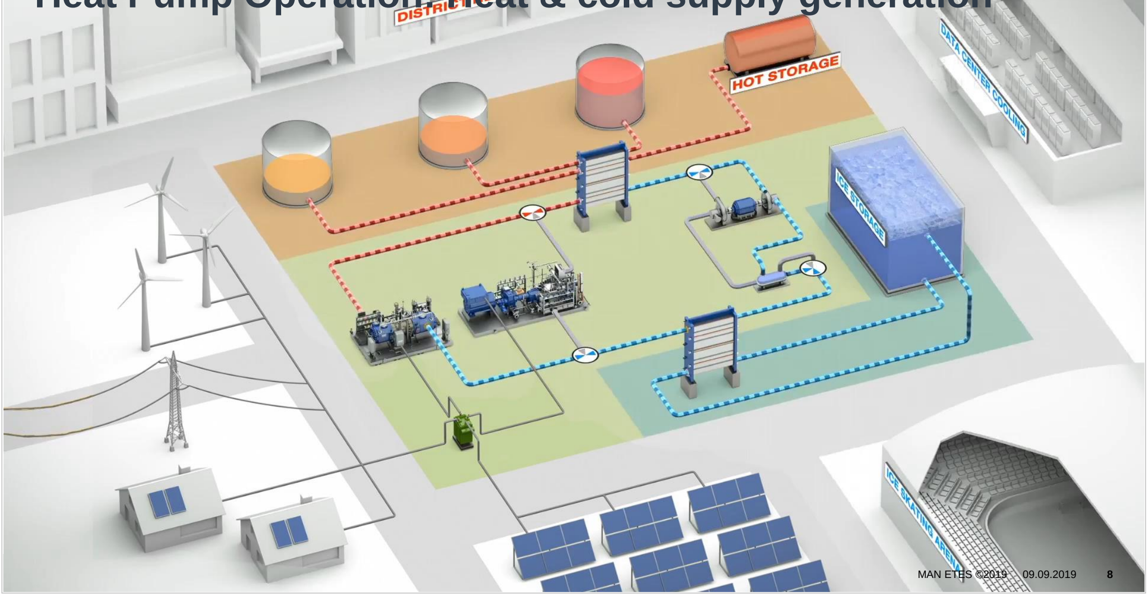
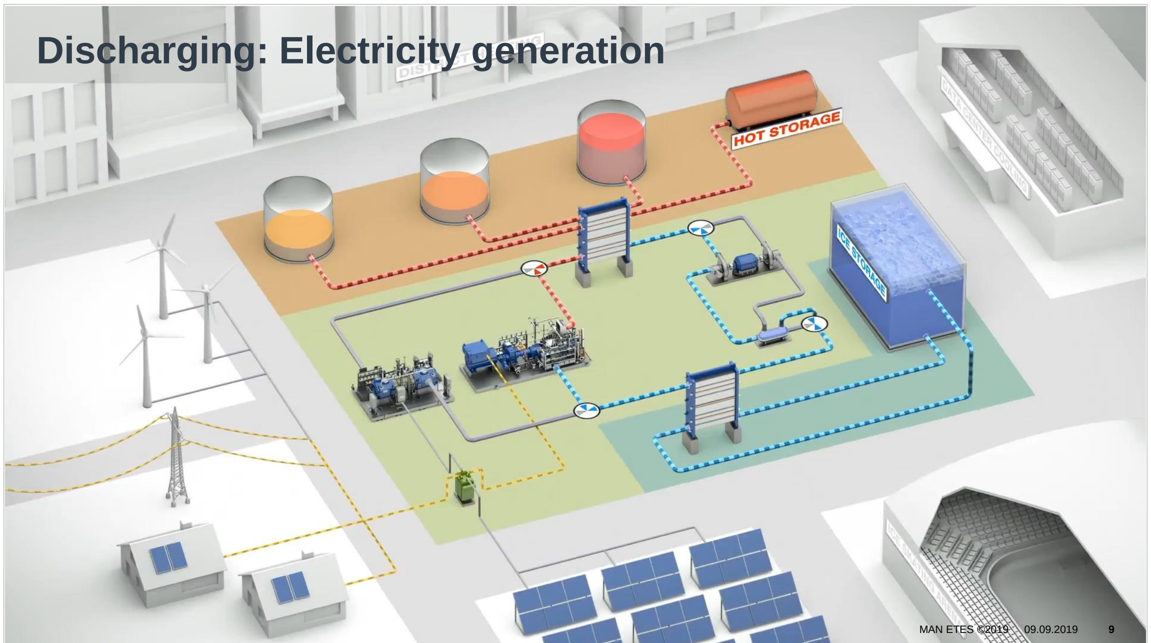
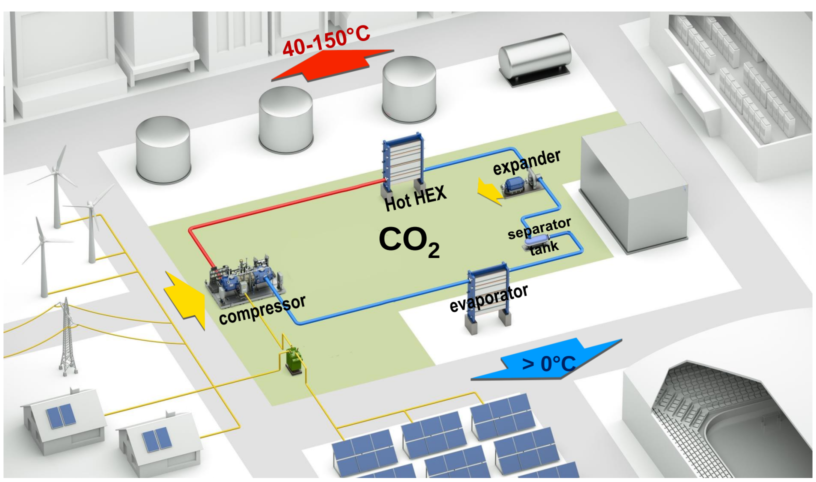
Heat Pump Operation: Heat & cold supply generation





Large Scale Modular High Temp Heat Pump

Fully customizable system for fully customized range of application and operation



1. **Baseline Configuration**
   - modular size (5 – 100 MW_{th})
   - Cold & Hot Temp levels (0°C – 150 °C)

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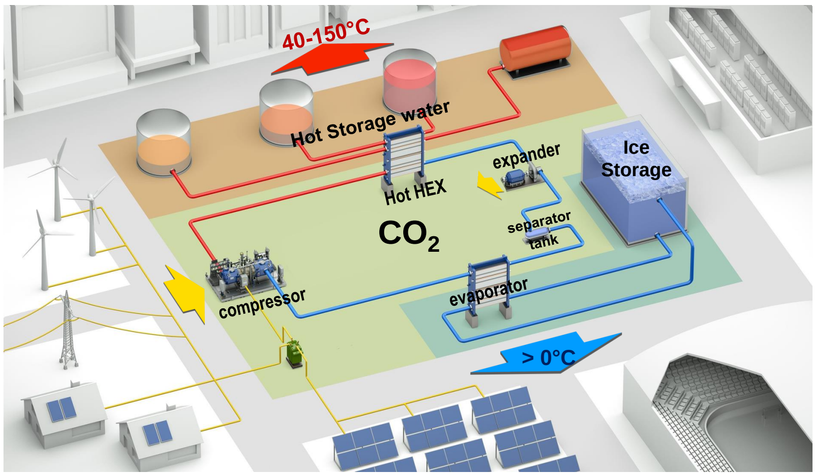
Large Scale Modular High Temp Heat Pump

Fully customizable system for fully customized range of application and operation



1. **Baseline Configuration**
   - modular size (5 – 100 MW_{th})
   - Cold & Hot Temp levels (0°C – 150 °C)
2. **Optional Storage**
   - modular capacity Cold & Hot (MWh)
   - flexible charging time (4 – 24h)
   - flexible thermal export power (MW_{th})

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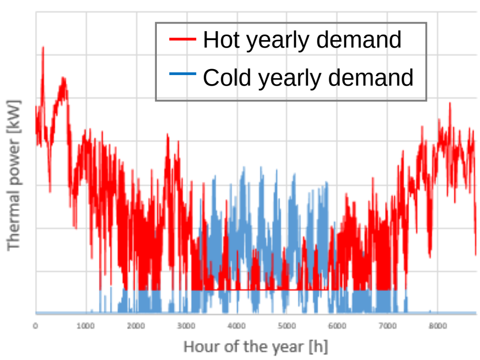
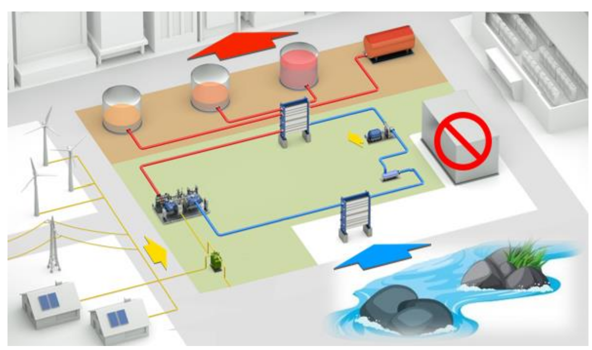
Suitable for seasonal demands

Integrated design with the available environmental conditions





→ Make the use of see / groundwater (typically in winter)

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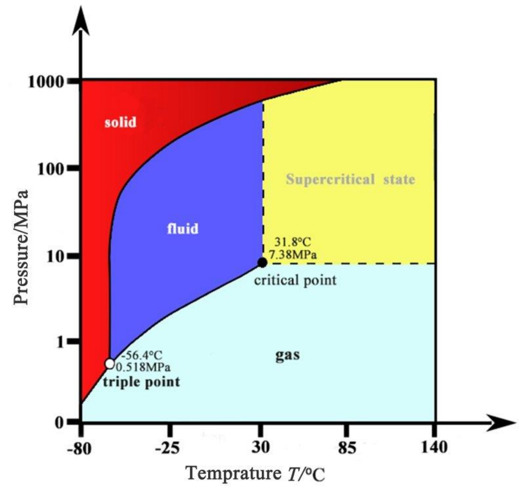
Why CO_2 ?

Supercritical medium



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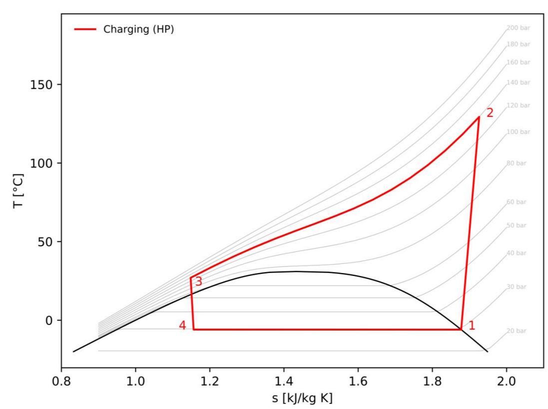
CO_2 adequate for high Power high Temp HP

- Low critical point and high energy density
  - ➢ favorable heat transfer
  - ➢ moderate design pressure of process key equipment ("off the shelve")
- Very compact turbomachinery
- Use of transcritical (for latent heat exchange) and supercritical conditions (for sensible heat exchange)
- Available everywhere at moderate costs (site independant)
- Stable, non-flammable and non-toxic (safety)
- Natural refrigerant, low GWP (sustainability)



T-s diagram of the process cycle with CO_2

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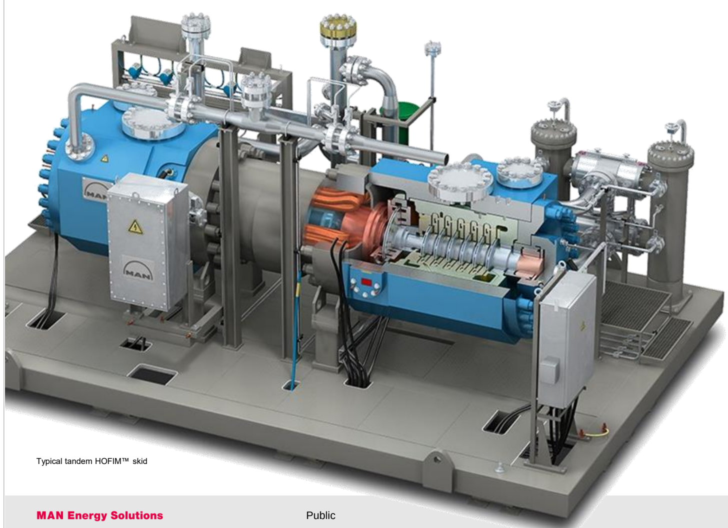
Main piece of equipment HOFIM™ Compressor

High speed Oil Free Integrated Motor compressor



Typical tandem HOFIM™ skid

- Barrel compressor – **robust design**
- Highspeed motor – **fast start/stop operation**
- Cooled by process gas – **heat losses reintroduced into process**
- Running on active magnetic bearings – **oil free**
- Reduced auxiliaries – **increased reliability**
- Fully electric – **remote control**
- Hermetically sealed – **no leackage, no gas emissions**
- Overall **cost optimization** through reduced footprint & weight

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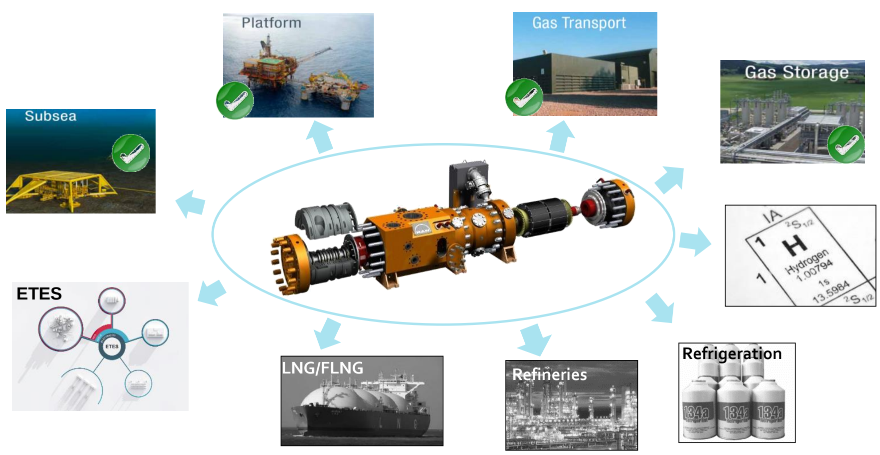
One product fits all applications



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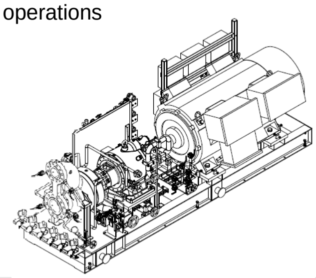
Turbo-Expander

Improve Heat Pump COP

- Suitable pressure and temperature design range
- Suitable and customized for sCO2 application
- Compact design (single radial expander impeller)
- Flexible operation range (25-100% power variation)
- Suitable for start-up/shut down operations



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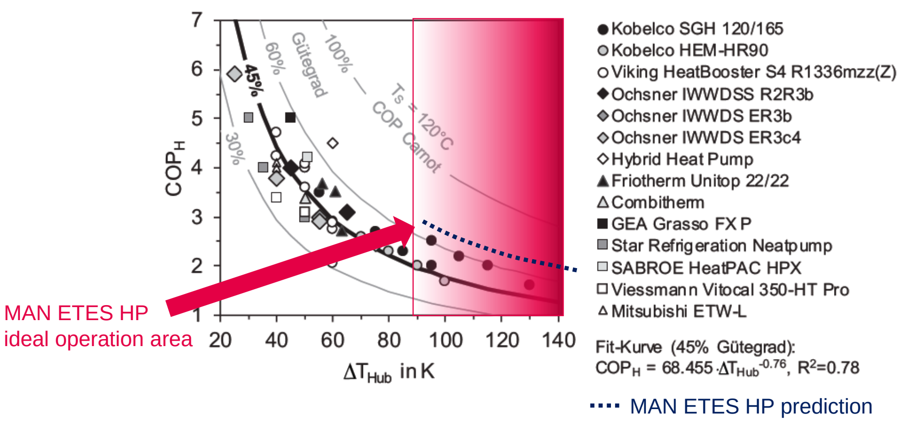
Comparison with other industrial HP solutions

COP Carnot Market Overview



Source: Dr. Cordin Arpagaus: Hochtemperaturwärmepumpen, Marktübersicht, Stand der Technik und Anwendungspotentiale, VDE Verlag 2019

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Example of a 7.5 MW MAN "ETES Heat Pump"

