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Authors		
Author(s) Name	Organisation	E-mail address
Ole Marius Moen	SINTEF Energi Research	Ole.moen@sintef.no
Joshua Dowdell	SINTEF Energy Research	joshua.dowdell@sintef.no
Magnus Rotan	SINTEF Energy Research	Magnus.rotan@sintef.no
Christian Schlemminger	SINTEF Energy Research	N/A

Abstract

This report presents the results based on the experimental work on the two-stage turbo compressor steam test rig. The results were presented at the 2024 High Temperature Heat Pump Symposium in Copenhagen, Denmark. This report contains the presentation and extended abstract for this conference presentation.

Cooperating project for this work is Horizon 2020 EU project, FRIENDSHIP.

Evaluation of turbo compressor performance for a water based HTHP to be utilized in solar assisted heat supply

Ole Marius Moen[1], *Magnus Rotan*[1], *Joshua Dowdell*[1], *Christian Schlemminger*[1]

[1] SINTEF Energi AS, Department of Thermal Energy, Trondheim, Norway,
Ole.Moen@sintef.no, Magnus.Rotan@sintef.no, Joshua.Dowdell@sintef.no

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High temperature heat pump, natural working fluids, turbo compressors, water, high pressure, solar thermal heat

Introduction

Heating systems in industrial processes, transport and buildings are extraordinarily carbon-intensive and highly polluting. Fossil fuels by far account for the largest share of the heat generation, covering 77% of the global heat demand in 2018. Industrial heat demand covers a significant fraction (about 25%, 105 EJ) of the total energy demand in the world, while being responsible for 23 % of the global energy related CO_2 emissions. A significant portion of the heat demand is required at temperatures between 100 – 200 °C [1]. This context arises the need to accelerate the use of renewable energy sources for heat supply for the industry. One such solution is to utilize solar assisted heating in combination with a High Temperature Heat Pump (HTHPs) system, supplying heat up to 200 °C.

The work presented here has been performed as part of the FRIENDSHIP project. FRIENDSHIP is an EU funded project within the Horizon 2020 research and innovation programme [2]. The project seeks to increase contribution from solar thermal energy for industrial heating and cooling processes. Current solar collectors are mainly used for low-temperature heat supply, such as building heating or domestic hot water production. Advanced solar collectors, which can produce higher temperature exist, but are not cost-effective today. The main solution in FRIENDSHIP is to combine solar thermal heat, based on Parabolic Through Collectors (PTCs) with a HTHP based on water (R-718) as working fluid to upgrade the temperature up to 180-200 °C, which can be used to supply both heating and cooling, the latter by use of absorption chillers. Thermal storage is used to balance the mismatch between industrial heating and cooling demand, and solar output.

The focus of the work presented here is the evaluation of water vapour compression performance using turbo compressors (two-stage) at suction pressures of 1 bara to 2 bara.

Methods

The concept features the use of PTCs which acts as the heat source for the HTHP and delivers sensible heat in the temperature range of 140-160 °C. The goal is to reach a heat supply up to 180-200 °C, which corresponds to condensation pressure of 10-15 bara. The overall concept of the HTHP is the use of water as working fluid and compression technology based on turbo compressors. There are several benefits of using water. As it belongs to the group of natural refrigerants, there is no climate footprint (GWP/ODP). Water is non-flammable and non-toxic (safety group A1). Alternative 4th generation synthetic refrigerants are under scrutiny due to recent findings showing that they break down to trifluoroacetic acid (TFA), which are potentially very harmful to humans and the environment [3]. Water has a very high critical temperature of 374 °C, and among the usual natural refrigerants selected for HTHP applications it is the only one that allows for condensation temperatures up to 200 °C.

Since the maximum discharge temperature of the superheated steam from the compressors will be in the range of 250 °C - 300 °C, compressor technology relying on lubrication oil has been avoided since the high temperatures quickly would degrade its lubricating properties. Instead, turbo-compressors were chosen, due to their oil-free operation. Turbo-compressors also have the advantage of high volumetric flow rates for their size, facilitating a potentially compact HTHP system.

The initial concept featured a closed loop HTHP with 3 compression stages and was modelled numerically in Modelica. For a temperature lift of 50 Kelvin (140 °C – 190 °C) between the heat source and sink, a pressure ratio (PR) of 1.8 was required for each compressor. Using a set maximum compressor isentropic efficiency of 0.75, resulted in a theoretical COP of 4.96.

Rotrex was selected as supplier of the compressors. The compressor technology is based on mass-produced automotive superchargers. Although the impeller of the compressor operates oil-free, the compressor is driven by a VFD (inverter) controlled DC electric motor coupled to an oil-filled gearbox to achieve its high rotational speed. Previous research and experience with this turbo-compressor technology in SINTEF, primarily through the DryFiciency project [4] uncovered issues related to the shaft sealing between the impeller and the gearbox leading to leakage of water into the gearbox oil. To handle this issue a novel sealing solution was developed. This relies on an air purge chamber situated between the impeller and the gearbox. The purpose is to use pressurised air at a pressure higher than the discharge pressure of the compressor to create an additional sealing barrier.

For the laboratory experiments of the turbo-compressors a solution based on a two-stage compression was selected, partly because of complexity, but also since the compressors can achieve PRs higher than the simulated 1.8 per stage. The existing testing facilities have been modified to allow for operation at elevated pressures. The experimental rig is a closed loop with two compression stages. Pressurized steam is generated in a steam generator tank. After each compression stage the working fluid is de-superheated by means of liquid injection. The maximum cycle pressure is regulated with a flow control valve. The steam is then recirculated back into the generator tank.

Results and discussion

The experimental work is currently ongoing, and the results presented here are based on the preliminary evaluation of the results so far. The compressors have been tested and characterized up to 85 % of their nominal speed, which corresponds to an impeller speed of 68.000 rpm. Due to motor limitations the maximum achievable speed is 90%, or around 72.000 rpm. At these speeds a PR of 2.03 (stage 1) and 1.97 (stage 2) has been achieved, resulting in a total pressure ratio of 3.6 when including for pressure losses between the compressors. The compressor mapping shows that the performance characteristics for stage 1 and stage 2 are similar, see Figure 1. The resulting isentropic efficiency is slightly lower than assumed in the initial concept and ranges from 0.55 at near choked conditions to 0.7 near the design conditions.