8h charging

Input

60 MWh electrical input

e.g. **7.5 MW$_e$** x 8 hours

Storage

235 MWh heat
(Ranges 20-90° C)

175 MWh cold

Thermal energy

Output

105 MWh heat supply
(Ranges 20-50° C)

130 MWh heat supply
(Ranges 50-90° C)

130 MWh heat supply
(Ranges ≥ 0° C)

Compressor ranges: 5 – +50 MW$_{th}$

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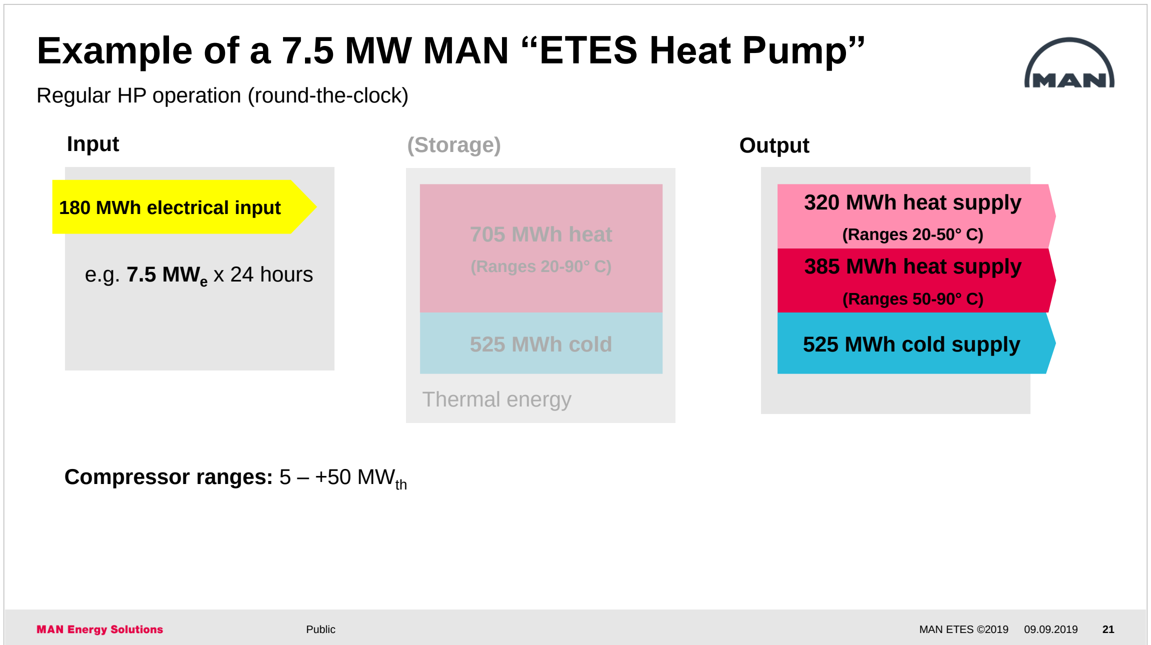
Example of a 7.5 MW MAN "ETES Heat Pump"
Regular HP operation (round-the-clock)
Input
180 MWh electrical input
e.g. 7.5 MWe x 24 hours
(Storage)
705 MWh heat
(Ranges 20-90° C)
525 MWh cold
Thermal energy
Output
320 MWh heat supply
(Ranges 20-50° C)
385 MWh heat supply
(Ranges 50-90° C)
525 MWh cold supply
Compressor ranges: 5 – +50 MWth
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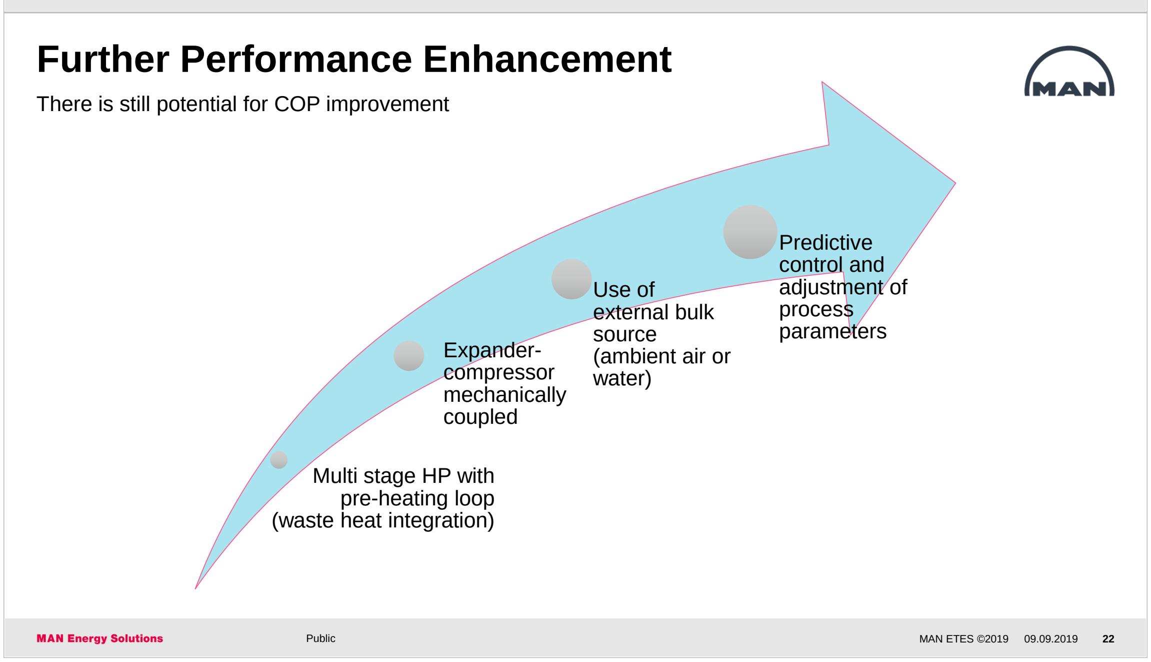
Further Performance Enhancement
There is still potential for COP improvement
Multi stage HP with pre-heating loop (waste heat integration)
Expander-compressor mechanically coupled
Use of external bulk source (ambient air or water)
Predictive control and adjustment of process parameters
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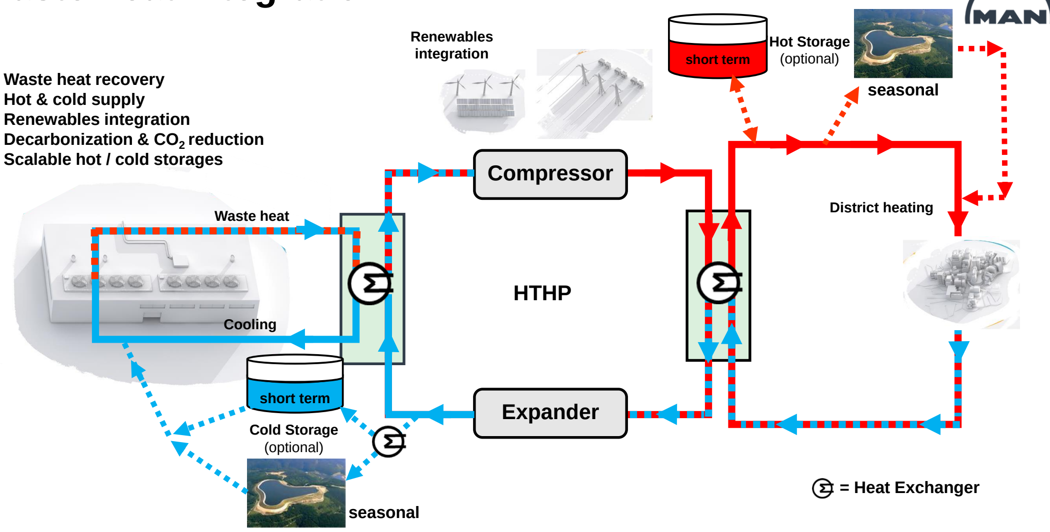
Waste heat integration

- Waste heat recovery
- Hot & cold supply
- Renewables integration
- Decarbonization & CO_2 reduction
- Scalable hot / cold storages



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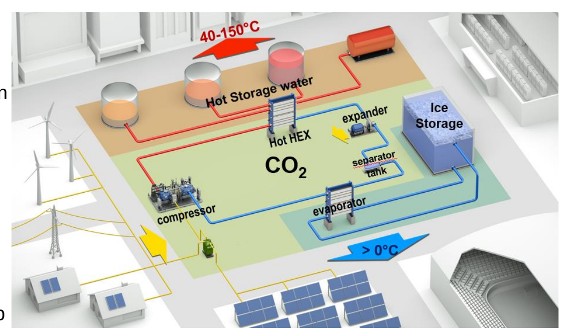
Summary

- Sector coupling and integration of large scale HTHP systems are key pillar solutions to make **decarbonization** happen.
- MAN ETES **trigeneration** system is a multifunctional **energy storage** solution for **heat & cold & electricity**
- Main part of this system is the **ETES Heat Pump**:
  - modular and free up-scalable system, based on proven and reliable technology (**Hofim oil-free compressor**)
  - For high Temp. applications (**0°- 150°C**) in single compression unit
  - Combining **hot & cold storage** for a customized design and flexible operation
- **Sustainable and environmentally friendly** system:
  - >25 years lifetime, no efficiency loss
  - Natural refrigerant CO_2 in an emission-free closed loop
  - Storage medium is water only



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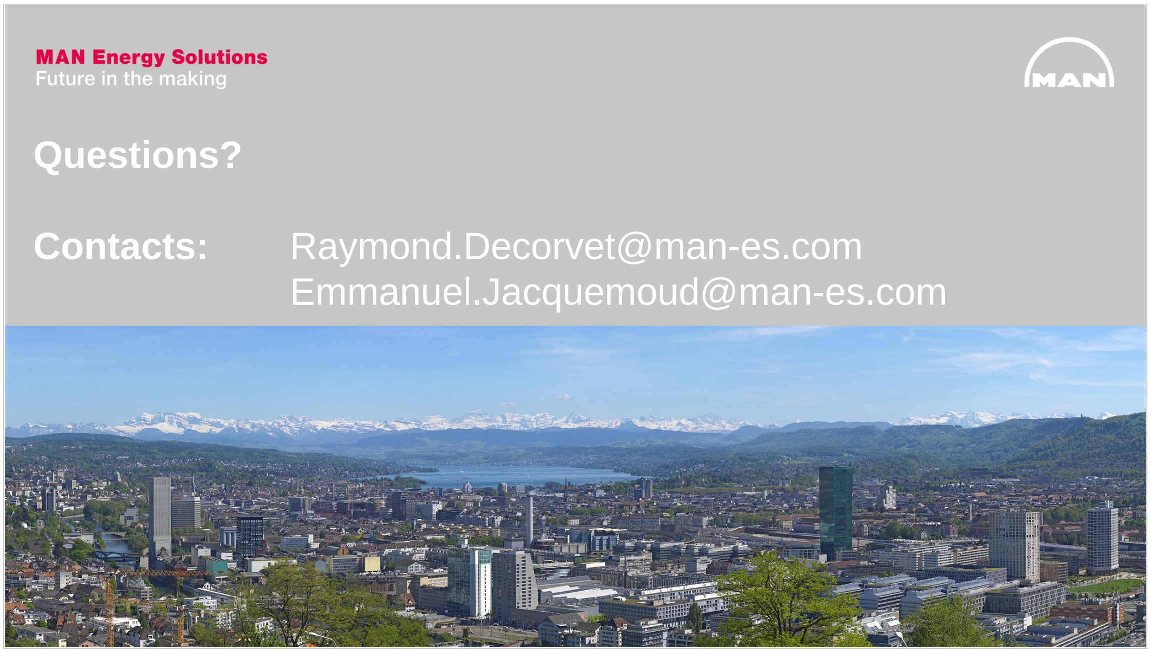
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Future in the making
MAN
Questions?
Contacts:
Raymond.Decorvet@man-es.com
Emmanuel.Jacquemoud@man-es.com

4 Poster Session

1.1 Development of a combined absorption-compression heat pump test facility at high temperature operation, Marcel Ahrens, NTNU

1.2 Turbo compressor for R718 (water) based heat pump applications, Christian Schlemminger, SINTEF

1.3 Optimized heat pump drived steam supply systems, Hans Madsbøll, DTI

1.4 Design of centrifugal compressors, Hans Madsbøll, DTI

1.5 Integration and optimization of a reversed Brayton cycle coupled with renewables and thermal storage in an oil refinery, Vergis Kousidis, DTU

1.6 Performance analysis of a high-temperature heat pump for compressed heat energy storage system using R-1233zd(E) as working fluid, Abdelrahman Hassan, UPV

1.7 Lubricant investigation for high temperature heat pump application, Nikhilkumar Shah, Ulster University

1.8 High-temperature refrigeration system for cooling of automotive PEM fuel cells, Steffen Heinke, TU Braunschweig

1.9 Development of a high temperature heat pump prototype with scroll compressor for industrial waste heat recovery, Carlos Mateu-Royo, Universitat Jaume I

1.10 High-temperature heat pump in a Swiss cheese factory, Cordin Arpagaus, NTB Buchs

1.11 Modelling of an open heat pump cycle for waste heat recovery in an industrial batch process, Andrew Marina, TNO

1.12 Dynamic measurements on a steam producing industrial heat pump, Andrew Marina, TNO

Development of a combined absorption-compression heat pump test facility at high temperature operation

Marcel Ulrich Ahrens, Armin Hafner, Trygve Magne Eikevik

Norwegian University of Science and Technology - NTNU,
Department of Energy and Process Engineering, Trondheim, Norway
marcel.u.ahrens@ntnu.no

Abstract

The present work deals with the analysis of the current demands and opportunities of the development of a combined absorption-compression heat pump (CACHP) test facility at high temperature operation.

The environmental impacts of human activities pose a threat to humankind through climate change. A sustainable and energy-efficient industry is central in continuing to support the needs of an increasing population. The importance of energy efficiency to protect and improve the global environment has been emphasized in several recent publications. Conti et al. (2016) have noted an increasing energy demand in the industrial sector with a clear trend for the future in recent years. In addition, in various industrial processes large amounts of low grade waste heat are not exploited due to the lack of waste heat utilization. Arpagaus et al. (2018) have found that industrial processes with a large heat demand in the temperature range up to 150 °C and useable waste heat streams in particular have a great potential for industrial high temperature heat pump applications. Simultaneously, the demand for environmentally benign working fluids such as the natural fluids, ammonia and water, become more dominant.

Due to the given properties, CACHPs with a zeotropic ammonia-water mixture as working fluid provide a good solution for industrial high temperature heat pump applications. The working principle is based on the Osenbrück cycle, which extends a vapour compression heat pump cycle with an additional solution circuit (Osenbrück, 1895). This extension provides the typical properties of CACHP systems such as the attainable high sink temperature combined with high temperature lift and non-constant temperature glide. In addition, the system offers the possibility of varying the composition and the circulation ratio of the working fluid in order to adapt the operating conditions, as well as achieving higher sink temperatures at relatively low discharge pressure. For this reason, the ammonia-water CACHP concept is interesting for industrial high temperature applications as for instance the utilization of waste heat streams. Furthermore. the functionality of this process in the industrial sector was already been proven by Nordtvedt et al. (2013) using standard available refrigeration components and achieving sink outlet temperatures up to 120 °C.

In recent years, various authors, such as Jensen (2015) and Nordtvedt (2005), have investigated the CACHP system to identify challenges and potentials for the optimization of process parameters. The development was constantly pushed forward due to the increasing interest in the use of ammonia in refrigeration systems and the associated efforts to optimize the components. They determined the compressor as a dominant constraint on achievable operating conditions due to the limitation of high pressure and discharge temperature. In addition, there is a lack of knowledge and experience in operating the system at higher operating conditions. In particular, this relates to the design of the absorber and the

expiration of the absorption process at high pressure and temperature levels. Current results and findings for the design of the system and the various components are to be used for the construction of the planned ammonia-water CACHP test facility.

In conclusion, it appears that the development of a test facility for conducting experimental trials offers promising opportunities for the further testing and development of the CACHP system and specific components. This includes the testing and optimization of operating parameters and strategies for various industrial applications as well as the verification and further development of numerical models.

Keywords:
Industrial high temperature heat pump, Combined absorption/compression heat pump, Ammonia/water mixture

References

Arpagaus, C., Bless, F., Uhlmann, M., Schiffmann, J., Bertsch, S.S. 2018. High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials. Energy 152, 985-1010.

Conti, J., Holtberg, P., Diefenderfer, J., LaRose, A., Turnure, J.T., Westfall, L. 2016. International energy outlook 2016 with projections to 2040 (No. DOE/EIA-0484 (2016)). USDOE Energy Information Administration (EIA), Washington, DC (United States). Office of Energy Analysis.

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Nordtvedt, S.R., Horntvedt, B.R., Eikefjord, J., Johansen, J. 2013. Hybrid heat pump for waste heat recovery in Norwegian food industry. Thermally driven heat pumps for heating and cooling, 57-61.

Osenbrück, A. 1895. Verfahren zur Kälteerzeugung bei Absorptionsmaschinen. Deutsches Reichspatent DPR 84084.

Development of a combined absorption-compression heat pump test facility at high temperature operation

HighEFF

NTNU

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e-mail: marcel.u.ahrens@ntnu.no

Abstract

The present work deals with the analysis of the current requirements and opportunities of the development of a combined absorption-compression heat pump (CACHP) test facility at high temperature operation. The CACHP system combines the technologies of an absorption and vapour compression heat pump with a mixture of ammonia and water as refrigerant. The functionality of this process was already been proven in the industrial sector using standard components In recent years, several studies have investigated the CACHP system to identify challenges and opportunities to increase the achievable temperature range and optimize process performance. The compressor and the absorber were identified as critical components for increasing the achievable temperature level. For this purpose, this study investigates the currently available solutions of the critical main components for use in the development of a planned CACHP test facility that can be used for experimental investigations and increases the achievable sink outlet temperature on the secondary side up to 140 °C to 180 °C.

Keywords: Industrial High Temperature Heat Pump, Hybrid Absorption-Compression Heat Pump, Ammonia-Water Mixture

Motivation

- Increasing energy demand in various industrial processes combined with great potential for heat pump applications to utilize waste heat
- The functionality of this process has already been proven in the industrial sector using available standard refrigerant components
- Utilization of the developments and improvements of the recent years

➢ **The aim is to develop a combined absorption-compression heat pump test facility for the conduction of experimental investigations**

Combined Heat Pump System with NH_3-H_2O

The combined NH_3-H_2O absorption-compression heat pump (CACHP) combines the technologies of an absorption and vapour compression heat pump with a mixture of ammonia and water as working fluid. Heat is extracted and released at non-constant temperature glides and the necessary compression ratio is lower compared to conventional vapour compression heat pumps. These properties make the CACHP system interesting for the use in various industrial high temperature applications.

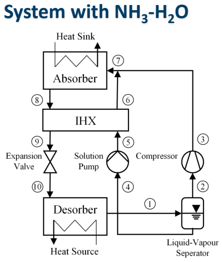

Figure 1: Simplified representation of a combined absorption-compression heat pump cycle

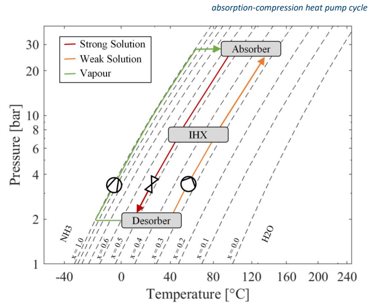

Figure 2: PTX diagram of a saturated liquid for an ammonia-water mixture with the main components of a combined absorption-compression heat pump

Results and Discussion

1. Different compressor solutions are conceivable for the compression:

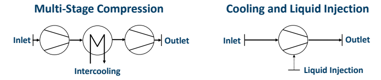

Table 1: Overview of solutions for the ammonia vapour compression

	Single-Stage	Multi-Stage	Liquid Injection
Advantages	Lower installation costs	Possible intercooling	Oil-free operation
Challenges	High discharge temperature	Higher complexity	Evaporation process

➢ **The occurring discharge temperature is the main limitation in achieving higher pressure levels of the CACHP system**

2. As Absorber an plate HX in the following absorption modes can be used:

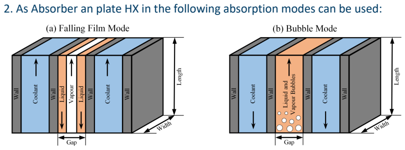

Figure 3: Representation of a plate heat exchanger with falling film and bubble absorption mode

➢ **There is a lack of knowledge about the design of the absorber and the process of the absorption at desired pressure and temperature levels**

Conclusions

- Increasing interest in industrial high temperature heat pump applications with natural refrigerants as working fluid
- For the development of the planned CACHP test facility, further modifications and improvements of the critical main components and the CACHP cycle design are required

➢ **The possibility of experimental research through the planned test facility offers great potential for the investigation of existing approaches and for the identification of further potentials**

Acknowledgement

The work is part of HighEFF - Centre for an Energy Efficient and Competitive Industry for the Future, an 8-year Research Centre under the FME-scheme (Centre for Environment-friendly Energy Research, 257632). The authors gratefully acknowledge the financial support from the Research Council of Norway and user partners of HighEFF.

HighEFF

Centre for an Energy Efficient and Competitive Industry for the Future

Turbo compressor for R718 (water) based heat pump applications

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SINTEF Energi AS, Department of Thermal Energy, Trondheim, Norway,
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Keywords:
High temperature heat pump, R718, compressor technology, Carnot efficiency

Abstract

Open loop heat pump systems, also known as Mechanical Vapour Recompression (MVR) systems, are a special form of heat pump technology using water (R718) as working fluid. Excess steam from industrial processes (e.g. superheated steam drying, distillation, evaporation) is compressed and the condensation heat is used to re-heat the process. Thereby, the natural gas burner can be replaced reducing CO_2-emissions and increasing the energy efficiency of the core process. The developed target is a condensation temperature of up to 160°C at a combined pressure ratio of 6, when compressing over two stages. At present, however, only limited compressors are available for this application, which is why turbo compressors are a suitable choice for steam compression, especially in small plant sizes (<1000 $kW_{thermal}$). In addition, they are a lubricant-free alternative to conventional piston or screw compressors. This technology has a high potential to be used in high-temperature heat pumps (HTHP) with heat sinks around 100°C, both in open and closed heat pump systems.

In the course of this work, a two-stage turbo compressor system for steam compression was developed, built and analysed. The used turbo compressor prototypes are a further development of C38-turbocharges from the automotive industry (Rotrex A/S, Copenhagen) and can be operated with a standard DC electric motor. Thanks to the special gearbox design, rotational speeds up to 80000 rpm can be realised for an impeller size of 15 cm and no external oil cooling system is required. To achieve the required pressure ratio of 6, two stages are necessary, since centrifugal compressors work ideally with pressure ratios from 2.0 – 3.0. Furthermore, de-superheating between the stages is required, as the water overheats strongly during compression.

The present evaluation of the preliminary test results is based on few operational points without optimisation or operation under optimal conditions. Nonetheless, the condensation temperature of the steam after the second stage was 143.5°C for operation with 76000 rpm. This corresponds to a temperature lift of 43.5 K regarding the heat sink temperature of 100°C (p_{in}=1.00 bar ± 0.05 bar) with a mass flow of about 800kg/h and a pressure ratio of 4.21 (p_{out}/p_{in}). The determined coefficient of performance considers the pressure drop of the system, the occurring losses of inverter, motor and gearbox, the total heat losses and compression losses of both compressor units. The COP of the system in heat pump operation was 4.2 and corresponds to 44.9 % of the Carnot efficiency. The achievable

pressure ratios and condensation temperatures, as well as the efficiencies will be higher for operation at 100 % speed (80000 rpm) and further improvement of the system.

The performed tests show that compression systems with lubricant-free compressions rooms have a high potential for open and closed HTHP applications. Turbo compressors can thus potentially be a cost-effective alternative to conventional compressors, since mass-produced and thus cost-effective components from the automotive industry can be used. Possible applications of the tested turbo compressors are classical mechanical vapor recompression (MVR) with significantly higher temperature lifts, heat pump applications of over 100°C, integrated cascade solutions with heat sources below 100°C and industrial steam generation with simultaneous heat recovery. Water (R718) as a refrigerant is suitable for heat source temperatures around 100°C, with high industrial acceptance and environmental compatibility at the same time.

The contents of this work have been achieved within the DryFiciency project (www.dry-f.eu). DryFiciency receives financial support from the European Commission under grant number 723576 within the Horizon 2020 programme.

Turbo compressor for R718 (water) based heat pump applications

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Introduction

Open loop heat pump systems also known as **Mechanical Vapour Recompression (MVR) systems** are a special form of heat pump technology using water (R718) as working fluid. Excess steam from industrial processes (e.g. superheated steam drying, distillation, evaporation) is compressed and the condensation heat is used to re-heat the process. Thereby, the natural gas burner can be replaced reducing process CO_2-emissions and increasing its energy efficiency. The open loop heat pump system developed by the DryFiciency consortium uses **advanced, low-cost turbo compressor technology** originating from the automotive sector. The developed target is a **condensation temperature of up to 160°C** at a combined pressure ratio of 6, when compressing steam over two stages.

Why water as working fluid?

- Excellent properties as refrigerant in this temperature range
- One of the most abundant and safest working fluid
- Undeniably climate-friendly

Why turbo compressors?

- Lubricant-free alternative to conventional piston or screw compressors
- High potential to be used in high-temperature heat pumps with heat sinks around 100°C, both in open and closed loop heat pump systems.

Where to classify the MVR?

- Heat Source Temperature: T=100°C, Temperature Lift: ΔT=55-60K

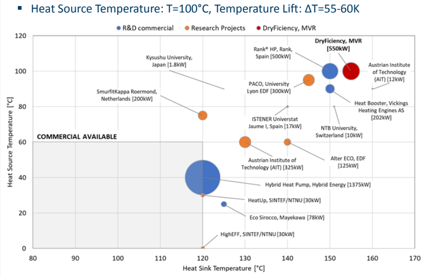

Figure 1: HTHP technology overview (based on Mateu-Royo et al. 2019)

Two-stage turbo compressor system

- Turbo compressor prototypes are further development of C38-turbochargers from the automotive industry (Rotrex A/S, Copenhagen)
- Steam centrifugal compressor design for high compression efficiency
- Special gearbox design which results in rotational speeds up to 80000 rpm for an impeller size of 15 cm
- Application of standard DC electric motor
- No external oil cooling system required
- Easy connection to the existing piping system

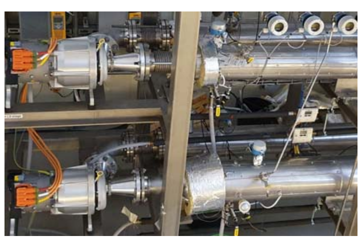

Figure 2: Installed EA42 turbo compressors for tests in superheated steam atmosphere

- Two stages necessary to achieve high pressure ratios, since centrifugal compressors work ideally with pressure ratios from 2.0 - 3.0
- Standard solutions require a complex setup and have a larger footprint
- De-superheating between the stages required, as the water overheats strongly during compression

Preliminary test results & further expectations

- Preliminary test results with the two-stage turbo compressor system under various operating conditions:

Table 1: Performance analysis of the first tests for a two-stage turbo compressor system (T_{in}=110°C ± 5K, p_{in}=1.0bar ± 0.05bar)

#	*Speed* (rpm)	$\dot{m}_t$ (kg/s)	p_{out}/p_{in} (-)	T_{lift} (K)	$\dot{Q}_{cond.}$ (kW)	P_{el} (kW)	COP (-)	η_{carnot}
1.	64000	0.179	**2.71**	**27.7**	410.3	72.8	**5.6**	**30.6 %**
2.	64000	0.164	**2.97**	**31.5**	376.3	67.0	**5.6**	**44.6 %**
3.	64000	0.149	**3.05**	**32.7**	340.7	62.9	**5.4**	**47.9 %**
4.	68000	0.183	**3.37**	**35.8**	420.4	83.0	**5.1**	**44.8 %**
5.	72000	0.201	**3.77**	**39.6**	464.4	100.2	**4.6**	**45.4 %**
6.	76000	0.221	**4.21**	**43.5**	515.1	122.1	**4.2**	**44.9 %**

- The determined coefficient of performance (COP) considers the pressure drop of the system, the occurring losses of inverter, motor and gearbox, the total heat losses and compression losses of both compressor units
- The present evaluation is based on few operational points without optimisation or operation under optimal conditions
- Higher pressure ratios to be expected at 80000 rpm (100% speed)
- The achievable condensation temperature will be above 150°C with further system related improvements

Conclusions

- Successful test of two-stage turbo compressor MVR-system with temperature lift of 44K and pressure lift of 4.2 resulting in condensation temperature of 144°C with a COP of 4.2
- Lubricant-free compression room has a high potential for open and closed loop HTHP applications.
- Compressors are a cost-effective alternative to conventional blowers and fans up to 2 MW_{th} MVR systems
- Water (R718) suitable for heat source temperatures around 100°C, with high industrial acceptance and environmental compatibility at the same time

Acknowledgements

The contents of this poster have been achieved within the DryFiciency project (www.dry-f.eu). DryFiciency receives financial support from the European Commission under grant number 723576 within the Horizon 2020 programme.

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MATEU-ROYO, Carlos, NAVARRO-ESBRÍ, Joaquín, MOTA-BABILONI, Adrían, AMAT-ALBUIXECH, Marta and MOLÉS, Francisco, 2019. State-of-the-art of high-temperature heat pump for low-grade waste heat recovery. *XI National and II International Engineering Thermodynamics Congress*, Universidad de Castilla-La Mancha, June 2019.

Technology for a better society

Optimized heat pump driven steam supply systems

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Keywords:
High temperature heat pump, R718, steam system, applications

Abstract

Currently, steam production in the industry is primarily based on traditional fossil fired boiler solutions. The green transition from fossil fuels to renewable energy sources requires the development of technologies for energy-efficient electric steam production. In this context, high-temperature heat pumps are an attractive solution since waste heat and other heat sources can be converted into steam with high efficiencies.

In the newly started project, a concept for the establishment and optimization of heat pump-based steam production systems will be developed based on the use of the latest technology in components and regulation. A demand-driven approach to system optimization and related methods, including necessary registrations, will be used in connection with an optimization tool for energy optimization and design of new steam systems as well as retrofitting of existing systems. In addition, the methods will be documented in a guide based on a methodological approach illustrated with examples, which show the situations of the participating companies, in order to indicate and illustrate possibilities in concrete cases.

The main results of the project will be:

- A concept for high-temperature heat pumps used for steam production
- A steam system optimization tool
- Design instructions for steam systems
- Reporting, including mapping potentials by sectors and technologies

Overall, industrial companies, which use steam in connection with production processes, and their advisors are provided with new knowledge, which enable them to implement electrically powered high-temperature heat pumps in the supply systems and to realize significant energy efficiency through the need to adapt the steam supply as well.

The project includes four case studies, which are based on existing steam supply systems, from four project partners. The existing steam supply systems are described in for example [1], and literature studies show that typical losses in existing systems can be expected to be in the range of 25 – 50% of

the total energy consumption due to multiple loss sources such as boiler efficiency, flash losses, steam trap losses, piping heat loss, etc. (for example [2])

The very first case study, an autoclave for processing pet food, has been modelled as shown on the poster. The analysis shows promising results, e.g. a COP in the range of 3, but it also shows the need for further technological development of high-temperature heat pumps, control strategies, and energy storage systems.

The other case studies are:

- Laundry – both washing tunnel and industrial dryers. In particular, the drying process will require further development of high-temperature heat pumps in the range of 180 – 200 °C.
- Tunnel oven with hot air in the range of 150 – 220 °C. The same situation applies, including some special requirements in connection with the choice of heat source.
- Parboil and boiling of vegetables – basically, the same issues concerning the choice of heat source for the heat pump are expected. Temperature levels are somewhat lower.

The project runs until April 2021.

[1] United Nations Industrial Development Organization (UNIDO), 2016, Manual for Industrial Steam Systems, Assessment and Optimization
[2] Swagelok Energy Advisors, Document no 33, Steam Systems Best Practices, Steam System Thermal Cycle Efficiencies – Part One, 2011

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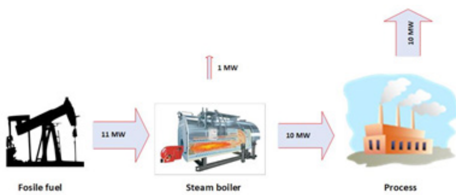
OPTIMIZED HEAT PUMP DRIVEN STEAM SUPPLY SYSTEM

ELFORSK 351 - 022



Traditional Gas Fired Steam System

Primary energy is added into the process as needed – and all the energy is eventually lost to the ambient.

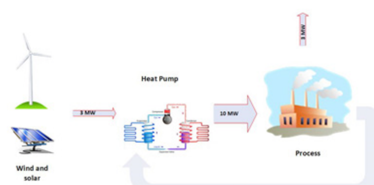
Potential recovery of 'waste heat'.

Heat Pump Driven System

Less primary energy is added as the heat pump mainly transfers energy between temperature levels instead of adding energy into the process.

It is necessary to keep track of the energy as heat source for the heat pump.

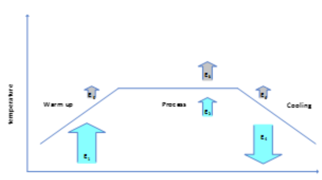
Simultaneously cooling and heating – or storage/buffer tank.

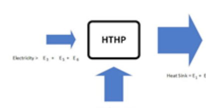


Example: Autoclave

Three phases: **warm up – process – cooling**

The heat pump needs the cooling load in order to deliver the heating. Due to the lack of simultaneity, a buffer tank is needed.





Steady State EES Model

Partners

Stryhns, C&D Foods, BERENDSEN part of elis, ardo, AURA energi, E-TEK Engineering, INNOTERM, Solid energy, DANISH TECHNOLOGICAL INSTITUTE

Design of Centrifugal Compressors

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Keywords:
High temperature heat pump, R718, compressor design, centrifugal, applications, software package

Abstract

One of the promising approaches to high-temperature heat pump designs is the utilization of steam as the fluid as a very competitive alternative to the synthetic HFOs, which might be introduced in the near future. The turbo compressor would be an obvious technology to apply, as would in particular the centrifugal compressor for the high-temperature range. The compressor is compact and potentially cost effective, the efficiency is high, oil free is an option, and steam is environmentally friendly, etc.

The design procedure for a compressor is a highly iterative process as there are several options and several design choices to be made.

Constraints
Specifications
Aerodynamic Design and Performance
Mechanical Design
Rotor Dynamics and Bearing design
Materials and Cost
Manufacturing Method and Cost

Danish Technological Institute holds a R&D license for the full, commercial software design package from ConceptsNREC, the world's leading company in connection with full compressor design. The software package has been used to design a number of both axial and centrifugal prototypes of water vapor compressors.

For the aerodynamic design, the package contains of two tools, a 1-D very fast software tool based on correlations, which have been calibrated by several hundred existing compressor designs, and a full 3-D geometry generation package, including 2-D and 3-D grid generation as well as CFD analysis.

A great deal of the optimization work is carried out with the 1-D package, where all the basic dimensions, blade angles, and speed can be varied along with many other parameters. Key figures like specific speed, blade loads, etc. are calculated together with all the detailed flow angles, velocities,

Mach numbers, static and total pressure, enthalpy, entropy, velocity, flow angle, etc. A total of more that 600 variables.

3-D designs are created on the basis of a few of the optimized 1-D designs for a full 3-D optimization, i.e. a slow and lengthy process to modify the geometry in order to meet the predicted performance from the 1-D analysis. Parameters such as hub and shroud contours, detailed blade angle distribution along the impeller, and detailed blade thickness distribution can be varied along with detailed diffuser and volute design. These variations are analyzed with comprehensive 3-D CFD calculations, where detailed flow phenomena can be studied and modified in order to track down the optimal design.

During this process, there is a close interaction with the mechanical design, i.e. material choice, peak stress calculations, shaft assembling method, and natural frequency calculations.

As there is a great number of design parameters, it is important always to optimize the design for the specific application in question. The final, optimized design depends on whether the design target is maximum pressure ratio, best efficiency, most compact, lowest cost, largest range or yet another constraint.

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DESIGN OF CENTRIFUGAL COMPRESSOR

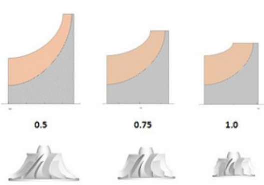
ConceptsNREC software package

1. Specifications
   – media, pressure ratio, mass flow, operational conditions, misc. limitations, materials, manufacturing method, drive, bearings, etc.

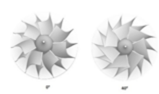
Iterations

2. Parameter study based on specifications Compal – 1D software for centrifugal compressors (AXIAL for axial compressors), for example:

Specific speed

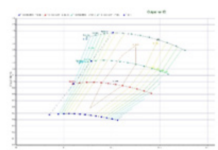


Backsweep angle



Number of blades

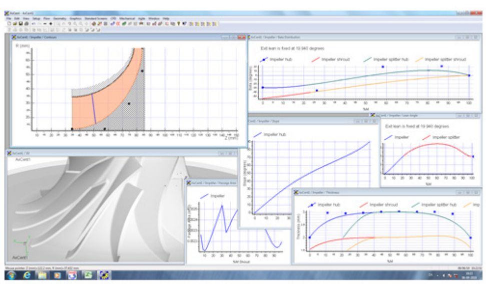
Hub/tip ratio

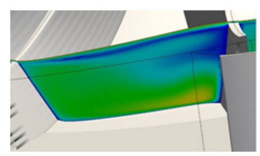


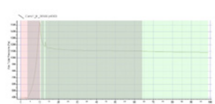
Output:
compressor map, stall margin, dimensionless key figures, rotational speed, flow angles, pressure, temperature, Mach numbers, etc.

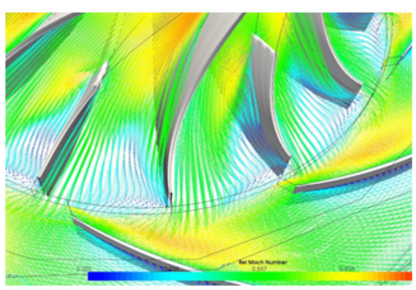
Iterations

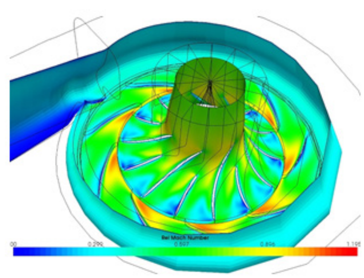
3. AxCent – 3D software for both axial and centrifugal compressors
   – typical design screen and CFD results











Integration and optimization of a reversed Brayton cycle coupled with renewables and thermal storage in an oil refinery

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Keywords:
R-744, Reversed Brayton Cycle, Energy mix optimization, Electrification, Industrial processes

Introduction

As greenhouse gas emissions from fossil fuel combustion are one of the main factors for global warming, the EU has imposed policies and regulations on climate and energy [1]. In the 2030's climate and energy framework, the goal is set to 40% reduction in greenhouse gas emissions from the level of those in 1990 [2]. Denmark has even more ambitious targets. The Energy Strategy of 2050 aims at Denmark being completely independent from fossil fuels [3]. For that reason, continuous research is ongoing for the removal and replacement of fossil fuels.

The share of renewable energy technologies in the energy mix has increased over the past years, mainly in electricity production. Society is gradually moving towards a future with electrified systems based on renewable sources. Concerning heat production, heat pumps are a highly attractive for electrification, which could substitute fossil fuels based boilers and furnaces. On an industrial level, there is a large demand in heat in high temperatures over 100 ˚C, which designates the potential of integration of High-Temperature Heat Pumps (HTHPs) [4]. Because of high temperature lifts accompanied with high temperature applications, the energetic performance of heat pumps deteriorates. Therefore, HTHPs could be considered in combination with large shares of renewable electricity sources. The renewables enable low levelized cost of electricity, which would improve the economic feasibility of the heat pump system.

In this study, the potential of a HTHP project is evaluated from a technoeconomic perspective when coupled with renewable electricity sources and thermal storage. Through optimization, the capacities of the considered technologies are determined, and the project is compared with conventional combustion technologies and electric boilers [5]. The concept is applied to the case study of an oil refinery and conclusions were extracted for such an industry.

Case Study

Crude Oil preheat trains are designed to reduce energy in terms of fuel combustion. Petroleum recovered from a reservoir is, at first, desalted and then heated in preheat Heat Exchanger Network (HEN). In a series of heat exchangers, heat from distillation cuts is transferred to crude oil, which is then heated in the Atmospheric Distillation Unit (ADU) furnace to a temperature close to 360 ˚C before it enters a fractionating column operating close to atmospheric pressure, wherein fractions with different boiling points are separated off. The remnants of atmospheric distillation are further heated and distilled in vacuum [6].

In this project, the furnace before the ADU is to be replaced with HTHPs leading to partwise electrification of the crude preheat train process and the removal of its most polluting components. The revamping of the heating process of crude is considered to be applied to an already retrofitted site, from where three crude oil and several distillation fraction streams were extracted and comprised subjects of the sink and source side of the heat pumps respectively [7].