The calculated COP and Carnot efficiency as a function of the total temperature lift is presented in Figure 2. For a given compressor speed the Carnot efficiency approach 0.45 when the pressure ratio, and therefore the temperature lift is maximised. At reduced temperature lift the Carnot efficiency is significantly reduced. The effect is seen as the opening of the flow control valve is increased. This reduces the backpressure, and the compressors move away from design conditions and towards choked flow. This leads to reduced isentropic efficiency and greater mass flows, which increases the system pressure losses.

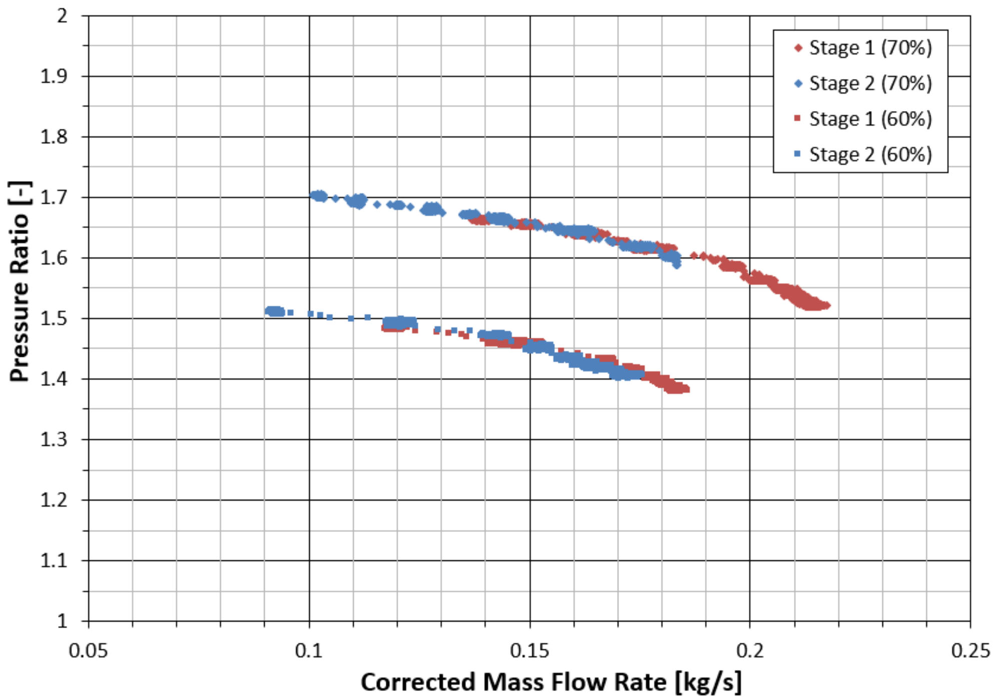

Figure 1: Compressor map for stage 1 and stage 2 compressor at speeds of 60% (48.000 rpm) and 70% (56.000 rpm) relating the pressure ratio (total pressure) to the corrected mass flow rate. Reference conditons are 1 atm and 120 degC

The preliminary test results at elevated suction pressures of 2 bar at stage 1 inlet, shows similar performance characteristics as achieved during testing with low suction pressures. At 70% operational speed the maximum achieved stage 2 discharge pressure is 5.25 bara, which corresponds to a steam condensation temperature of 154 °C. Since the system is operating with greater mass flows at elevated pressures the overall cycle pressure ratio is slightly reduced at elevated suction pressures. It is therefore expected that the system efficiency will be reduced as the stage 1 suction pressures are intended to be increased up to 3 bara in future testing. Due to the greater mass flows at elevated suction pressures, the electric motors are operating closer to their maximum capacity and may therefore become a bottleneck, limiting the maximum potential compressor speed and correspondingly, the maximum achievable discharge pressure of the HTHP. At 2 bara suction pressure for stage 1, the thermal output achieved at 70% speed is around 700 kW with mass flows exceeding 1000 kg/h. Selected key performance parameters from the testing results are presented in Table 1.

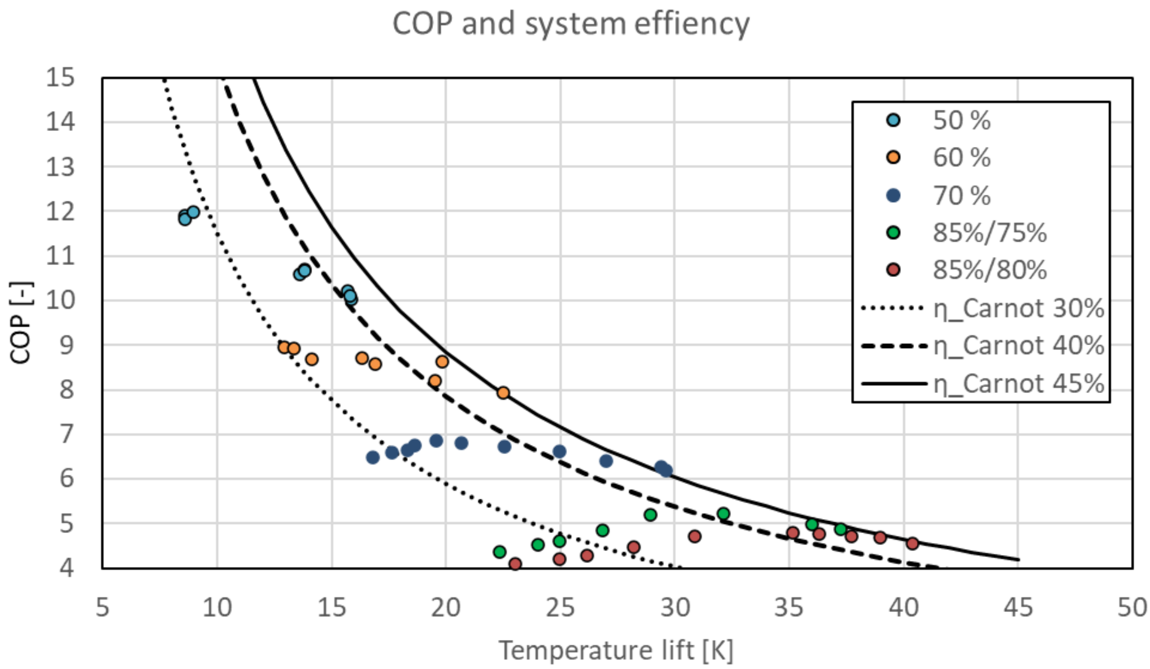

Figure 2: COP and Carnot efficiency for various compressor speeds (%) as function of temperature lift

Introduction of the novel purge chambers to avoid water leakage into the gearbox oil have given satisfactory results. Although, there has been some issues with achieving a full seal of the high-pressure air at the ports of the chamber. Two different settings of the purge chamber have been tested, whereas the so-called dead-end configuration, where the air is trapped within the purge chamber has worked best. Oil samples have been extracted after every test. Because the oil has a very low water saturation ratio, any water contamination will be clearly visible when it occurs. When correctly operated i.e. with purge air supply pressure > compressor discharge pressure, no water contamination is observed at all, which is a success.