Methods

Heat Pump Integration Scenarios

For heat pump integration, alternative cases were distinguished and investigated. Two following scenarios were formulated; the crude is heated to the (1) desired temperature (≈360 ˚C, 34.4 MW) and (2) to a lower temperature (=300 ˚C, 16.96 MW) before it enters the ADU. The latter is formulated as lower temperature lifts will result on a better energetic performance and additionally the temperature at the outlet of the compressor is going to be lower. Also, heat exchangers are more susceptible in fouling as crude is heated in higher temperatures [8].

For each scenario stated, different sub-scenarios were created, depending on the number of heat pumps and how they are integrated in order to transfer heat from distillation cuts to the crude. In sub-scenario 'A', one heat pump is integrated, where heat is supplied indirectly from fractions to crude. Through a HEN distillation, fractions increase the temperature of a heat transfer medium that acts as source in the HTHP. On the sink side, heat is received from another heat transfer medium and is then applied to the crude streams through another HEN. The chosen Heat Transfer Fluids (HTFs) were mineral oil for source and solar salt for sink, as they are considered to be relatively cheap and stable at the temperature levels studied [9]. In sub-scenario 'B', there are three heat pumps, a distillation fraction stream acts as source at each HTHP and the heat is applied at the sink immediately to the crude. Lastly, 'C' is similar to 'B'. There are six heat pumps and the crude streams are divided before they enter the HTHPs, where distillation fraction streams act as source.

Reversed Brayton Cycle

Because of high temperature lifts, there is a high-pressure ratio in HTHPs. That enables the mounting of a turbine in the expansion process so that work is recovered. For the recovered work to be utilized, the turbine is mounted on the same shaft as the compressor. The cycle will operate at supercritical conditions to ensure gas phase of the working fluid. R-744 was chosen, as it is a natural refrigerant with stable operation at required temperatures that also has good heat transfer properties. In the cycle there is also an Internal Heat Exchanger (IHX) which ensures that the working fluid is at appropriate temperature levels to receive and deliver heat at the source and sink respectively.

The HTHPs were designed assuming the isentropic efficiency of the compressor and turbine, as well as the pinch temperature at the source and sink, while the Coefficient Of Performance (COP) was optimized. For optimization of the COP the decision variables were the low and high pressure of the cycle and the degrees of superheat after the expansion process [10].

Heat Storage Integration

Due to very large requirements in heat demand in industrial sector that could be covered by HTHPs, there could be potential on dimensioning the heat pump in an increased capacity and couple it with a heat storage system in order to benefit from the time variance of electricity prices. For this integration, a two-tank configuration was considered in both source- and sink-side.

This would only be applicable in sub-scenario ‘A’, where heat is transferred from distillation cuts to crude through HENs. As the HTHP operates at levels above the heat demand requirements, part of mass-flow of HTFs will flow through the HENs to cover the demand, while the rest will accumulate on the Low-Temperature (LT) and High-Temperature (HT) tanks on the source and sink side, respectively. If the HTHP operates at levels below heat demand, HTF will flow from the aforementioned tanks to the HEN and then back to the HT and LT tanks of the source- and sink-side.

Energy Mix Optimization

Cost models were developed concerning reversed Brayton cycles, wind turbines, photovoltaics and heat storage and were combined with weather data and electricity prices from grid time series in order to formulate the optimization problem. The problem was of linear programming and was implemented in GAMS software [10]. Aim of the programming was the minimization of Levelized Cost of Heat (LCOH), while the optimum capacities of the considered technologies were determined.

Results

The average optimized COP of HTHPs for each scenario and their respective sub-scenarios are depicted in Figure 1. The COP is rather low due to high temperature lifts. The sub-scenarios including more heat pumps most likely designate higher COP, because of better utilization of high temperature distillation fraction streams.

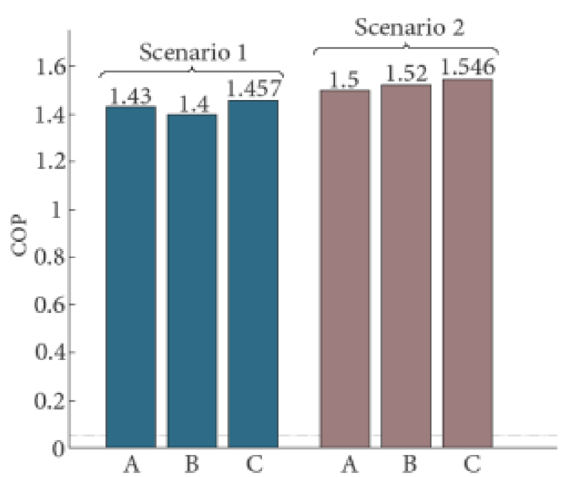

Figure 1. Average optimized COP of Sub-Scenarios

Although the COP is higher in these cases, after optimization of the energy mix capacities and the extraction of the LCOH, the tendencies are different. Due to economy of scale, introducing more heat pumps will lead to higher investment costs and the LCOH of Sub-scenario ‘A’ is lower, even though there are additional costs for the HENs. As that, only sub-scenarios ‘A’ were selected for further investigation. The LCOH values are depicted in Figure 2.

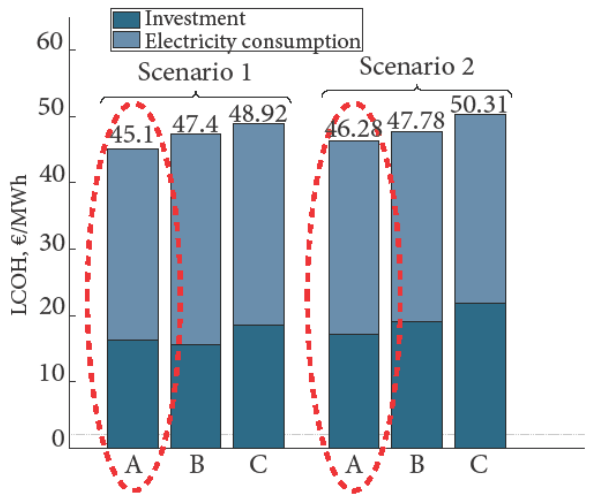

Figure 2. LCOH comparison between sub-scenarios

A comparison of the LCOH of the chosen configuration for each scenario with conventional technologies could be observed in Figure 3. The LCOH will fluctuate between 44 €/MWh and 46 €/MWh, indicating that a LCOH higher than that value for conventional technology will result to a feasible HTHP project. According to those, HTHPs are economically superior to electric and biogas boilers. Although the former may have low investment, it has worse economic performance due to larger electricity consumption, while the latter has very high prices for procurement. The LCOH of natural gas and biomass is of lower value, indicating economic inferiority of HTHPs even when considering the Energy Savings Scheme in Denmark as subsidy [12].

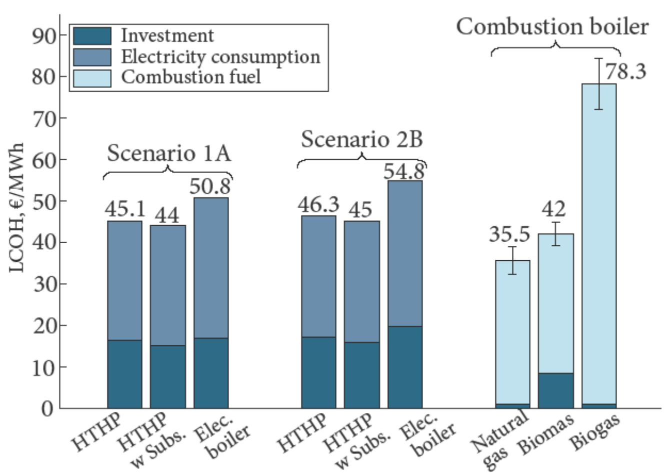

Figure 3. LCOH of HTHPs with and without subsidy and of conventional technologies

Yet, considering projected increases in both biomass and natural gas prices and taxation, in the future there is potential of HTHPs to become more competitive and viable.

The aforementioned results refer to an optimized energy mix. For each scenario the capacities are given in Table 1 and Table 2, along with the COP and the renewable penetration. Wind turbines are chosen in both scenarios and they are coupled with heat storage is scenario 1 and with PVs in scenario 2.

Table 1. Optimal Energy Mix for scenario 1A

TECHNOLOGY	CAPACITY
HTHP	39.6 MW
Wind turbines	28 MW
Heat storage	117.8 MWh
Heat demand	34.4 MW
COP	1.429
Renewable Penetration	37%

Table 2. Optimal Energy Mix for scenario 2A

TECHNOLOGY	CAPACITY
HTHP	16.96 MW
Wind turbines	10.5 MW
PVs	3.8 MW STC
Heat demand	16.96 MW
COP	1.5
Renewable Penetration	34%

Conclusions

This work analysed the techno-economic feasibility of reversed Brayton cycles in an oil refinery and it was concluded that configurations with higher amount of heat pumps introduced high investments and resulted in worse economic performance in terms of economic feasibility. The energy technologies mixture optimization designated that all considered technologies are eligible for application. Wind turbines consist a permanent choice of optimization algorithm, while the choice of heat storage was very much dependent on the COP and the heat demand. PVs consisted mostly a filler option to wind turbines. HTHPs were demonstrated to be superior to electric and biogas boilers, but the contrary when compared to biomass and natural gas boilers. Although they seem not that competitive to those boilers, cost projection of these fuels points that HTHPs would be more viable in the future.

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Integration of a Reversed Brayton Cycle coupled with Renewable Energy Systems in an Oil Refinery

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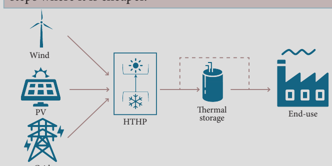
[1] Technical University of Denmark, [2] Danish Technological Institute

INTRODUCTION

Greenhouse gas emissions are one of the main factors for global warming. Denmark contemplates to be a future completely independent of fossil fuels by 2050. For replacement of fossil fuel technologies, heat pumps consist an attractive alternative as they can be efficiently operated with renewable electricity. The demand on high temperature heat in industries necessitates focus on research in the field of High-Temperature Heat Pumps (HTHPs).

OBJECTIVE
The evaluation and optimization of a concept of HTHPs operating with a combination of renewable electricity sources to replace current fossil fuel furnaces in an oil refinery.

APPROACH
Wind turbines and photovoltaics (PVs) along with the grid were considered to supply electricity to HTHPs, combined with heat storage, which could allocate electricity consumption to time steps where it is cheaper.

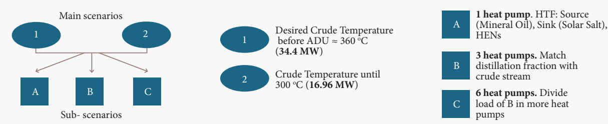


CASE STUDY

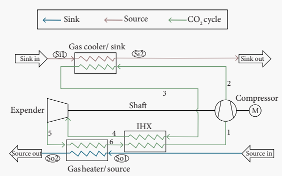
Oil and gas industry is very energy consuming since large amounts of heat are needed to increase the temperature of crude oil in high levels for the distillation to happen. Before atmospheric distillation, crude is heated in a furnace. Aim of the study was the replacement of the furnace before ADU with HTHPs.
Three crude oil streams, to be heated, were subject of the source side and several distillation fraction streams were identified to provide heat at the source side.
The crude oil and distillation fraction streams were selected after a retrofit was applied to preheat network.

METHODS

HEAT PUMP INTEGRATION SCENARIOS

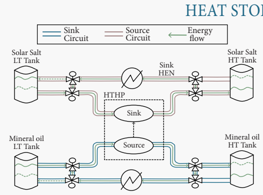


REVERSED BRAYTON CYCLE



- Large temperature lift leads to high pressure ratios. There is potential in mounting a turbine in the same shaft as compressor and recover work in the expansion process.
- The Internal Heat Exchanger (IHX) exists so that the working fluid is at the appropriate temperature levels to receive or extract heat at the source and sink respectively.
- R744 is utilized as refrigerant. Natural refrigerant with stable operation in transcritical conditions and good heat transfer properties.
- Coefficient of Performance is optimized with low and high pressure and degrees of superheat after expansion as decision variables.

HEAT STORAGE INTEGRATION



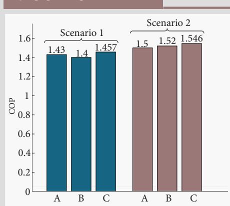
- Two tank storage configuration on both source and sink side with a low and high temperature tank.
- Applicable in Sub-Scenario A only where there are heat transfer mediums on each side.
- For heat storage integration, the heat pump should have capacity higher than the nominal requirements in heat demand.

ENERGY MIX OPTIMIZATION

OBJECTIVE → Maximization of feasibility index/Minimization of Levelized Cost of Heat (LCOH).

DETERMINATION → Optimum capacities of wind turbines, PVs, HTHPs and heat storage.

RESULTS

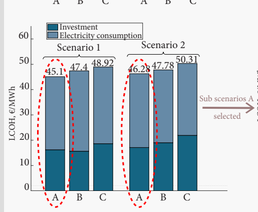


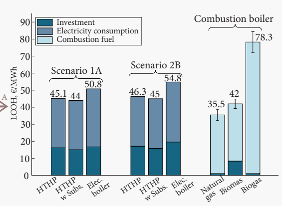
SCENARIO 1A

TECHNOLOGY	CAPACITY
HTHP	39.6 MW
Wind turbines	28 MW
Heat storage	117.8 MWh
Heat demand	34.4 MW
COP	1.429
Renewable Penetration	37%

SCENARIO 2A

TECHNOLOGY	CAPACITY
HTHP	16.96 MW
Wind turbines	10.5 MW
PVs	3.8 MW STC
Heat demand	16.96 MW
COP	1.5
Renewable Penetration	34%





For an optimized average COP, configurations with more heat pumps may provide a better COP.
Sub-Scenarios A, meaning only one heat pump, have the lowest LCOH and only those were further investigated. HTHPs were found to be economic superior to biogas and electric boilers, but demonstrated worse economic performance in comparison with biomass and natural gas.
The two scenarios designated a different optimal energy mix. Wind turbines were chosen in both cases. Heat storage is chosen as the ratio of heat demand to COP is getting larger. Photovoltaics consist a filler option.

CONCLUSIONS

Configurations with higher number of heat pumps introduced high investments and resulted in worse economic performance.

Wind turbines, PVs and heat storage are all eligible for application in HTHP project when aiming in feasibility index optimization.

HTHPs were designated to be superior to electric and biogas boilers, but the analysis showed them to be inferior to natural gas and biomass boilers. Expected future increases, though in both these fuels indicate that HTHP projects would be more viable in the future.

Performance analysis of a high temperature heat pump for compressed heat energy storage system using R-1233zd(E) as working fluid

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Keywords:

High temperature heat pump, R-1233zd(E), Numerical modelling, Compressed heat energy storage

Abstract

The current work studies numerically the performance of a high temperature heat pump (HTHP), which is a part of compressed heat energy storage (CHEST) system, adapting R-1233zd(E) as refrigerant. This work is performed under the framework of the European project CHESTER (www.chester-project.eu). The main goal of this project is to find an innovative and feasible way to store the surplus electricity from renewable energy sources (RES) in form of thermal energy, which can be used later to run a heat engine to produce electricity in high demand peaks. To do so, a HTHP is required to pump the heat from a low-temperature source to a high-temperature sink which, in this case, is a thermal energy storage (TES) system. The results are promising, for compressor speed= 1500 rpm, source temperature= 85 °C, and sink temperature= 133 °C the proposed HTHP can reach a coefficient of performance (COP_{HTHP}) of 4.86.

Introduction

for many decades and the world looks forward to harnessing the RES in feasible and efficient ways to replace the current fusel fuels that have negative impacts on the environment. Moreover, most fusel fuels face an accelerated depletion and near extinction. However, the main disadvantage of almost all RES is the intermittency. To try solving this, an ambitious European project was kicked off in the last year entitled **C**ompressed **H**eat **E**nergy **ST**orage for **E**nergy from **R**enewable sources (CHESTER) [1].

CHESTER project aims to develop an innovative CHEST system that allows managing, storing, and discharging of energy using different RES through the combination of electricity and heat sectors [1]. In this system a HTHP should be utilized to pump the energy from low-temperature sources, such as industrial waste heat, seasonal pit heat storage system, etc., to a high-temperature TES system using the electrical power from RES. In this early stage of the project, the first milestone is to design and test a CHEST system laboratory prototype with 10 kWe capacity. The present work comprises a preliminary design of CHEST system prototype to estimate the system capacities, characterization and selection of the HTHP's components, and, finally, a parametric study to assess the HTHP's performance for different compressor speeds and source temperatures.

Methods

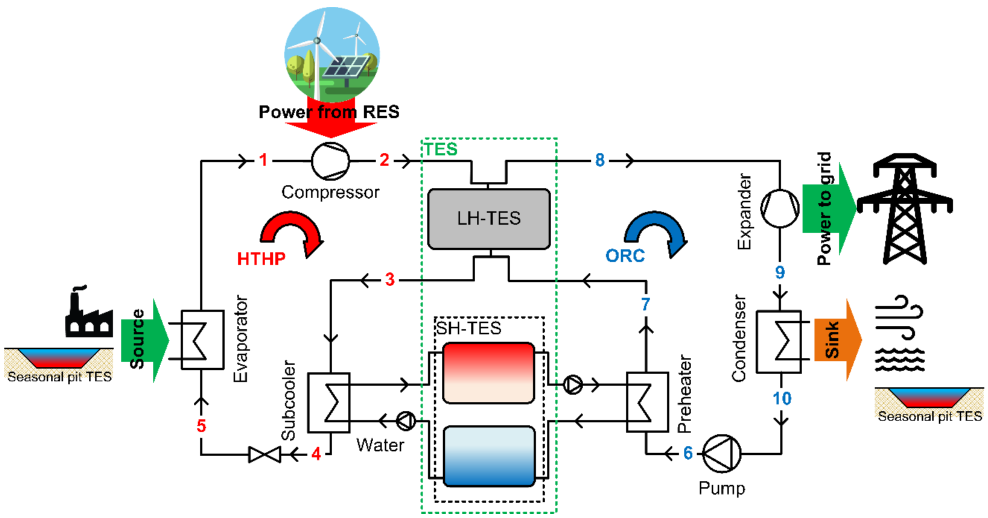

Figure 1. CHEST general concept.

Figure 1 shows the schematic of CHEST system where the HTHP is used to charge the TES system using the surplus power generated from RES. Later, in high demand periods, the stored thermal energy is converted to electrical power through an organic Rankine cycle (ORC). It can be noticed that the TES system consists of two sub-systems. The first sub-system is the latent heat thermal energy storage (LH-TES) and the second one is the sensible heat thermal energy storage (SH-TES).

The design process of the required HTHP for the CHEST system prototype passed through three main steps. As first step, the proposed CHEST system (Figure 1) was modelled using Engineering Equation Solver (EES) programme [2] to identify the required capacity for each component.

Based on the results of the EES-CHEST model, the second step was to size and select the main HTHP's components. The heat exchangers were sized using SWEP SSP G8 selection tool [3]. On the other hand, the required compressor was selected from Viking Heat Engines (VHE) catalogue [4].

The final step was to model the proposed HTHP using IMST-ART© simulation tool [5] and implement different parametric studies to assess the global performance and operating limits.

Results and Discussion

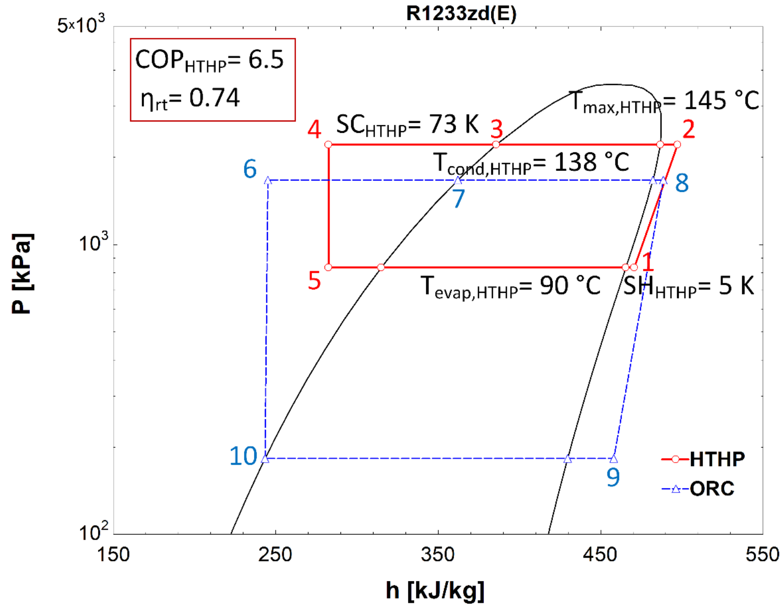

Figure 2. P-h diagram for CHEST system prototype using R-1233zd(E) for HTHP's source and ORC's sink temperatures of 100 °C and 25 °C, respectively.

EES-CHEST model's simulation (Figure 2) shows that the system can reach a roundtrip efficiency (η_{rt}) of 0.74 for HTHP's source and ORC's sink temperatures of 100 and 25 °C, respectively. The roundtrip efficiency is the ratio between the net output power from ORC to the total input power to HTHP. It is worth mentioning that R-1233zd(E) was utilized as working fluid in both HTHP and ORC. R-1233zd(E) was chosen based on recommendations of many authors in the literature [6], [7]. The following HTHP's components capacities were estimated:

- Evaporator capacity= 75.5 kW.
- Condenser capacity= 44.8 kW.
- Subcooler capacity= 41.4 kW.
- Compressor input power= 13.2 kW.

Regarding the estimated capacities, Table 1 summarizes the specifications for main selected components of HTHP used for the CHEST system prototype.

Table 1. Specifications of the main selected components for the HTHP prototype.

Component	Manufacturer	Selection tool	Main specifications
Compressor	VHE	VHE's catalogue	• Model: HBC-511 • No. of cylinders: 1 • Swept volume: 511 cm^3 • Speed: 500-1500 rpm
Evaporator	SWEP	SSP G8	• Model: V120T • No. of plates: 60 • Heat transfer area: 7.66 m^2
Condenser	SWEP	SSP G8	• Model: B200T • No. of plates: 106 • Heat transfer area: 13.4 m^2
Subcooler	SWEP	SSP G8	• Model: B86 • No. of plates: 62 • Heat transfer area: 3.6 m^2

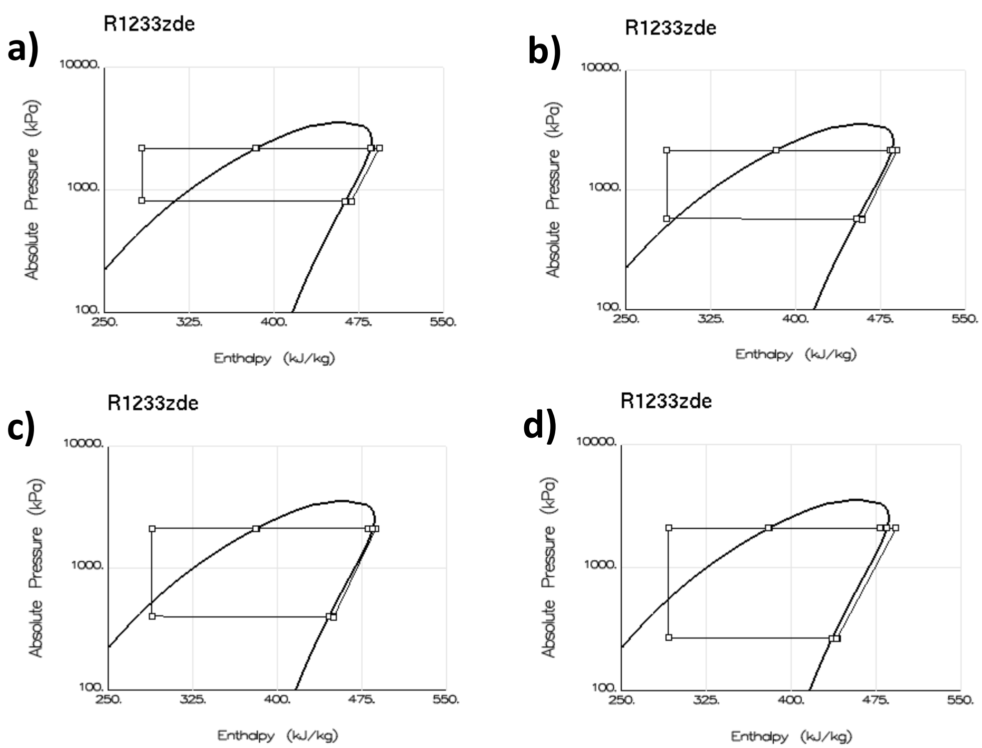

Figure 3. The HTHP's cycle on P-h diagram for compressor speed= 1500 rpm, sink temperature= 133 °C, and source temperatures of a) 100 °C, b) 85 °C, c) 70 °C, and d) 55 °C.

The size and specifications of these components were introduced as inputs to the IMST-ART© in order to estimate the global HTHP's performance for different compressor speeds and source temperatures. Figure 3 shows the P-h diagrams of the HTHP for source temperature ranges between 55 and 100 °C, fixed compressor speed of 1500 rpm, and sink temperature (PCM's melting temperature) of 133 °C.

Tables 2 and 3 summarize the IMST-ART© results of the HTHP prototype for compressor speeds of 500 and 1500 rpm, respectively. In this study the inlet and outlet water temperatures through the subcooler were fixed at 43.8 and 133 °C, respectively.

Table 2. HTHP's performance for compressor speed of 500 rpm and sink temperature of 133 °C.

Component	Source temperature [°C]	100	**85**	70	55
Compressor	Total Power input [kW]	4.49	**3.62**	2.73	2.00
	Ref. mass flow rate [kg/s]	0.14	**0.09**	0.06	0.03
Condenser	Capacity [kW]	15.71	**9.86**	5.92	3.26
	T_{cond} (bubble) [°C]	136.01	**135.40**	134.89	134.59
Subcooler	Capacity [kW]	13.05	**8.04**	4.67	2.27
	Total subcooling [K]	64.63	**61.81**	58.36	52.93
Evaporator	Capacity [kW]	25.58	**15.35**	8.66	4.13
	T_{evap} (dew) [°C]	91.79	**76.89**	62.75	48.42
Global Performance	Total heat provided [kW] (condenser + subcooler)	28.77	**17.90**	10.58	5.53
	COP_{HTHP} [-]	6.41	**4.95**	3.88	2.77

Table 3. HTHP's performance for compressor speed of 1500 rpm and sink temperature of 133 °C.

Component	Source temperature [°C]	100	**85**	70	55
Compressor	Total Power input [kW]	13.21	**10.43**	7.89	5.86
	Ref. mass flow rate [kg/s]	0.39	**0.25**	0.15	0.081
Condenser	Capacity [kW]	41.98	**26.66**	16.27	9.13
	T_{cond} (bubble) [°C]	137.83	**136.76**	135.92	135.36
Subcooler	Capacity [kW]	38.85	**24.02**	14.12	7.19
	Total subcooling [K]	70.50	**67.38**	64.34	61.54
Evaporator	Capacity [kW]	71.31	**43.17**	24.72	12.13
	T_{evap} (dew) [°C]	89.08	**74.70**	61.05	47.07
Global Performance	Total heat provided [kW] (condenser + subcooler)	80.83	**50.68**	30.40	16.33
	COP_{HTHP} [-]	6.12	**4.86**	3.85	2.79

Conclusions

- CHEST system is a promising and feasible technology for storing and managing the thermal energy. It also gives a better way for integrating the RES into electricity grid.
- R-1233zd(E) is a low-GWP, non-toxic, and non-flammable fluid with a critical temperature of 166 °C. These make R-1233zd(E) to be a potential candidate for HTHP sub-critical applications.
- Under the nominal conditions (compressor speed= 1500 rpm, source temperature= 85 °C, and sink temperature= 133 °C) of the proposed HTHP prototype, IMST-ART© estimated a COP_{HTHP} of 4.86 for total heat provided of 50.68 kW and input power of 10.43 kW.

Acknowledgment

This work has been partially funded by the grant agreement No. 764042 (CHESTER project) of the European Union's Horizon 2020 research and innovation program.

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Performance Analysis of a High Temperature Heat Pump for Compressed Heat Energy Storage System using R-1233zd(E) as Working Fluid

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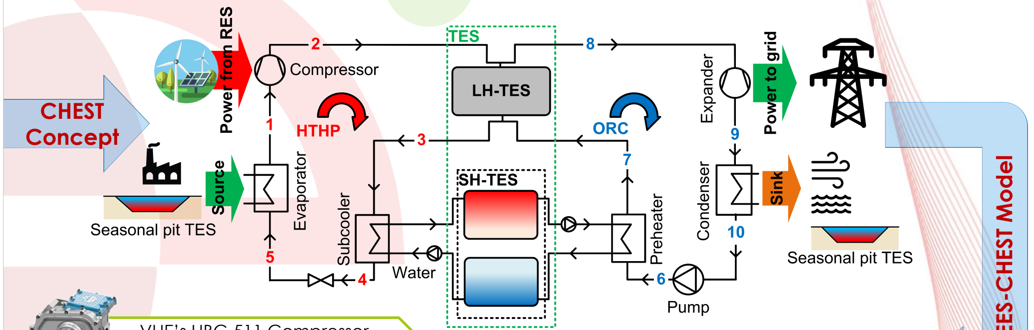

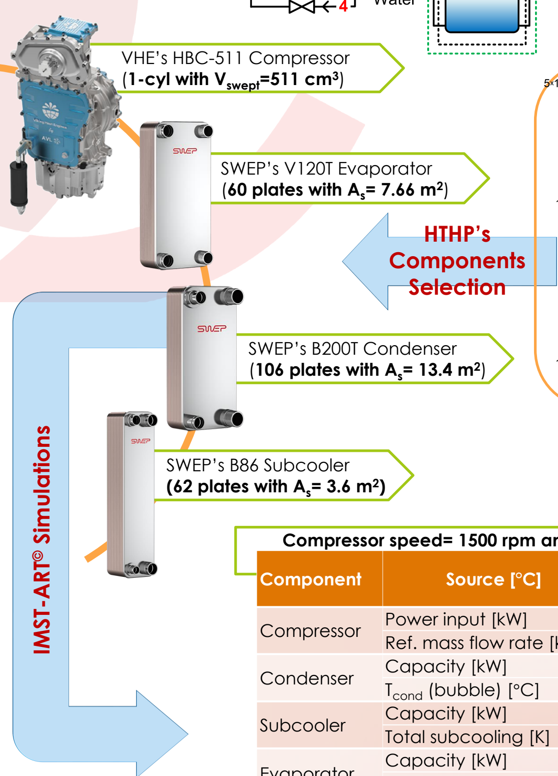

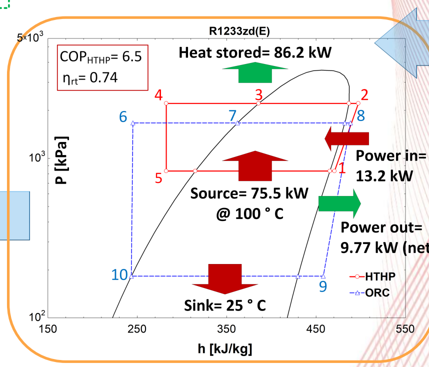

Compressor speed= 1500 rpm and HTHP's sink temperature= 133 °C

Component	Source [°C]	100	85	70	55
Compressor	Power input [kW]	13.21	**10.43**	7.89	5.86
	Ref. mass flow rate [kg/s]	0.39	**0.25**	0.15	0.081
Condenser	Capacity [kW]	41.98	**26.66**	16.27	9.13
	T_{cond} (bubble) [°C]	137.83	**136.76**	135.92	135.36
Subcooler	Capacity [kW]	38.85	**24.02**	14.12	7.19
	Total subcooling [K]	70.50	**67.38**	64.34	61.54
Evaporator	Capacity [kW]	71.31	**43.17**	24.72	12.13
	T_{evap} (dew) [°C]	89.08	**74.70**	61.05	47.07
Global Performance	Total heat provided [kW]	80.83	**50.68**	30.40	16.33
	COP_{HTHP} [-]	6.12	**4.86**	3.85	2.79

This work has been partially funded by the grant agreement No. 764042 (CHESTER project) of the European Union's Horizon 2020 research and innovation program.

Lubricant Investigation for High Temperature Heat Pump Application

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Abstract

Lubricant plays crucial role in safe and efficient compressor operation especially for high temperature application. Lubricant and refrigerant mixture viscosity analysis and behaviour during heat pump operation requires careful investigation. As a part of European funded CHESTER project, lubricant (oil) behaviour was investigated (visual) experimentally using a high temperature (100°C to 135°C) heat pump test rig developed at Ulster University. In addition, a separate test rig was developed to investigate lubricant oil (POE) and refrigerant (R1233zd(E)) viscosity analysis at source temperature in a range of 50°C to 110°C.

The results from HTHP test rig showed that oil temperature increased with source temperature and in order to protect compressor components and maintain suitable lubricity, additional cooling was required along with a comprehensive start-up, operation and shut-down strategy. The viscosity results of R1233zd(E) and POE mixtures were obtained in terms of a Daniel chart (w% of refrigerant in the oil) in a temperature range of 40°C to 100°C. The viscosity variation showed a range of 80 cSt to 8 cSt between 60°C to 90°C suction temperatures. The analysis also calls for further investigation with other oils and a development of high viscosity grade lubricant suitable for R1233zd(E) and other refrigerants suitable in high temperature heat pump application.

Keywords:

High temperature heat pump, Lubricant, oil, viscosity, compressor cooling

1 Introduction

Energy efficiency and waste heat recovery in industrial sector has a huge potential of 370 TWh (waste heat) per year in Europe alone [1]. High temperature heat pump (HTHP) could provide energy/carbon emission saving for heating/cooling and integration in existing process or with thermal energy storage or district heating/cooling network could provide further flexibility required for demand side management. There are few commercial products available in the market where maximum temperature of 165°C is achievable with source temperature in a range of 35° to 70°C [2]. However, HTHP still possess some challenges due to high source/sink temperature, new refrigerant and requires special attention to system components, cooling and lubrication. There are limited investigation using alternative or low GWP/ODP refrigerant in HTHP application such as R1233zd(E) [3]. Refrigerant and lubricant/oil compatibility also plays critical role in terms heat transfer, fluid flow and hence, in overall efficiency. Moreover, oil and refrigerant viscosity varies significantly at suction pressure and temperature and is a less investigate area of research for HTHPs. As a part of EU funded-CHESTER project, HTHP and oil-refrigerant viscosity test-rig were developed at Ulster University to understand oil temperature behaviour and to measure oil-viscosity mixtures at CHESTER test conditions.

2 Methods

In order to assess the behaviour of oil temperature, mixtures in compressor and oil cooling requirements, a HTHP test-rig was developed at Ulster University. The heat pump was designed at Tcon=125°C and Tevp= 50°C with SH=20K (evaporator + liquid-suction heat exchanger) and SC=9K. Further details about test set-up can be found in [3]. After initial tuning, the system was operated using R245fa as a reference at a fixed evaporation temperature (e.g. 50°C) and varying condensing temperature between 85°C to 125°C. Polyester oil HARP POE68 was recommended by the supplier for higher temperature applications viscosity ranges from 65.5 cSt at 40°C to 9.3 cSt at 100°C. Oil temperature and other parameters were measured experimentally whereas oil and refrigerant level was observed visually for analysis purposes.

A separate experimental test bed was constructed to measure the properties of lubricant-refrigerant mixtures. The test bed was based on two separate units, one was a refrigerant storage vessel and the other was a single continuous loop for the circulation of lubricant and lubricant-refrigerant mixtures. Two thermal baths were used to conduct the experiments which has an operational range of -20°C to 180°C. Most of the components of this test-rig was bespoke designed and a viscometer was used to measure the dynamic viscosity of oil-refrigerant compositions with a precision of ± 2% deviation. Reference oil calibration and density of oil/mixtures was measured using a mass flow meter. Viscosity assessment was carried out for R1233zd(E) and POE 320 between a temperature range of 40°C to 105°C at different concentration levels. However, for CHESTER project the focus was to determine oil-refrigerant viscosity at maximum suction temperature (e.g. 90°C). A DT85 datalogger was used to record parameters such flow rate, temperature, pressure, density and power. Data was measured at an interval of 30s using two data acquisition system and stored in a dedicated PC for data analysis purpose.

3 Results and Discussion

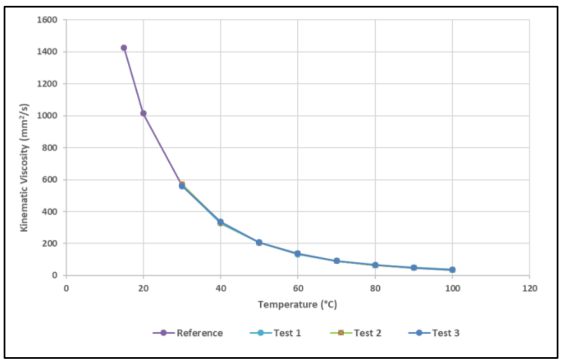
Analysis for oil temperature in HTHP and viscosity of oil-refrigerant mixture was measured using test-rigs as shown in Figure 1. It was evident form HTHP operation that the oil temperature increased to 90°C while operating at 105°C condensing temperature. Hence, the test rig was modified to accommodate oil cooling to maintain temperatures with the range of 60-80°C when operating at condensing temperature above 100°C as certain viscosity is crucial for compressor operation. Further details on additional cooling and temperature rise can be found in [3].

Figure 1 Test rig used for lubricant analysis at Ulster University: HTHP (left), viscosity measurement (right)

The main study focused on viscosity analysis where investigations were carried out on the second test rig with pure oil POE 320 as a reference. Figure 2 shows pure oil viscosity measurement between 20°C and 100°C. Additional visual analysis at 20°C showed that the oil possesses entrainment of bubbles (perhaps due to gear pump) but disappears at 100°C.

The initial tests looked at validating the operation of the test rig to manage the introduction of refrigerant to the lubricant loop. These tests ran across the range of temperatures from 30°C to 100°C introducing refrigerant at 10°C lower temperature with associated pressure. The results (Figure 3) show viscosity vs temperature constantly reducing to 6 cTs at 100°C. The first test was done with pure lubricant oil the second with lubricant oil that had refrigerant recovery completed. In the repeat test it was clear that the lubricant oil properties had changed slightly when combined with the refrigerant, however this would take place under standard operating conditions where the mass concentrations would be in continuous adjustment.