Table 1: Selected key performance data from compressor mapping

Compressor speed		Mass flow, $\dot{m}$	Pressure ratio, Π_t	P_{inlet} stage 1	$P_{disc.}$ Stage 2	T_{cond}	T_{lift}	Heat delivery	COP	η_{Carnot}	η_{isen} Compressors	
Stage 1 [%]	Stage 2 [%]	[kg/h]	[-]	[bara]	[bara]	[°C]	[K]	[kW]	[-]	[-]	Stage 1 [-]	Stage 2 [-]
50	50	356	1.71	1.03	1.76	116	16	234	10.12	0.410	0.66	0.63
60	60	490	2.10	1.08	2.27	124	22	317	7.94	0.449	0.63	0.65
70	70	551	2.63	1.04	2.73	130	30	364	6.19	0.455	0.64	0.55
80	80	752	3.42	1.06	3.64	140	39	495	4.83	0.455	0.634	0.64
85	80	763	3.58	1.05	3.76	141	40	502	4.56	0.44	0.61	0.60
60	60	803	2.00	1.98	3.97	143	23	514	8.24	0.46	0.68	0.63
70	70	1073	2.55	2.06	5.25	154	33	688	6.41	0.49	0.67	0.62

Conclusion

The work presented here is the evaluation of the performance of a water vapour compression system using turbo-compressors in a two-stage layout at suction pressures of 1 bara to 2 bara. The overall goal is to achieve heat supply at temperatures up to 180-200°C, thus demonstrating a part of the FRIENDSHIP project's concept of utilizing solar assisted heating in combination with a High Temperature Heat Pump (HTHPs) for industrial heating and cooling. The compressors have been mapped at atmospheric suction pressures, resulting in compressor isentropic efficiencies up to 0.7. The preliminary test results show similar performance characteristics at elevated pressures, which is promising. A maximum cycle pressure at stage 2 discharge of 5.3 bara has been achieved, corresponding to a condensation temperature of 154°C. The introduction of the novel purge chamber has been satisfactory as it is able to create an efficient seal between the steam and the gearbox oil, thereby avoiding any contamination. Future testing up to 3 bara suction pressures is planned. However, limitations such as electric motor capacities could limit the maximum attainable discharge pressure before the actual potential of the compressors is reached.

References

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2. FRIENDSHIP, https://www.friendship-project.eu/
3. Atmosphere. 2022. The Rising Threat of HFOs and TFA to Health and the Environment, https://atmosphere.cool/hfo-tfa-report/
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Evaluation of turbo-compressor performance for a water based HTHP

to be utilised in solar assisted heat supply

Ole Marius Moen, Magnus Rotan, Joshua Dowdell, Christian Schlemminger

Researcher, SINTEF Energy Research
Ole.moen@sintef.no

High-Temperature Heat Pump Symposium
23-24 January 2024 – Copenhagen, Denmark

Background

- Solar thermal energy is a growing energy source
- Global capacity: 542 GW_{th}[1]
- Mostly low-temperature heating like building heating and DHW
- Despite great potential – little adoption to industrial heating and cooling
  - Industrial heat demand: 25% of final energy demand[2]
  - Solar heat for industrial processes inst. cap.: 856 MW_{th}[1]
- Advanced high-temperature solar thermal not cost-effective today

Can we couple solar thermal and HTHP for industrial heating and cooling?

[1](IEA-SHC, 2023)
[2](IEA Renewables, 2019)

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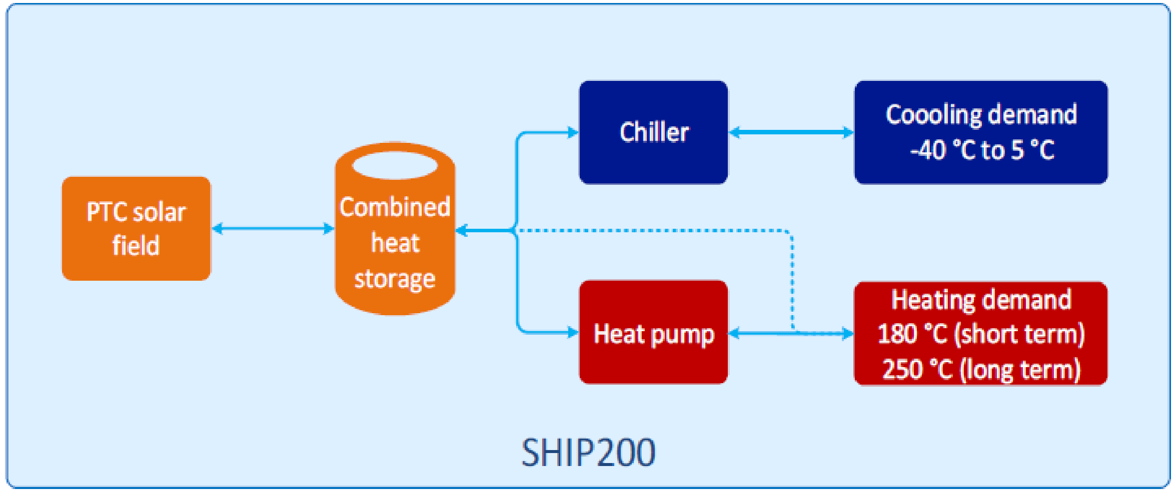
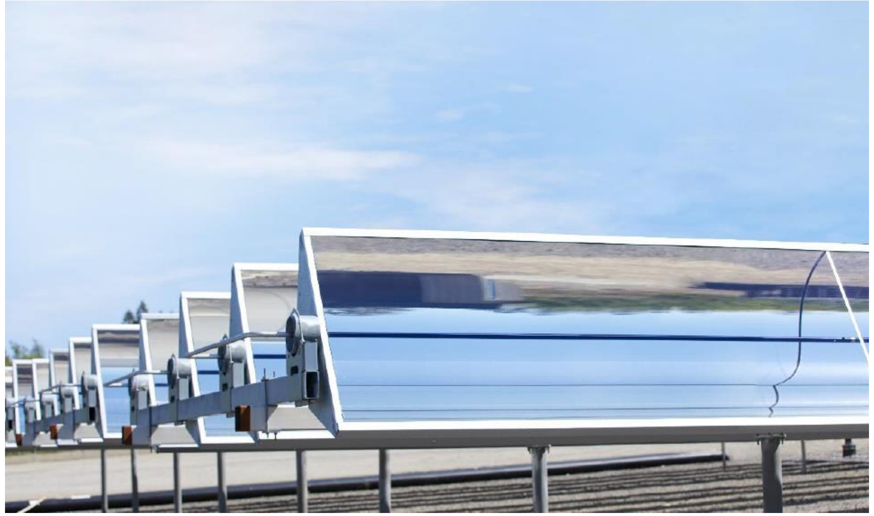
FRIENDSHIP Project