Figure 4 shows kinematic viscosity of POE320 and R1233zd(E) mixture at 10%, 20% and 30% concertation in the form of a Daniel chart. With 10% R1233zd(E) in mixtures, it provided viscosity in a range of 15 to 140 cSt whereas around 8 to 83 cSt and 6 to 49 cSt at 20% and 30% R1233zd(E) respectively. However, test set-up was designed for dynamic operation and it is difficult to obtain exact amount of mixture for repeatability purposes and pure oil (three tests) average standard deviation of 2.4 cSt is taken as a reference for repeatability purpose and its comparison with manufacture data provided ±0.5% error.



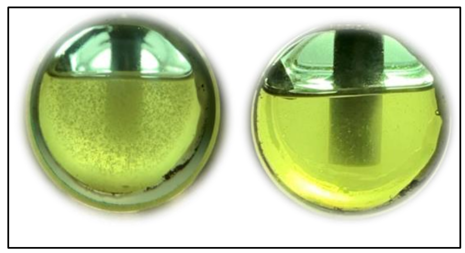


Figure 2 POE 320 (pure oil): a.) viscosity (left), b.) oil visuals (right): @20°C (left) and 100°C (right)

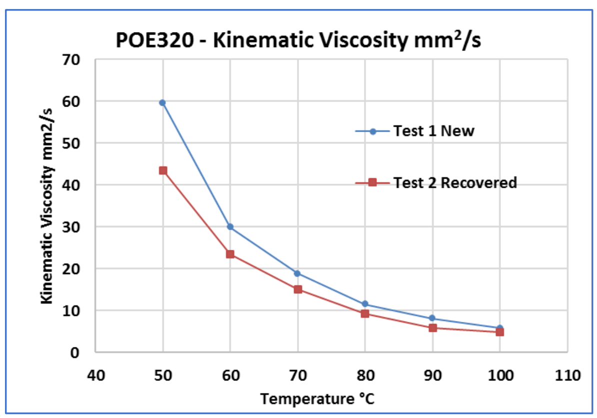


Figure 3 R1233zd(E) and POE 320 viscosity variation with temperature

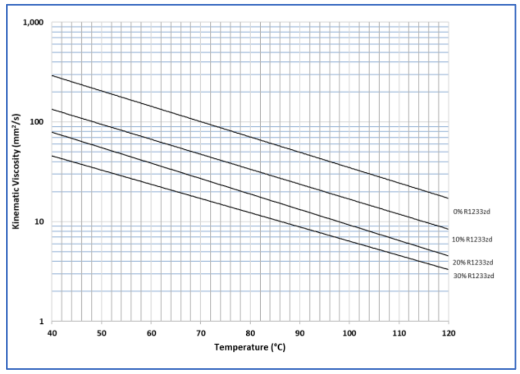


Figure 4 R1233zd(E) and POE320 mixture viscosity (Daniel Chart)

4 Conclusion

Oil interaction and viscosity plays a crucial role for safe operation of a compressor especially for HTHP applications as viscosity of oil decreases with temperature. Test results from the HTHP clearly emphasised the requirement of EEV and oil cooling ensuring longevity of compressor and EEV body/motor. However, a trade-off between cooling and heat/efficiency loss must be considered. Due to high temperature and high viscosity of the lubricant, it is important to have a defined start-up, operation and cool down strategy, which involves pre-heating of the oil in order to avoid sudden migration of refrigerant to the compressor.

The test results from POE320 with R1233zd(E) mixture indicated that there may be justification in moving to the higher viscosity lubricant. POE320 lubricant oil could meet in part the criteria set out by the compressor manufacture if temperature is maintained around 90°C and if temperature can be maintained up to 65°C then it can work with up to 30% concentration. However, it is unclear without further clarification of the refrigerant concentrations and pressures in the HTHP sump during expected operational conditions, whether the lubricant will be suitable. In addition, further clarification and additional testing and cooling strategy would be investigated as a part of on-going research.

5 Acknowledgement

The authors gratefully acknowledge the support of the European Unions' Horizon 2020 research and innovation programme through the Compressed Heat Energy Storage for Energy from Renewable Sources (CHESTER) project (Grant No. 764042).

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Lubricant Investigation for High Temperature Heat Pump Application

ABSTRACT

Lubricant plays a crucial role in safe and efficient compressor operation, especially for high temperature applications. Lubricant and refrigerant mixture viscosity analysis and behaviour during heat pump operation requires careful investigation. As a part of European funded CHESTER project, lubricant oil behaviour was investigated visually and experimentally using a high temperature (100°C to 135°C) heat pump test rig developed at Ulster University. In addition, a separate test rig was developed for lubricant oil (POE) and refrigerant R1233zd(E) viscosity analysis, to investigate viscosity of mixture at source temperature in a range of 40°C to 100°C

The results from HTHP test rig showed that oil temperature increases with source temperature and in order to protect compressor components and maintain suitable lubricity, additional cooling would be required along with start-up, operation and shut-down strategies. The viscosity results of R1233zd(E) and POE mixtures were obtained and plotted on a 'Daniel chart' (w% of refrigerant in the oil) in a temperature range of 40°C to 100°C. The viscosity variation showed a range of 8cSt to 80cSt between 60°C to 90°C suction temperatures and requires careful consideration for compressor manufacture recommendation. The analysis also calls for further investigation with other oils and development of a high viscosity grade lubricant suitable for R1233zd(E) and other refrigerant suitable for high temperature heat pump applications.

OBJECTIVES

1) Identification of refrigerant oil mixtures suitable for use in HTHP.
2) Design and build a bespoke test bed to measure the viscosity of lubricant and refrigerant mixes.
3) Conduct testing on POE 320 lubricant and R1233zd(E) refrigerant mixes and carry out analyse on the test results.
4) Determine the suitability of lubricant and refrigerant combination to the meet requirements of the HTHP within the CHESTER project.

BACKGROUND

A HTHP test-rig (Fig-1) was developed at Centre for Sustainable Technologies (CST) at Ulster University to assess the behaviour of lubricant oil in the compressor and cooling requirements. The heat pump was designed at Tcon=125°C and Tevp=50°C with superheat=20K (evaporator + liquid-suction heat exchanger) and subcooling= 9K. During testing a number of issues were encountered at temperatures >90°C which required modifications to equipment and lubricant oil cooling systems.

Fig-1: HTHP test bed

Increased fluid temperatures observed within the sump, highlighted the need for managed cooling of the lubricant oil within the crankcase. These increased temperatures of the bulk lubricant oil refrigerant mixture presented the possibility of reduced kinematic viscosity causing increased wear to mechanical components and reduced overall compressor efficiency. It is difficult to determine the characteristics of lubricant oil refrigerant interaction within a working compressor, therefore dedicated testing using a specialised test bed and equipment is required. Testing conducted with bespoke apparatus also provides the ability to measure the kinematic viscosity across a range of lubricant oil and refrigerant compositions with increased control of temperature and pressure set-points.

METHODS

- **Experimental Study** (Viscosity testing)

A test bed (Fig-2) was designed and built in Ulster University to test the viscosity of lubricant oil refrigerant compositions. The test bed was used to cover a range of temperatures 40°C to 100°C and was split into a fluid circulation loop and refrigerant charging station. This separation provided a simple platform to ensure precise measurements of refrigerant charge (% mass) into the fluid circulation system. The main equipment (Fig-3) used was a viscosimeter, mass flow meter, gear pump with inverter, thermal baths and a laboratory scales.

Reference data for the lubricant oil was provided by the OEM in relation to density and viscosity. Thermodynamic properties for the refrigerant R1233zd(E) was obtained from REFPROP V9.1

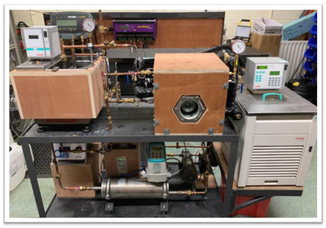

Fig-2: Test bed

An experimental test procedure was developed through initial operation and calibration of the test bed. This procedure was used to perform repeat testing within clearly defined steps.

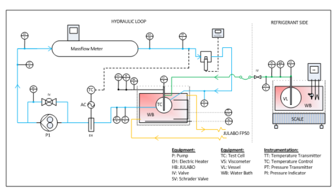

Fig-3: Viscosity test bed flow diagram

RESULTS

Testing Lubricant Oil

Initial testing was completed to compare reference data to lubricant oil tested (Fig-4, example POE 320), specifically for kinematic viscosity (mm²/s) and density across the range 40°C to 100°C. This testing corresponded well to reference data, providing a baseline for testing refrigerant lubricant oil compositions.

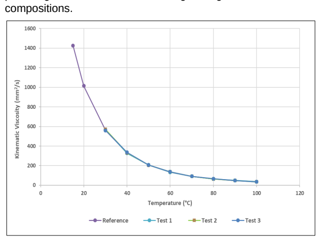

Fig-4: Pure lubricant oil viscosity validation testing

RESULTS

Reniso SEZ 320 and R 1233zd(E) mixtures

Initial tests were completed to validate the operation of the test bed and equipment. These tests were conducted between 50°C and 100°C with refrigerant introduced at 10°C lower temperature and associated pressure. Tests were conducted with pure oil and repeated with oil that had refrigerant recovery completed. These ramp-up test results show viscosity reducing with increasing temperature to 6 cTs at 100°C.

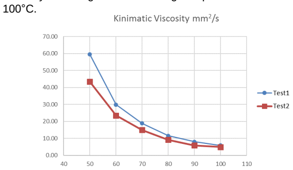

Fig-5: Test bed

Specific tests were carried out at 90-100°C split at 7.3bar gauge pressure representing the HTHP operational parameters. These results showed a kinematic viscosity of 8 cTs, which corresponded to previous tests.

Table-1: Testing 90\100°C Split

Test No	Temperature Ref (°C)	Temperature Ref_Oil (°C)	Pressure (bar_g)	Density Mix (kg/m³	Dyn Viscosity (mPas)	Kin Viscosity (cSt)
Test 1	90	100	7.30	1001	8.10	8.09
Test2	90	100	7.30	997	8.40	8.43

Composition tests between 10% and 30% by mass of R1233zd(E) across a temperature range of 40°C to 105°C were completed. These results were plotted on a 'Daniel chart' and exponential trendlines created to show regular lines graphically, which were extrapolated to 120°C. The results clearly show the upper working limit 6 to 49 cSt was reached at approximately 30% concentration.

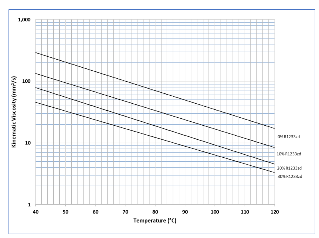

Fig-6: Daniel chart

CONCLUSION

Results indicate that POE320 lubricant used in conjunction with R1233zd(E) will provide increased lubricity at higher operational parameters up to 10% concentration and up to 30% concetation with adequate cooling. However, viscosity rating is determined by compressor manufacturer and further testing of lubricant oils with higher viscosity index is required to comprehensively assess suitability in achieving operational conditions for the HTHP.

FUTURE WORK

Testing refrigerant lubricant oil compositions

- Repeat testing to generate a complete mapping across pressure, viscosity and temperature
- Test lubricants with higher viscosity indexes.

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High-Temperature Refrigeration System for Cooling of Automotive PEM Fuel Cells

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Keywords:
High temperature heat pump, High temperature refrigeration system, R-1234ze(Z), R-600, automotive cooling system, PEM fuel cell

Abstract

High-temperature heat pumps (HTHP) are usually utilized for waste heat recovery in order to reduce primary energy consumption. However, a similar system can also be used to provide cooling capacity on a high temperature level. An example where high temperature cooling is needed is a fuel cell electric vehicle (FCEV). Due to the relatively low working temperature of PEM fuel cells (approx. 70 – 90 °C) and therefore low exhaust gas energy, most of their waste heat has to be dissipated by a cooling system at low temperature differences to the ambient. A state of the art liquid cooling system can limit the fuel cell power due to insufficient cooling capacity, especially at high ambient temperatures. A HTHP, or more precisely a high-temperature refrigeration system (HTRS), can be used to provide additional cooling capacity by decoupling of the fuel cell working temperature from the heat rejection temperature.

The aim of this work is the experimental and numerical investigation of a HTRS, build up solely with standard components of automotive R-134a or R-1234yf refrigerant systems. The natural refrigerant R-600 (n-Butane) and the synthetic refrigerant R-1234ze(Z) were identified as the most promising candidates for the working fluid. The focus for the selection was on high volumetric cooling capacity (VCC) due to the expected high cooling loads and therefore high volumetric flow rates.

A test rig with a HTRS composed of standard automotive components was set up in two steps: first a smaller version with cooling loads up to 10 kW which was then enhanced to cooling loads up to 40 kW. At the smaller test rig both refrigerants, R-600 and R-1234ze(Z), were investigated, whereas at the enhanced version only the investigation of R-600 was possible. Both refrigerants showed similar performance at the smaller test rig, with slightly higher COPs and VCCs for R-600 at the same operating conditions. For the enhanced test rig COPs in the range of 14 – 4 were measured for cooling loads of 7 – 25 kW. Due to high pressure losses in the condenser the cooling capacity of the system was limited, showing the need to adapt at least some of the components to the higher volume flow rates.

Despite the practical limitations of the test rig in this work the usage of a HTRS is identified as a promising solution to overcome the issue of limited cooling capacity in FCEVs. In principal it is possible to build up such a system based on standard automotive refrigeration components with small adaptions. Optimisation possibilities of the system and the interactions with the fuel cell and liquid cooling system of an FCEV are currently being investigated using detailed physical simulation models.

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High-Temperature Refrigeration System for Cooling of Automotive PEM Fuel Cells

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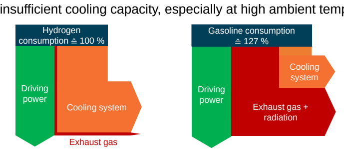
Introduction

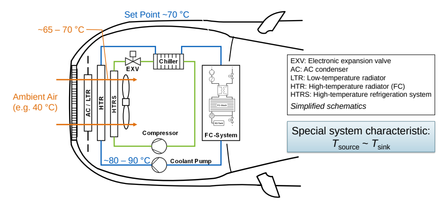
PEM fuel cells (PEMFC) provide several advantages for the application in vehicles, e.g. high efficiency, short refueling time and local emission free operation. Due to their low working temperature of (approx. 70–90 °C) and therefore low exhaust gas energy, most of the waste heat has to be dissipated by a cooling system at low temperature differences to the ambient. A state of the art liquid cooling system can limit the available fuel cell power due to insufficient cooling capacity, especially at high ambient temperatures.



Comparison of energy flows in a vehicle with PEM fuel cell and internal combustion engine, values refer to full load operation (adapted from [1]).

Objective

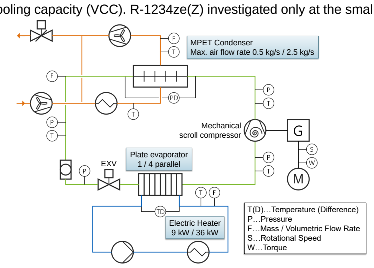
- Can a high-temperature refrigeration system (HTRS) provide additional cooling capacity in a fuel cell electric vehicle (FCEV)?
- Can a HTRS be build solely with automotive HVAC components?



Simplified schematic of the investigated cooling concept for FCEVs including a high-temperature refrigeration circuit to provide additional cooling capacity in critical driving conditions.

Experimental Setup

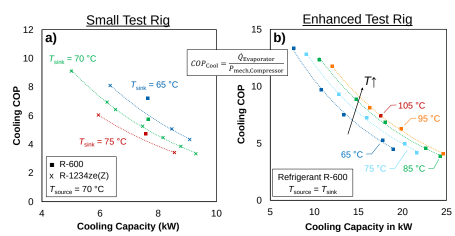
- HTRS test rig build up with automotive R-134a/R-1234yf components
- Secondary loops for heat source (coolant) and heat sink (air)
- Two versions of test rig (small & enhanced) as indicated in P&I diagram
- The natural refrigerant R-600 (n-Butane) and synthetic refrigerant R-1234ze(Z) were selected based on theoretical COP and volumetric cooling capacity (VCC). R-1234ze(Z) investigated only at the small test rig.



Simplified P&I diagram of the laboratory high-temperature refrigeration systems. Main differences between the small and enhanced test rig are indicated in the blue boxes.

Results

- Both refrigerants showed similar performance in the small test rig, with slightly higher COPs and VCCs for R-600.
- For the enhanced test rig the pressure drop of the condenser was the limiting factor for maximum cooling capacity.
- No negative influence on the components was observed due to the operation at elevated temperatures.



a) Comparison of COPs for refrigerant R-600 and R-1234ze(Z) at the small test rig depending on temperature lift and b) COPs for R-600 at the enhanced test rig for a fixed temperature lift of 0 K and varying heat source and sink temperatures. (Lines are only guide for the eye.)

Conclusion

- A high-temperature refrigeration system can significantly increase the available cooling capacity in a FCEV at moderate power consumption.
- In principal it is possible to build up such a system with standard automotive HVAC components. However, an adaption of this components to higher volumetric flow rates and to higher temperatures is advisable.
- Both investigated refrigerants, R-600 (n-Butane) and R-1234ze(Z) showed similar performance with slightly higher COPs and VCCs for R-600.

Outlook

In addition to the experimental studies a detailed physical model of the laboratory HTRS was developed. It is currently being used to investigate

- the interaction of the HTRS with the fuel cell and liquid cooling system in a virtual FCEV model and
- possible optimizations of the HTRS on component as well as on overall system level.

Possible cases of application include passenger cars and commercial vehicles up to heavy duty trucks.

References

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Development of a high temperature heat pump prototype with scroll compressor for industrial waste heat recovery

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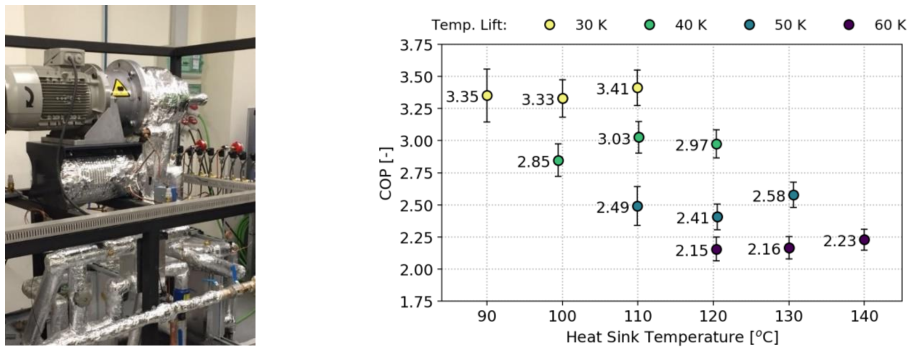
Keywords: High-temperature heat pump, scroll compressor, prototype, low-GWP refrigerants

Introduction

High-Temperature Heat Pumps (HTHPs) are a promising energy conversion technology that can substitute fossil fuels boilers and contribute to improve the overall efficiency of the energy-intensive industry sector and advance in the decarbonisation process. This sector demands innovative sustainable energy systems to meet the targets of the Paris Agreement for the climate change mitigation and reduce the predicted global temperature increase. This work presents the first experimental results of a novel HTHP prototype with scroll compressor for low-grade waste heat revalorization. This prototype is designed to reach heat sink temperatures above 140 °C while keeping its reliability and efficiency. Although it has been designed to be compatible with most of the synthetic low GWP refrigerants with high critical temperature, this work shows the results of the experimental tests with HFC-245fa that are going to be used as a reference for future refrigerant drop-in replacements. Moreover, exergy analysis is included to find the system improvement possibilities along with a semi-empirical alternative low-GWP refrigerants assessment and environmental evaluation of the HTHP integration in a waste heat recovery system. The results of this study may provide guidelines for the further design and development of HTHPs for low-grade waste heat recovery.