Concept

- Combine solar thermal energy and HTHP to produce industrial heating and cooling
- Parabolic Through Collectors (PTC) - 160 °C
- Further upgrade to 180-200 °C through HTHP
- Thermal storage to buffer solar variations



Demo site Grenoble (Fr), Annual DIN 1,400 kWh/m^2



Absolicon T160 parabolic through solar collector (PTC), absolicon.com

HTHP: Why steam and turbo?

Water/steam (R-718)

- Natural: no GWP/ODP
- High critical temperature 374 °C → condensation temperature of 200 °C
- Non-toxic, non-flammable
- Potentially direct steam supply to process or MVR in e.g. drying processes

High speed turbo compressors:

- Compact – high volumetric capacity
- No lubrication required
- Cost-effective

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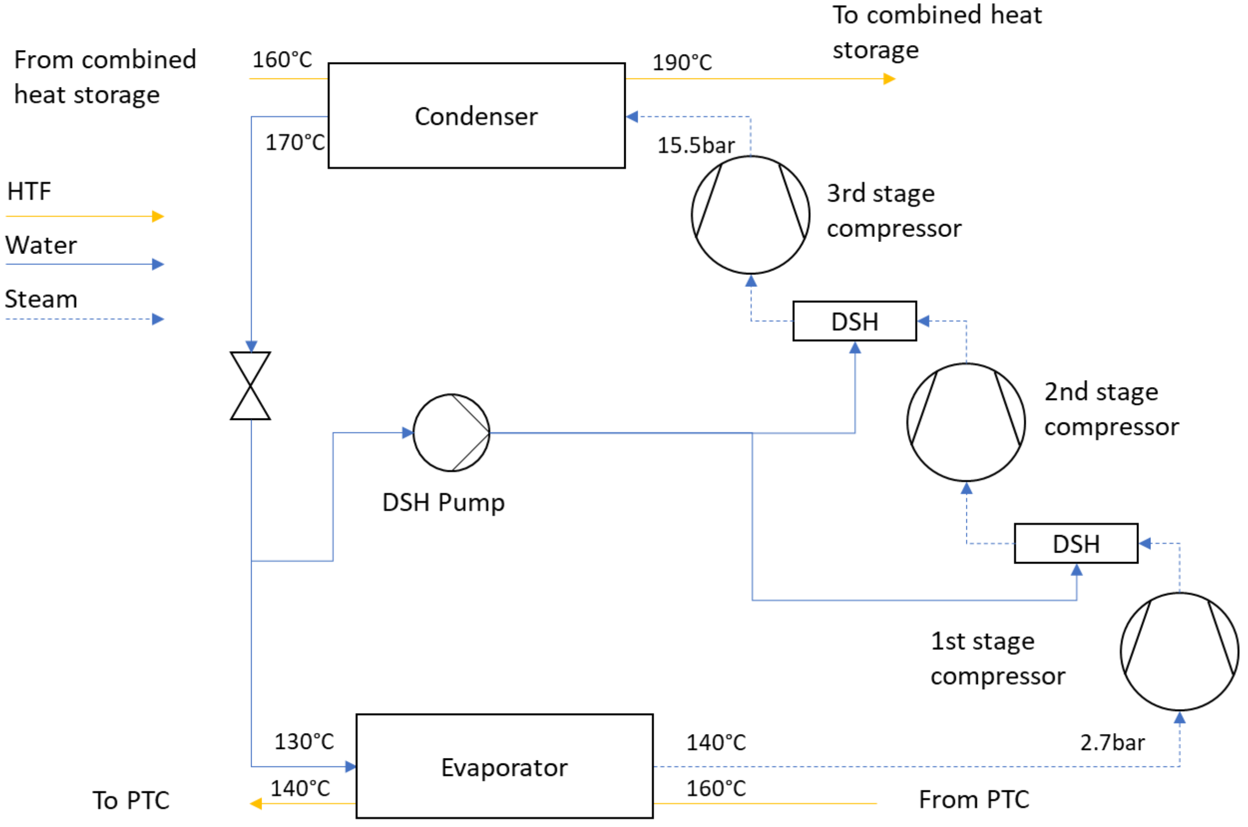
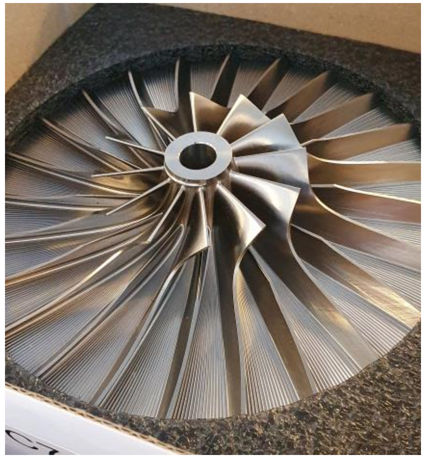
HTHP Concept