Results and Discussions

The experimental prototype, shown in Fig. 1a, was developed at the ISTENER laboratories of the Universitat Jaume I (Castelló de la Plana, Spain). The test bench is composed of the main vapour compression circuit, and two closed secondary circuits in which the fluid is a thermal oil, one for the heat source and the other for the heat sink along thermally connected to another water cooler closed circuit. The heat source circuit simulates the potential low-grade waste heat available from typically found industrial processes and the heat sink, the high-temperature demands.

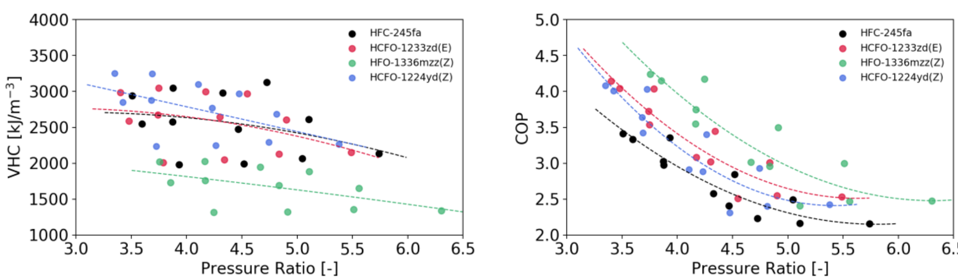


(a) (b)

Fig. 1. (a) Photo of the experimental prototype and (b) experimental COP.

Although the compressor power consumption and heating capacity increase as the heat sink temperature increases, the specific enthalpy difference at the compressor presents an increase of the specific enthalpy differences at the condenser. Thus, the refrigerant mass flow rate has no direct effect on the COP; rather, the major effect is caused by the compression ratio and therefore, the temperature lift, as shown in Fig. 1b. It is observed that the available waste heat and the heat demand conditions have a great influence on the energy efficiency of the HTHP system. Hence, the highest COP, with a value of 3.41 ± 0.1), is achieved at a 110 °C heat sink temperature and 80 °C heat source temperature. Nevertheless, the most interesting performance value for this application is achieved at heat source temperature of 80 °C and heat sink temperature of 140 °C with a COP value of 2.23.

To provide more in-depth knowledge of the potential of alternative low-GWP refrigerants, a semi-empirical simulation was carried out. HCFO-1224yd(Z), HCFO-1233zd(E), and HFO-1336mzz(Z), were selected because of their similarities with the traditional HFC-245fa. Results are presented in Fig. 2 where it can be observed that the alternative refrigerants have comparable trends to the HFC-245fa. While HCFO-1224yd(Z) and HCFO-1233zd(E) have similar behaviour to HFC-245fa, HFO-1336mzz(Z) shows a different trend, and hence a worse adaptation to the test setup can be supposed.



(a) (b)

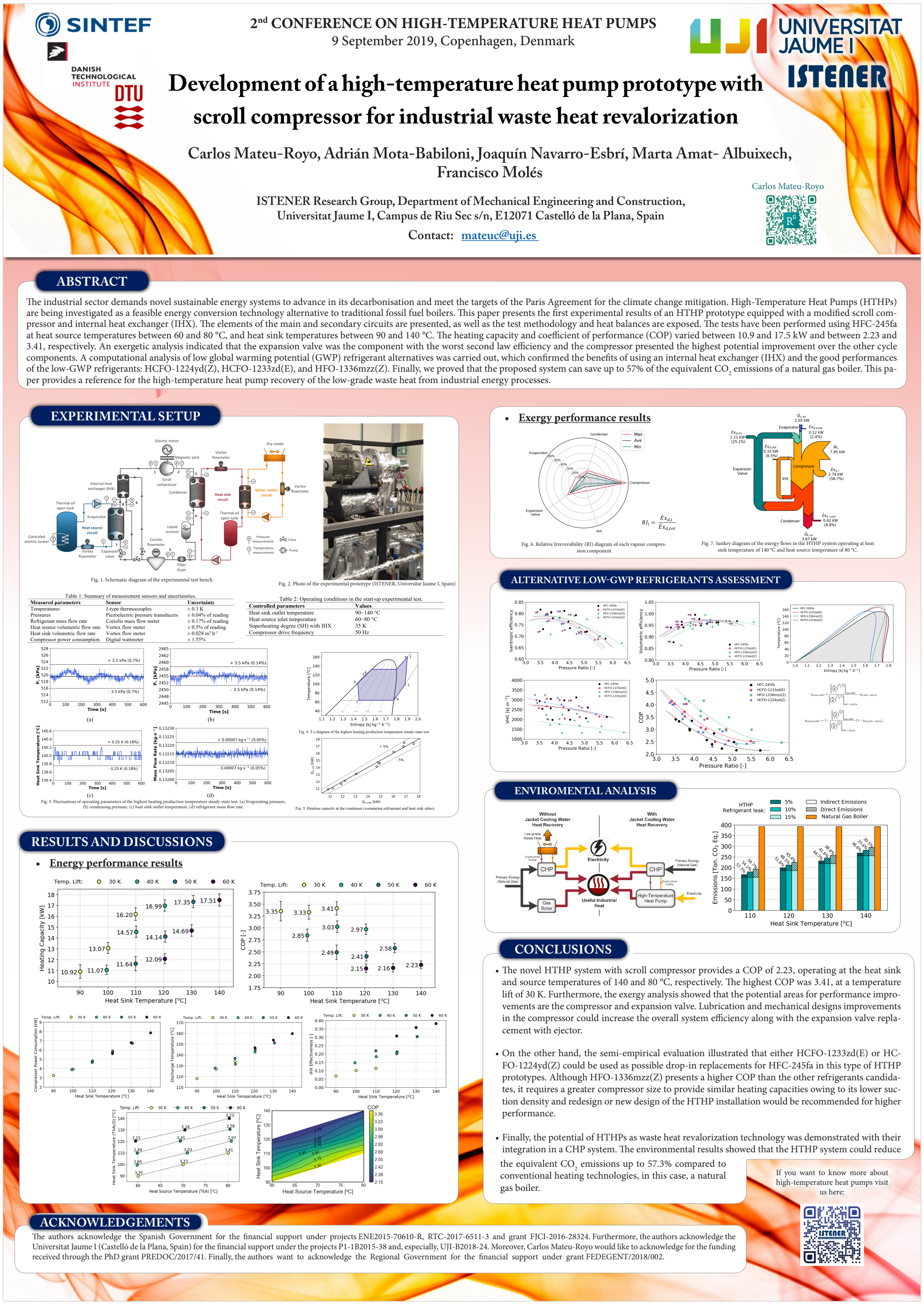
Fig. 2. Estimated performance parameters for alternative low-GWP refrigerants: (a) Volumetric heating capacity (VHC) and (b) COP.

Conclusions

The novel HTHP system with scroll compressor provides a COP of 2.23, operating at the heat sink and source temperatures of 140 and 80 °C, respectively. The highest COP was 3.41, at a temperature lift of 30 K. Furthermore, the exergy analysis showed that the potential areas for performance improvements are the compressor and expansion valve. Lubrication and mechanical designs improvements in the compressor could increase the overall system efficiency along with the expansion valve replacement with ejector.

On the other hand, the semi-empirical evaluation illustrated that either HCFO-1233zd(E) or HCFO-1224yd(Z) could be used as possible drop-in replacements for HFC-245fa in this type of HTHP prototypes. Although HFO-1336mzz(Z) presents a higher COP than the other refrigerants candidates, it requires a greater compressor size to provide similar heating capacities owing to its lower suction density and redesign or new design of the HTHP installation would be recommended for higher performance.

Finally, the potential of HTHPs as waste heat revalorization technology was demonstrated with their integration in a CHP system. The environmental results showed that the HTHP system could reduce the equivalent CO_2 emissions up to 57.3% compared to conventional heating technologies, in this case, a natural gas boiler.



2nd CONFERENCE ON HIGH-TEMPERATURE HEAT PUMPS
9 September 2019, Copenhagen, Denmark
Development of a high-temperature heat pump prototype with scroll compressor for industrial waste heat revalorization
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ABSTRACT
The industrial sector demands novel sustainable energy systems to advance in its decarbonisation and meet the targets of the Paris Agreement for the climate change mitigation. High-Temperature Heat Pumps (HTHPs) are being investigated as a feasible energy conversion technology alternative to traditional fossil fuel boilers. This paper presents the first experimental results of an HTHP prototype equipped with a modified scroll compressor and internal heat exchanger (IHX). The elements of the main and secondary circuits are presented, as well as the test methodology and heat balances are exposed. The tests have been performed using HFC-245fa at heat source temperatures between 60 and 80 °C, and heat sink temperatures between 90 and 140 °C. The heating capacity and coefficient of performance (COP) varied between 10.9 and 17.5 kW and between 2.23 and 3.41, respectively. An exergetic analysis indicated that the expansion valve was the component with the worst second law efficiency and the compressor presented the highest potential improvement over the other cycle components. A computational analysis of low global warming potential (GWP) refrigerant alternatives was carried out, which confirmed the benefits of using an internal heat exchanger (IHX) and the good performances of the low-GWP refrigerants: HCFO-1224yd(Z), HCFO-1233zd(E), and HFO-1336mzz(Z). Finally, we proved that the proposed system can save up to 57% of the equivalent CO2 emissions of a natural gas boiler. This paper provides a reference for the high-temperature heat pump recovery of the low-grade waste heat from industrial energy processes.
EXPERIMENTAL SETUP
Fig. 1. Schematic diagram of the experimental test bench
Fig. 2. Photo of the experimental prototype (ISTENER, Universitat Jaume I, Spain)
Table 1: Summary of measurement sensors and uncertainties.
Table 2: Operating conditions in the start-up experimental test.
Exergy performance results
Fig. 6. Relative Irreversibility (RI) diagram of each vapour compression component
Fig. 7. Sankey diagram of the exergy flows in the HTHP system operating at heat sink temperature of 140 °C and heat source temperature of 80 °C.
ALTERNATIVE LOW-GWP REFRIGERANTS ASSESSMENT
ENVIROMENTAL ANALYSIS
RESULTS AND DISCUSSIONS
Energy performance results
Heat Sink Temperature [°C]
Heating Capacity [kW]
COP [-]
CONCLUSIONS
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Finally, the potential of HTHPs as waste heat revalorization technology was demonstrated with their integration in a CHP system. The environmental results showed that the HTHP system could reduce the equivalent CO2 emissions up to 57.3% compared to conventional heating technologies, in this case, a natural gas boiler.
If you want to know more about high-temperature heat pumps visit us here:
ACKNOWLEDGEMENTS
The authors acknowledge the Spanish Government for the financial support under projects ENE2015-70610-R, RTC-2017-6511-3 and grant FJCI-2016-28324. Furthermore, the authors acknowledge the Universitat Jaume I (Castelló de la Plana, Spain) for the financial support under the projects P1-1B2015-38 and, especially, UJI-B2018-24. Moreover, Carlos Mateu-Royo would like to acknowledge for the funding received through the PhD grant PREDOC/2017/41. Finally, the authors want to acknowledge the Regional Government for the financial support under grant FEDEGENT/2018/002.

High temperature heat pump in a Swiss cheese factory

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Keywords:
High temperature heat pump, HFO, district heating, data centre, cheese factory, energy savings

Extended Abstract

The Swiss cheese factory in Gais Appenzell processes almost 10 million litres of milk per year and produces various semi-hard and mountain cheese specialities, as well as raclette cheese. Around 60 milk suppliers from the Appenzellerland region supply the milk.

Next to the cheese factory is the new data centre of Eastern Switzerland, which offers the highest levels of energy efficiency and security. By cooling the computer servers, the data centre produces waste heat of around 1.5 MW at 20 °C, which is fed into a local district-heating network.

A high temperature heat pump (Ochsner type: IWWHS 570 ER6c2) in the mountain cheese factory is connected to the district heating network and transforms parts of the heat into process heat at temperatures levels of up to 100 °C. This way, the cheese factory is able to replace the energy of around 1.5 million kWh of natural gas per year.

The process heat produced by the heat pump is temporarily stored in a stratified storage tank from where the individual processes in the cheese production (e.g. for cheese vats, cleaning water, multi-purpose heater, and pasteurisation) are supplied with heat. The lower heat levels of the storage tank are used for hot water heating and space heating (e.g. for cheese storage house).

The high temperature heat pump provides approximately 520 kW heating capacity at 100% part load. Low GWP HFO refrigerant R1234ze(E) (GWP_{100} of 6) is applied as an alternative to R134a (GWP_{100} of 1'430). The economizer cycle of the heat pump with vapor injection into a two-stage screw compressor is an efficient solution for high temperature lifts as part of the condensed refrigerant is expanded to a medium pressure level and is evaporated to saturation by subcooling the remaining condensate. This way, the economizer cycle enables:

1. high refrigerant mass flow at compressor outlet, resulting in high heating capacity (i.e., even at high temperature lifts and low evaporation temperatures),
2. reduced compressor outlet temperature, which is positive with regard to the compressor temperature limits, and
3. strong subcooling of the condensate to increase the COP.

Depending on the operating conditions, the COP of the heat pump varies between 2.55 and 2.85 at 74 K temperature lift (W18-14/W82-92) and between 3.75 and 4.20 at 47 K lift (W18-14/W55-65).

This case study in the small Swiss village of Gais shows how large amounts of heat can be transferred across industries (waste heat from a computer centre to a cheese factory). It is hoped that such synergies for heating and cooling will also be recognised at other locations in order to further decarbonise the industry.

Reference: An extended version of this case study has been published in the HPT Magazine 2/2019
Arpagaus, C.: From Waste Heat to Cheese, HPT Magazine, Vol. 37, No. 2, 2019 (Newsletter), (Article) (HPT Magazine)

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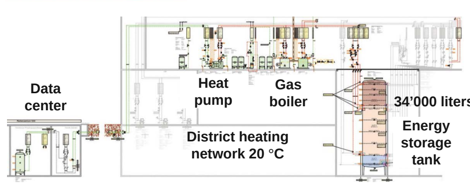
High Temperature Heat Pump in a Swiss Cheese Factory

From Waste Heat to Cheese

Application

The **mountain cheese factory in Gais Appenzell** transforms **waste heat at 20 °C from the neighboring data center** into **process heat of up to 100 °C** using a **high temperature heat pump** in order to heat and process the milk for cheese production.
This **saves** the mountain cheese factory around **1.5 million kWh of natural gas per year**.



Cheese factory in Gais Appenzell

- Energy demand approx. 1'800 MWh per year
- Approx. 10 mio. liters of milk per year (60 milk suppliers)
- **Approx. 300 tons of cheese per year**

Temperature levels:

- **Waste heat recovery** (washing, ventilation heating): **<42 °C**
- **Space heating and hot water** (cheese storage house): **65 °C**
- **Process heat 1** (cheese vats, cleaning water): **92 °C**
- **Process heat 2** (multi-purpose heater, pasteurisation): **105 °C**

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Technical data of the HTHP

Heat pump type	IWWHS 570 ER6c2
HFO R1234ze(E) (GWP of 6)	I: Industrial heat pump W: Water as heat source W: Water as heat sink H: High temperature heat pump S: Screw compressor 570: Heating capacity range in kW E: Economizer cycle R: Shell and tube heat exchanger 6: Refrigerant HFO R1234ze(E) c2: 2-stage compressor
Heating capacity	approx. 520 kW
Heat source	18/14 °C (in/out)
Heat sink	82/92 °C or 55/65 °C (in/out)
Heat source	Cooling water (waste heat) from the neighboring data center (16 to 20 °C)
Compressor type	2-stage screw with vapor injection
Refrigerant	R1234ze(E) (130 kg, safety group: A2L, mildly flammable)
First operation	2020/21 (using waste heat from the data center)

- The **mildly flammable** (A2L) HFO refrigerant R1234ze(E) demands special measures for **fire protection (e.g. gas sensors, ventilation)** and **escape routes**.

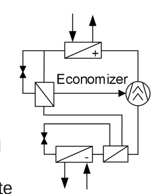
Performance data of the heat pump

	High temperature (W18-14/W82-92)			Low temperature (W18-14/W55-65)		
Part load (%) (by slide valve control)	**100***	75**	50**	**100***	75**	50**
Effective part load (%)	**100**	81	62	**97**	75	54
Condenser capacity (kW)	**520**	419	321	**505**	390	279
Condenser water flow rate (m^3/h)	**44.7**	36.0	27.6	**43.4**	33.5	24.0
Temperature difference condenser (K)	**10.0**	10.0	10.0	**10.0**	10.0	10.0
Evaporator capacity (kW)	**338**	264	195	**385**	293	205
Evaporator water flow rate (m^3/h)	**82.7**	82.7	82.7	**82.7**	82.7	82.7
Temperature difference evaporator (K)	**3.5**	2.7	2.0	**4.0**	3.0	2.1
Compressor power (kW)	**182**	155	126	**120**	98	74
COP_H (-)	**2.85**	2.70	2.55	**4.20**	4.00	3.75

(from data sheet of Ochsner Energie Technik GmbH, * experimentally tested data, ** extrapolated)

Economizer cycle with vapor injection

- **Efficient solution for high temperature lifts**
- **Higher refrigerant mass flow** ↑ at compressor outlet results in **higher heating capacity** ↑
- **Reduced compressor discharge temperature** ↓ is positive with regard to the compressor temperature limits
- **Stronger subcooling** ↑ of condensate increases the COP ↑



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Modelling of an open heat pump cycle for waste heat recovery in an industrial batch process

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Keywords:
High temperature heat pump, mechanical vapor recompression, batch process

Abstract

This study investigates the feasibility of using an open cycle mechanical vapor recompression (MVR) heat pump concept for recovery of heat generated by a multitude of coupled batch reactors. The batch reactors form part of a polymerization process, which requires heating to initiate the reaction, after which the process is cooled to maintain the process within a certain temperature bound. The heat generated from the exothermic reaction is used as a waste heat source for the heat pump process (figure 1). The heat pump consists of a flash vessel, which receives flow from the reactors after an expansion process leading to generation of a small mass fraction of steam. Attached to the vapor exit of this flash vessel is the MVR system which upgrades the pressure of the flash steam from an absolute pressure in the order of 1 $bar_{(abs)}$, up to 12 $bar_{(abs)}$, which can then be fed into an existing steam network. This is achieved through a three stage centrifugal compression process with intercooling between stages.

In the current case, there is the requirement to achieve a constant supply of 12 $bar_{(abs)}$ steam with a design mass flow rate of 20 tonne/hr. This is challenging for such a batch process whereby the availability of waste heat from the coupled batch reactors varies strongly, and in some cases, it is not available at all. To achieve this, an additional heat source in the form of 3 $bar_{(abs)}$ steam which can be provided from existing infrastructure was connected to the inlet of the MVR compressors. In the absence of a waste heat supply, the 3 $bar_{(abs)}$ steam can be used as a heat source until waste heat from the reactors becomes available once more.