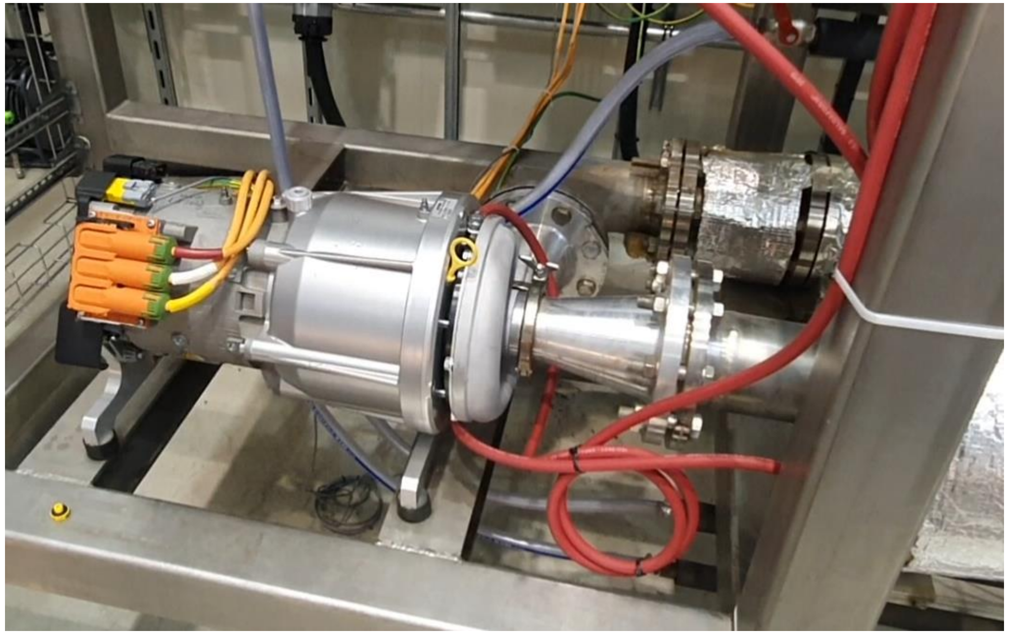
- Initially 3 compressor stages, intermittent DSH
- Temperature lift 70 Kelvin
  - Evap: 130 °C , 2.7 bara
  - Cond: 200 °C 15.5 bara
- Limited pressure and temperature lift with turbo compressors
- Numerically simulated in Modelica
  - Dedicated turbo compressor model, design efficiency, η_{isen} (0.75)
  - COP: 4.96
  - Per stage pressure lift: 1.8



Turbo compressors

- Two compact turbo-compressors coupled in series
- Supplied by Rotrex (DK), derived from automotive performance supercharger technology
- Length x width x height: 0.7 x 0.36 x 0.52 (m)
- < 80 kg
- Capable of pressure ratio > 2.3
- Inverter controlled DC motor (60 kW nom. Cap)
- Lubricated and cooled gearbox, transmission ratio 6.0
- Impeller max. rotational speed: 80 000 rpm





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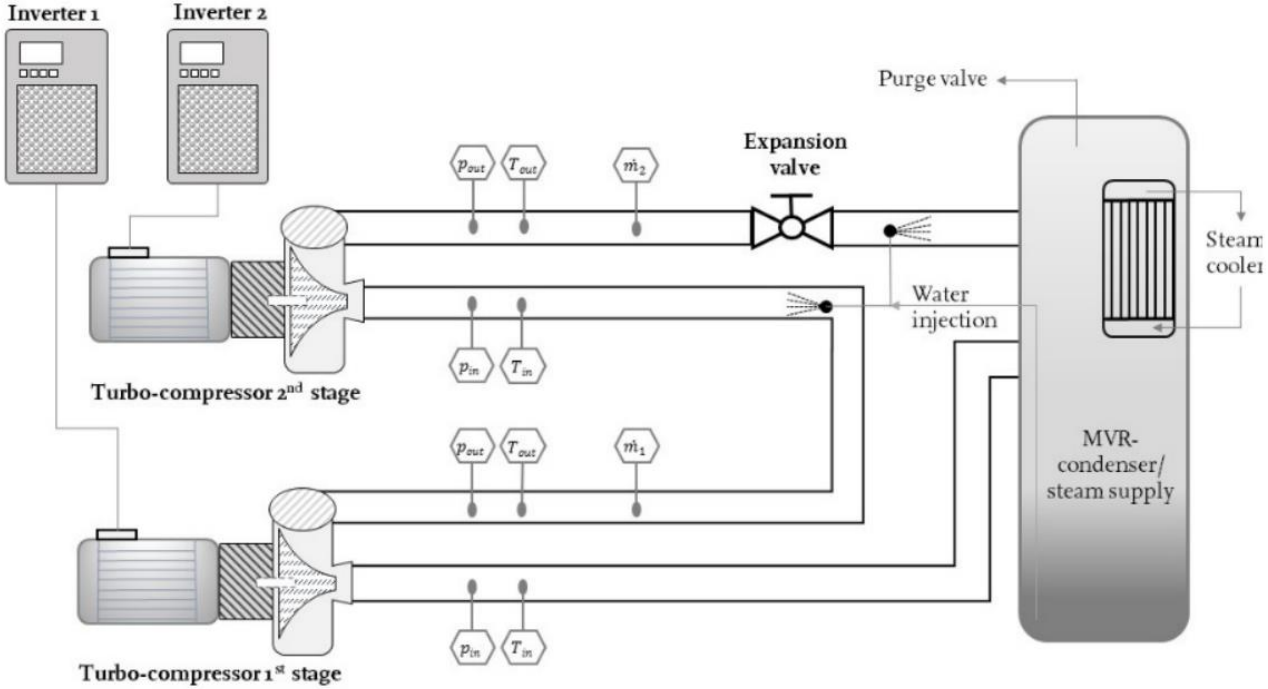
Test rig

- Two-stage compressor test rig at SINTEF lab built by Epcon (DryFiciency)
- Steam generator/condenser tank
- DSH via water injection
- Modified for testing at elevated suction pressure (3 bara)



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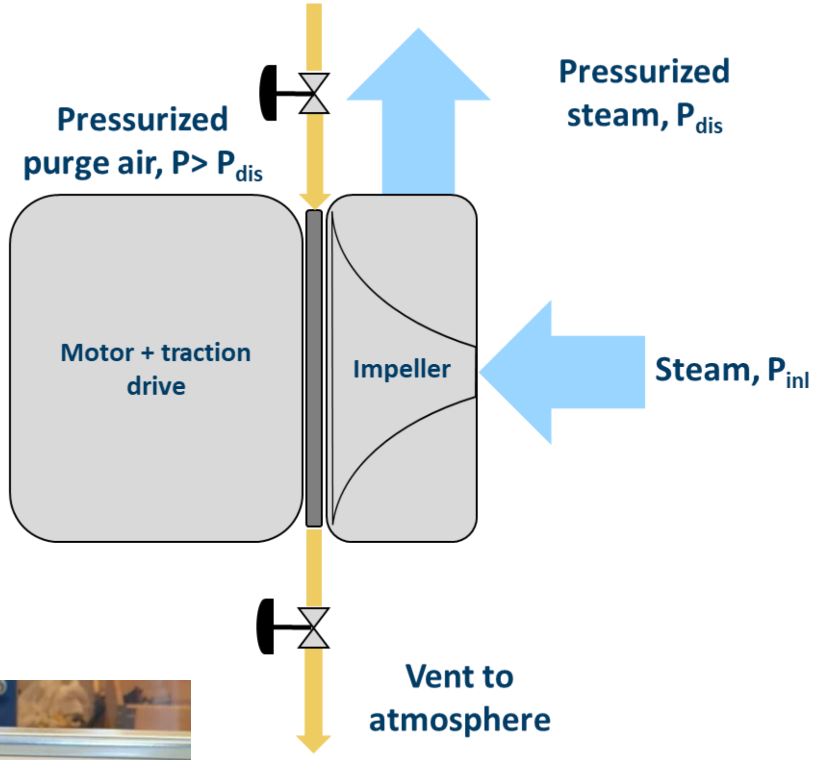
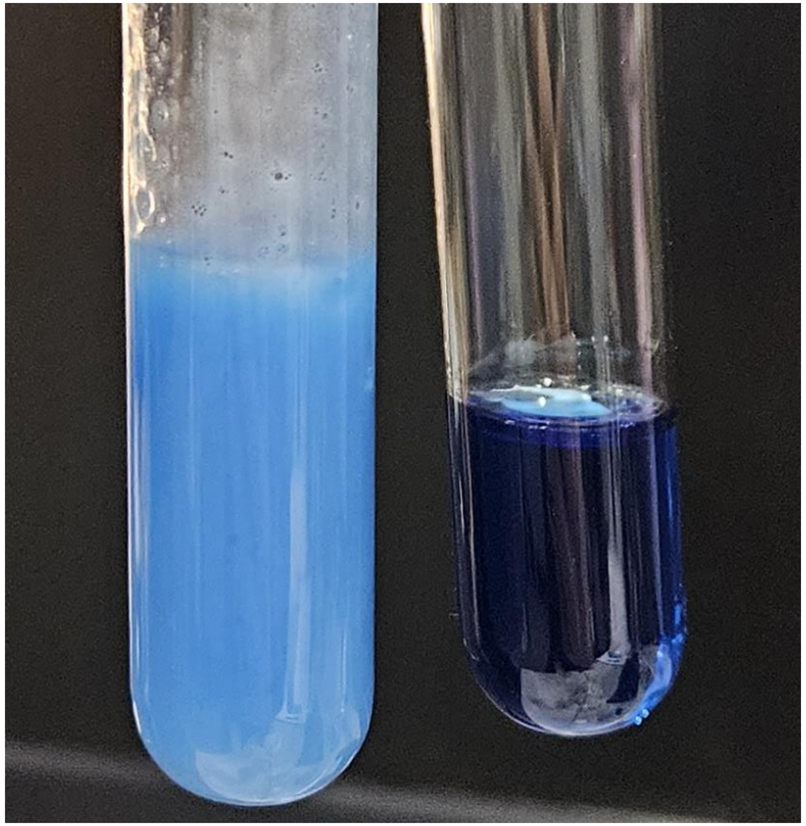
Novel sealing solution – Purge chamber



- Previous testing experiences (DryF) uncovered steam leakage into gearbox → degradation of oil lubrication
- Addition of purge chamber to create additional barrier between steam and oil
- Located between the impeller and gearbox
- Disc with an air channel from end to end
- Pressurise air into the channel and along the surface of the impeller shaft
- To configurations have been tested:
  - Open (atmospheric vent open)
  - Dead-end (atmospheric vent closed)

Novel sealing solution - results

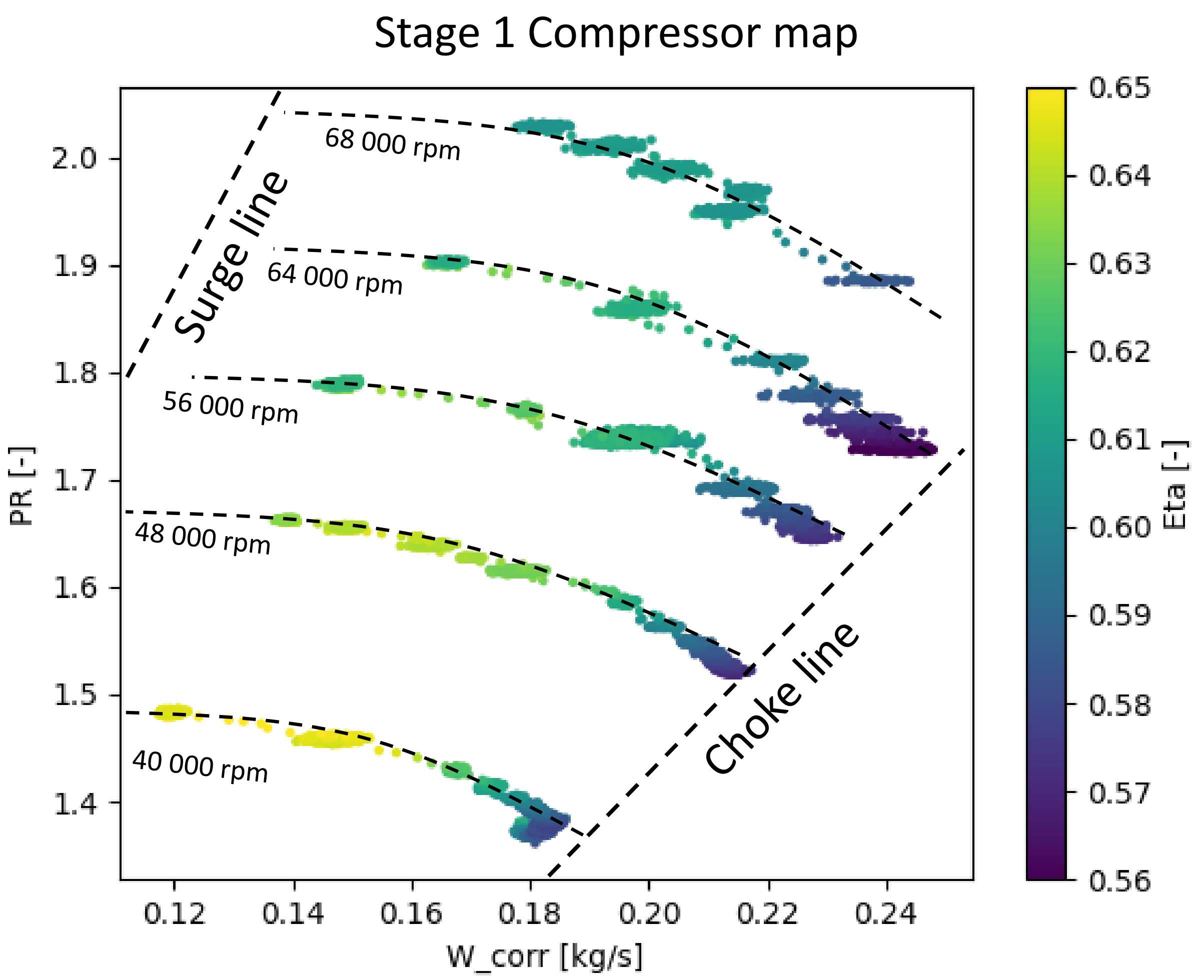
- Oil is sampled after every test run
- No visible water contamination when correct air pressure is applied $P_{air} > P_{comp.\ disch.}$
- Oil has a very low water saturation rate, any contamination should be clearly visible
- Best results with closed dead-end config