In order to evaluate the technical concept, a numerical model of the process was created using the Dymola simulation environment, based on the Modelica language. Thermophysical properties of the working fluid, as well as the majority of components used were from the commercially available TIL media and TIL suite libraries. The return conditions (flow and specific enthalpy) from the batch reactors, for which five days of actual process data are available for, were used as inlet conditions to the model, specifically the flash vessel. The flash vessel was sized at 500 m^3, which is equivalent to approximately 5 minutes liquid hold-up time. Operation maps of the centrifugal compressors were derived from existing compressor designs and re-scaled to fit design specifications of the system, giving confidence the maps are realistic and that the compressors are available in practice. For the modelling conducted in this study, the compressors were coupled to a single shaft and speed controlled to achieve the target design mass flow of 12 $bar_{(abs)}$ steam.

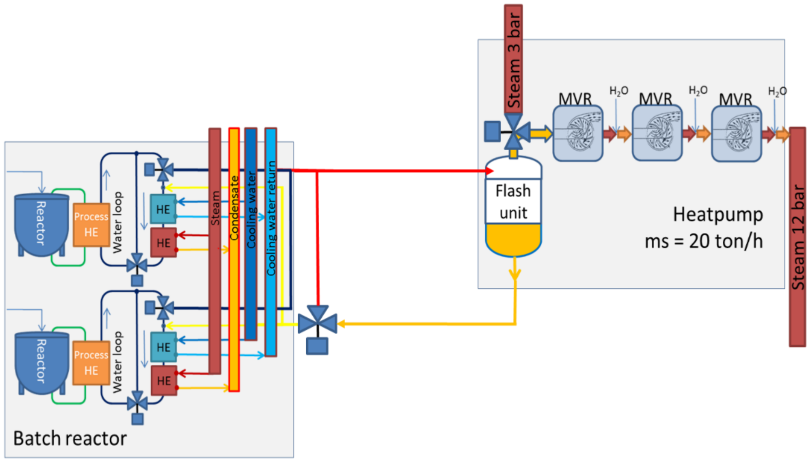

Figure 1: Batch reactor coupled to proposed open heat pump cycle

Preliminary modelling works provided insight into operation of the system. It was determined that when the pressure in the flash vessel drops below 0.8 $bar_{(abs)}$ (T_{sat} = 93.5°C) that the high pressure compressor approaches surge conditions. To avoid surge in the compressor, this pressure (0.8 $bar_{(abs)}$) was set as a lower bound in the flash vessel for switching heat source. To avoid rapid switching of the heat sources for the MVR, a hysteresis band was implemented with an upper pressure which must be exceeded in the flash vessel before the flash vessel can be used as a heat source once more. Too narrow a hysteresis band results in rapid switching between heat source, whilst too large a band is at the detriment of energy efficiency. For the size of flash vessel chosen, a hysteresis band of 0.15 bar was found to prevent significant fluctuations of heat source in the process.

The results of the study, demonstrated that the choice of control strategy resulted in the process operating as expected. When the mass and thus enthalpy flow from the batch reactors approaches zero, in all cases, the 3 $bar_{(abs)}$ steam network is utilized. This is also the case in a few circumstances when the waste heat supply is limited in nature. The results also showed that it was indeed possible to achieve a relatively constant supply of 12 $bar_{(abs)}$ steam output utilizing waste heat from the batch process. The enthalpy flow of the 12 $bar_{(abs)}$ steam was found to vary within 10% of the average value in 94% of the simulation. On average, the system produces 14.4 MW of process heat requiring 3.3 MW of electrical power to the compressor shaft leading to an average COP_{elec} for the system of 4.39. A peak COP_{elec} of 6.52 was recorded, which occurs when using the 3 $bar_{(abs)}$ steam network as a heat source. The use of 3 $bar_{(abs)}$ steam is not free, and as such the $COP_{elec+heat}$ has been calculated, which considers the electricity and steam as input sources. This $COP_{elec+heat}$ goes to 1 at times when the 3 bar steam is used as a heat source for the MVR and had an average value of 3.49 during the simulation. Based on the energy analysis, and with the reasonable assumptions of an electricity price of €50/MWhr, and a steam price of €15/tonne, implementation of this MVR open heat pump concept has a potential to reduce OPEX costs by 563 k€/year.

Future work on this topic will focus on reducing the reliance on the 3 $bar_{(abs)}$ steam network in the absence of waste heat from the reactors through integration of a sensible heat storage, which could be charged during periods of surplus waste heat, and discharged when there is a shortage. Additionally, a more detailed cost analysis of the system and the various layouts will be conducted to gain insights into the economics of differing design options.

MODELLING OF AN OPEN HEAT PUMP CYCLE FOR WASTE HEAT RECOVERY IN AN INDUSTRIAL BATCH PROCESS

› 2ND CONFERENCE ON HIGH TEMPERATURE HEAT PUMPS

› SEPTEMBER 9, 2019 – COPENHAGEN, DENMARK

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Background

This current work investigates the feasibility of using an open cycle mechanical vapor recompression (MVR) heat pump concept for recovery of heat generated by a multitude of coupled batch reactors. The batch reactors form part of a polymerization process, which requires heating to initiate the reaction, after which the process is cooled to maintain the process within a certain temperature bound. The heat generated from the exothermic reaction is used as a waste heat source for the heat pump process (Figure 1). The heat pump consists of a flash vessel, which receives flow from the reactors after an expansion process leading to generation of a small mass fraction of steam. Attached to the vapor exit of this flash vessel is the MVR system which upgrades the pressure of the flash steam from an absolute pressure in the order of 1 $bar_{(abs)}$, up to 12 $bar_{(abs)}$, which can then be fed into an existing steam network. This is achieved through a three stage centrifugal compression process with intercooling between stages.

In the current case, there is the requirement to achieve a constant supply of 12 $bar_{(abs)}$ steam with a design mass flow rate of 20 tonne/hr. To achieve this, an additional heat source in the form of 3 $bar_{(abs)}$ steam which can be provided from existing infrastructure was connected to the inlet of the MVR compressors. In the absence of a waste heat supply, the 3 $bar_{(abs)}$ steam can be used as a heat source until waste heat from the reactors becomes available once more.

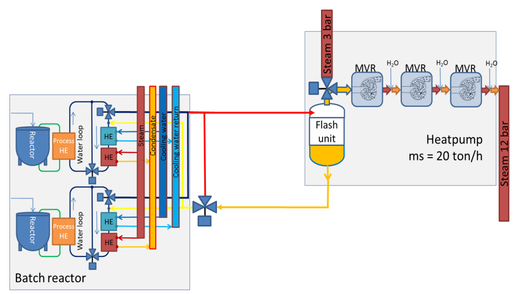

Figure 1: Batch reactor process (box, left), modified to include an open cycle MVR heat pump concept

Approach

- A numerical model of the process was created using the Dymola simulation environment, based on the Modelica modelling language.
- Thermophysical properties of the working fluid, as well as the majority of components used were from the commercially available TIL media and TIL suite libraries.
- Five days of actual process data were made available by the process owner for evaluation of the concept.
- The return conditions (flow and specific enthalpy) from the batch reactors were used as inlet conditions to the model, specifically the flash vessel.
- The flash vessel was sized at 500 m³, equivalent to approximately 5 minutes liquid hold-up time in the vessel.
- Compressor maps of the centrifugal compressors were derived from existing compressor designs and re-scaled to fit design specifications of the system.
- Compressors were coupled to a single shaft and speed controlled to achieve the target design mass flow of 12 $bar_{(abs)}$ steam.
- Intercooling of the flow was conducted between stages to achieve a target superheat of 5 K at the inlet of the next compression stage

Results

Preliminary modelling studies provided insight into operation of the system:

- The pressure of 0.8 $bar_{(abs)}$ (T_{sat} = 93.5°C) was set as a lower limit in the flash vessel before switching heat source. For lower pressures, the high pressure compressor approaches surge conditions.
- To avoid rapid switching of the heat sources for the MVR a hysteresis band was implemented (see Figure 2) with an upper pressure which must be exceeded in the flash vessel before the flash vessel can be used as a heat source once more. A hysteresis band of 0.15 bar was found to prevent significant fluctuations of heat source in the process.

The main outcomes from the study are as follows:

- When the mass and thus enthalpy flow from the batch reactors approaches zero, in all cases, the 3 $bar_{(abs)}$ steam network is utilized (see Figure 3). This is also the case in a few circumstances when the waste heat supply is limited in nature.
- It was possible to achieve a constant steam output supply utilizing waste heat from the batch process. The enthalpy flow of the 12 $bar_{(abs)}$ steam was found to vary within 10% of the average value in 94% of the simulation (see Figure 4).
- The system produces on average 14.4 MW of process heat, whilst requiring 3.3 MW of electrical power leading to an average COP_{elec} for the system of 4.39. The $COP_{elec+heat}$, which considers the electricity and steam as input sources had an average value of 3.49 during the simulation (see Figure 5).
- Assuming an electricity price of €50/MWhr, and a steam price of €15/tonne, implementing this heat pump can lead to a potential to reduction of OPEX costs by 563 k€/year.

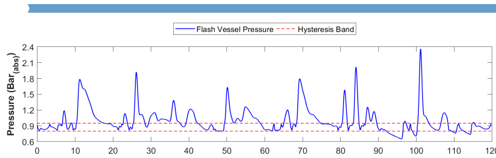

Figure 2: Pressure in the flash vessel, showing the hysteresis band where pressure switching occurs

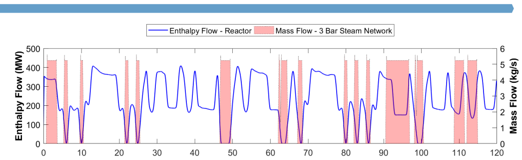

Figure 3: Enthalpy flow from the exit of the reactor (left axis) input to the flash vessel. Heat source is switched to 3 bar steam (right axis) during periods of low waste heat availability from the reactor

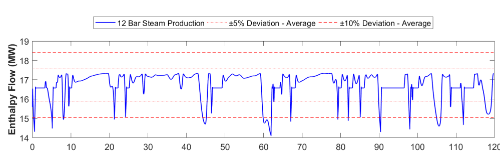

Figure 4: Enthalpy flow at the exit of the MVR demonstrating relatively constant supply of high pressure steam

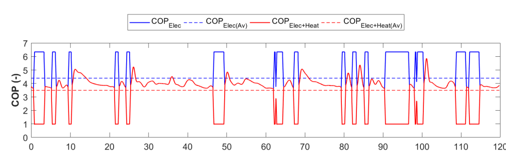

Figure 5: COP of the system during the simulation. High COP_{elec} is achievable when using the 3 bar steam network as a heat source, although this does not necessarily provide any energetic benefits

Dynamic measurements on a steam producing industrial heat pump

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Keywords:
High temperature heat pump, dynamic response, flexibility

Abstract

Industrial heat pumps are a rapidly developing technology that are able to upgrade the temperature of waste heat to be reused in the process with the input of (renewable) electricity. The waste heat from industrial processes is usually at too low a temperature level to be reused and is therefore discarded to the ambient. Its re-use in a process through application of a heat pump can lead to large improvements in process efficiency. In this way, heat pumps are an electrification option that are able to achieve reductions in both primary and final energy consumption.

The value of electrification options such as heat pumps may be maximized if they offer flexibility and take advantage of temporal behaviour of energy markets. This may be considered in combination with traditional process heating equipment and sources (gas). The flexibility characteristics (start and stop times, response to ramping power up or down) of electrification options are therefore critical to understand in detail. Currently, there is limited information available regarding the ability of industrial heat pumps to operate in a flexible manner.

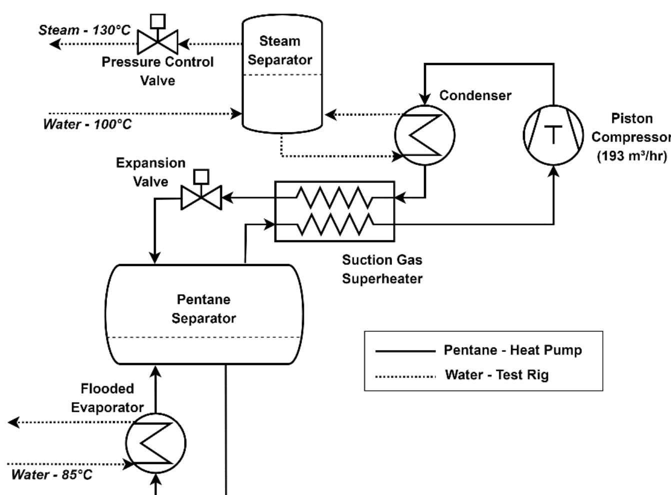

Figure 1: Process flow diagram of steam producing industrial heat pump

This study involves the investigation into the dynamic thermal response of a pilot scale steam producing industrial heat pump. Specifically, the goal was to determine the thermal power ramping response of the system. The heat pump itself is a two-stage flooded system using Pentane (T_{crit} = 197°C) as the working fluid. For the purpose of these experiments, only a single stage was utilized (process flow diagram seen in figure 1). The heat pump uses an open type piston compressor manufactured by Mayekawa which has a rated capacity of 193 m^3/hr at 1450 RPM.

For conducting the experiments, the heat pump is connected to a specially designed heat pump test rig, which is able to simulate conditions (temperatures, flows) of industrial processes at both the source (evaporator) and sink (condenser) of the heat pump. In the current work, the set point for the evaporator inlet temperature was 85°C, whilst the condenser steam pressure was 2.7 $bar_{(abs)}$, equivalent to a saturation temperature of 130°C. Two experiments were conducted in series to determine the response time of the heat pump to both positive and negative ramping of thermal power. Ramping of the thermal power was achieved by actuation of the compressor speed, with switching between a minimum value of 900 RPM to a maximum value of 1600 RPM and vice versa.

The results of the experiments were characterized in their entirety by fluctuations in the inlet temperature to the evaporator, caused by a poorly configured control loop. The temperature fluctuated between 83°C and 88°C, leading to fluctuations in the measured thermal and electrical power and therefore making it difficult to determine the true response rate of the system. With the compressor operating at 1600 RPM, the system produced an average of 83.4 kW of process heat. When operating the compressor at 900 RPM, the thermal power was reduced to an average value of 55.0 kW, a turn down factor of 34%. To determine the response time of the heat pump, the time was measured from when the compressor speed is initially actuated, until the gradient of measured thermal power in the condenser reached a value of zero. When ramping up in power, the response rate was measured to be 6 minutes, 20 seconds, whilst when ramping down in power the response was 5 minutes 40 seconds.

The results of the experiments give a first indication that industrial heat pumps may be suitable for operating in the balancing electricity market. Further work on this topic will initially focus on achieving a more stable evaporator temperature, which should lead to improved ability to interpret results. Following this, an attempt will be made to further characterize the heat pump dynamics, including gaining insight to the response times for start-up and shut down, actuating power through means other than compressor speed, dynamic response at different temperature levels and the main factors which affect response of the heat pump system.

DYNAMIC MEASUREMENTS ON A STEAM PRODUCING INDUSTRIAL HEAT PUMP

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The value of electrification options such as heat pumps may be maximized if they offer flexibility and take advantage of temporal behaviour of energy markets. This may be considered in combination with traditional process heating equipment and sources (gas). The flexibility characteristics (start and stop times, response to ramping power up or down) of electrification options are therefore critical to understand in detail. Currently, there is limited information available regarding the ability of industrial heat pumps to operate in a flexible manner.

This study involves the investigation into the dynamic thermal response of a pilot scale steam producing industrial heat pump. Specifically, the goal was to determine the thermal power ramping response of the system. The heat pump itself is a two-stage flooded system using Pentane (T_{crit} = 197°C) as the working fluid. For the purpose of these experiments, only a single stage was utilized (process flow diagram seen in figure 1). The heat pump uses an open type piston compressor manufactured by Mayekawa which has a rated capacity of 193 m³/hr at 1450 RPM.

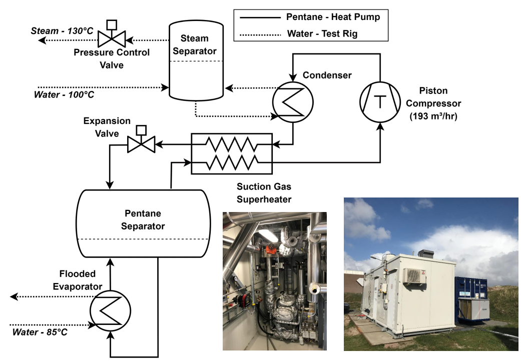

Figure 1: Process flow diagram of steam producing industrial heat pump (main) and images of the heat pump and compressor (inserts)

Approach

- The heat pump is connected to a specially designed heat pump test rig, which is able to simulate conditions (temperatures, flows) of industrial processes at both the source (evaporator) and sink of the heat pump (condenser).
- Both the heat pump and test rig contain sufficient instrumentation to measure flows and temperatures and allow calculation of thermal powers for the various system components.
- The set point for the evaporator temperature was 85°C, whilst the condenser steam pressure was 2.7 bar(abs), equivalent to a saturation temperature of 130°C.
- Two experiments were conducted in series to determine the response time of the heat pump to both positive and negative ramping of thermal power.
- Ramping of the thermal power was achieved by actuation of the compressor speed, with switching between a minimum value of 900 RPM to a maximum value of 1600 RPM and vice versa.

Results

- The results of the experiments were characterized in their entirety by fluctuations in the inlet temperature to the evaporator, caused by a poorly configured control loop.
- The temperature in the flow to the evaporator fluctuated between 83°C and 88°C, leading to fluctuations in the measured thermal power
- Temperature fluctuation make it difficult to determine the true response rate of the system
- With the compressor operating at 1600 RPM, the system produced an average of 83.4 kW of process heat. When operating the compressor at 900 RPM, the thermal power was reduced to an average value of 55.0 kW, a turn down factor of 34%.
- The response time of the heat pump was measured as the time from when the compressor speed is initially actuated, until the gradient of measured thermal power in the condenser reached a value of zero.
- When ramping up in power, the response time was measured to be 6 minutes, 20 seconds
- When ramping down in power the response time was measured to be 5 minutes 40 seconds.

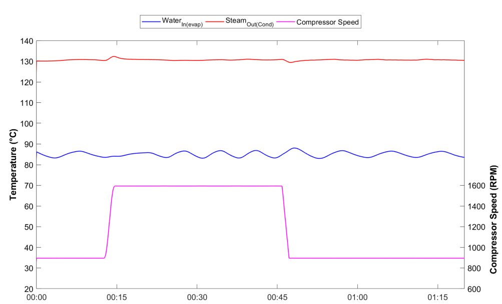

Figure 2: Temperature of the process flow streams during the experiment. Fluctuation in the evaporator temperature caused by poorly configured control loop characterizes the experiments

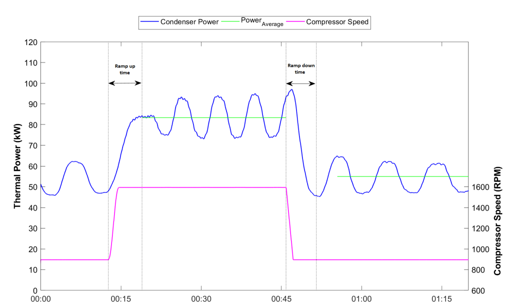

Figure 3: Calculated condenser power for ramping up and down compressor speeds demonstrating the response time. Average power for high and low compressor speed is overlaid

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