Left: Heavily contaminated sample (about 1:1 oil-water vol. ratio)
Right: Test sample – no contamination

Compressor mapping

- Relates the pressure lift and flow rate
- Predicts performance at various operating conditions
- Conditions:
  - Stage 1 inlet pressure: 1 bara
  - Up to 85 % of max speed
  - Stage 2 max discharge pressure: 3.8 bara / 141 °C (cond.)
- Pressure ratio:
  - Stage 1: 2.03 (68 000 rpm – 85 %)
  - Stage 2: 1.97 (64 000 rpm – 80 %)
- Isentropic efficiency (Eta)
  - 0.55 – 0.7
  - Slightly higher efficiencies at lower speeds



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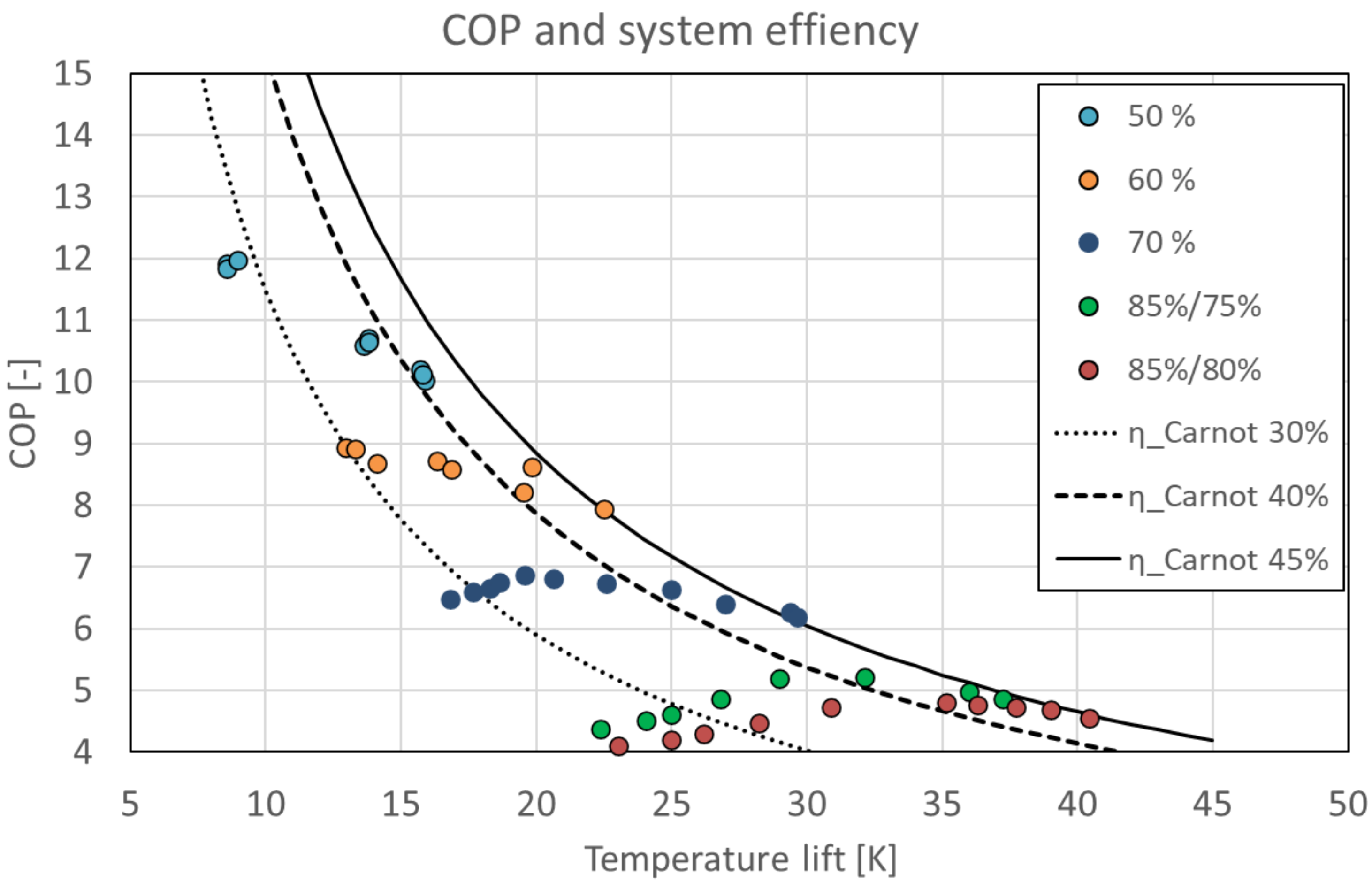
COP and system performance

- COP values highly dependent on overall temperature lift
- Carnot efficiency is used to evaluate system efficiency
- Max Carnot efficiency 46 %
- Max system T lift: 40 K → expect up to 50 K
- At constant speed, lower temperature lift reduces the Carnot efficiency:
  - Reduced compressor isentropic efficiency
  - Higher mass flows gives higher pressure loss between stage 1 and stage 2 compressor



$$COP = \frac{\dot{Q}_{Heating}}{P_{EL}} \qquad COP_{Carnot} = \frac{T_H}{T_H - T_L} \qquad \eta_{Carnot} = \frac{COP_{real}}{COP_{Carnot}}$$

Results from initial testing at elevated pressures

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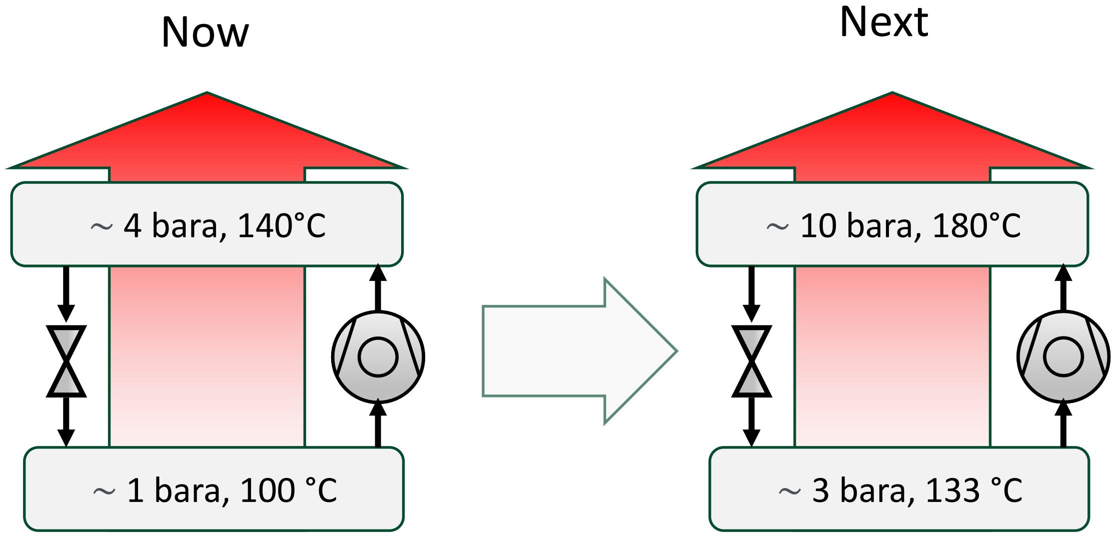
Heat delivery at 180-200 °C require higher than 1 bara/100 °C cycle evap. temperature/pressure

- Ongoing work of testing at elevated stage 1 compressor suction pressures:
  - 2 bara, 120 °C (evap.)
- Similar performance as atmospheric suction pressures:
  - $\eta_{\text{isen Stage 1/2}}$ 0.67 / 0.62
- Max cycle pressure and temperature, 70 % (56 000 rpm):
  - 5.25 bara, 154 °C (cond.)



Thermal capacity at various operating conditions

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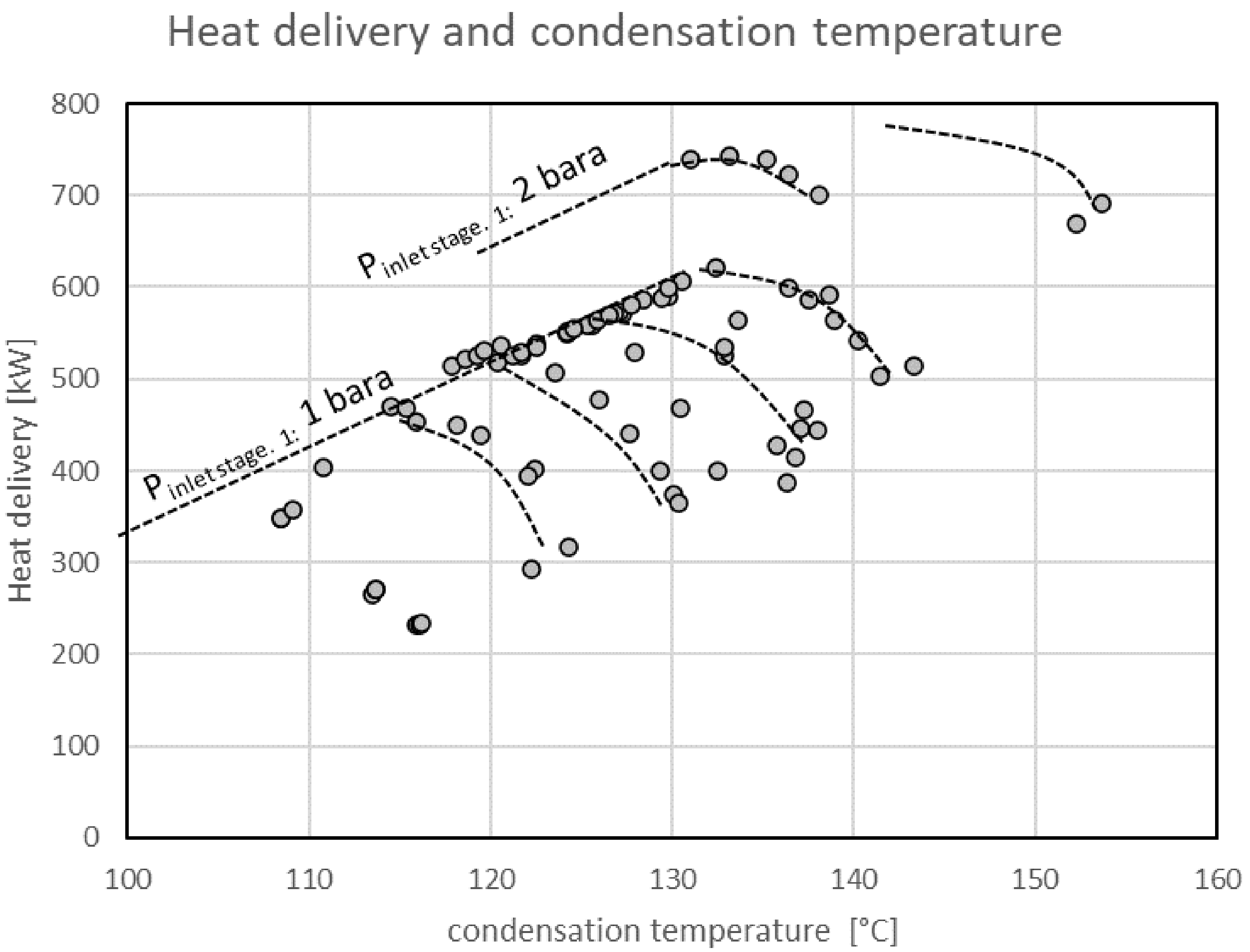
- Maximum thermal capacity > 700 kW, steam mass flow > 1100 kg/h
- Higher capacities at elevated suction pressures due to increased mass flow rate
- Limited load flexibility



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Future work and outlook

- FRIENDSHIP ending in 2024
- Goal of reaching condensing temperatures of 180-200 °C
  - At elevated suction pressure up to 3 bara
- High mass flow and P/T putting equipment limits to the test
  - Maximum capacity of electric motors
  - Very high compressor discharge temperatures, up to 300 °C
  - Other sealing issues due to high P/T

Selected key performance data

Compressor speed		Mass flow, $\dot{m}$	Pressure ratio, Π_t	P_{inlet} stage 1	$P_{disc.}$ Stage 2	T_{cond}	T_{lift}	Heat delivery	COP	η_{Carnot}	η_{isen} Compressors	
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