



14<sup>th</sup> IEA Heat Pump Conference  
15-18 May 2023, Chicago, Illinois

# Performance of a new ultra-high temperature industrial heat pump

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## Abstract

This paper presents a new ultra-high temperature heat pump, working on the Stirling cycle, with helium as working medium. The heat pump can generate heat up to 200°C from sources as low as -10°C with high energy efficiency. A 400 kW prototype installation at a biogas facility is described, where the heat pump supplies steam and cooling to the CO<sub>2</sub> capture process. The performance of the heat pump is presented across a wide range of source and sink temperatures, in terms of heating capacity and share of Carnot COP. The experimental setup is described in detail, as well as the simulation model used for comparing simulated and experimental data. The performance figures are compared with published data for other heat pump cycles used for high temperature heat pumping. The results indicate that helium heat pumps may be more efficient than vapor compression heat pumps for temperature ratios above 1.3 K/K (sink temperature/source temperature).

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*Keywords: ultra high-temperature lift heat pump; UHTHP; very high temperature heat pump; VHTHP; high temperature lift; steam generating heat pump; helium; CO<sub>2</sub> capture*

## 1. Introduction

A prototype of a new stirling-cycle heat pump is installed in a test rig at the biogas facility IVAR near Stavanger, Norway. The pilot installation is shown in Fig.1.

This paper presents the performance of a novel high temperature difference heat pump. The performance is measured at a non-commercial installation of the heat pump, operating with a heat source varying between

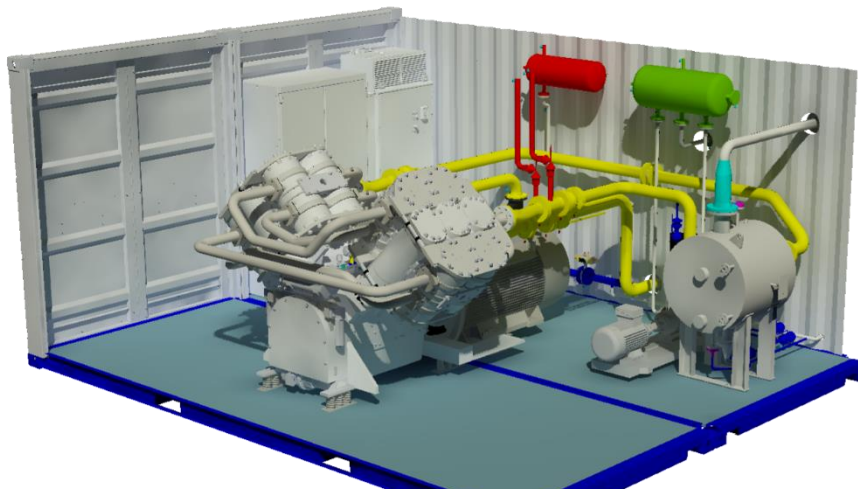


Fig. 1. Pilot installation with steam generator

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5°C and 45°C and with heat sink temperatures between 140°C and 190°C. The heat pump performance and COP are simulated across a wide range of temperatures and measured at the temperatures that were possible at the plant.

The first sections give a description of the heat pump, the thermodynamic process, the process layout and the instrumentation.

The next sections present the results from testing the heat pump at different loads and the comparison between the measured parameters and simulated values. The discussion and conclusions are given in the last section.

## 2. HoegTemp ultra high-temperature heat pump

The ultra-high temperature heat pump HoegTemp (Fig.2) from Enerin, is a Stirling-cycle heat pump with helium (refrigerant R-704) as working medium. As the Stirling cycle relies on compression and expansion of a gas, Stirling-cycle heat pumps are not characterized by boiling points and condensing temperatures of their refrigerant, but rather by the ratio of absolute temperatures for the source and sink.

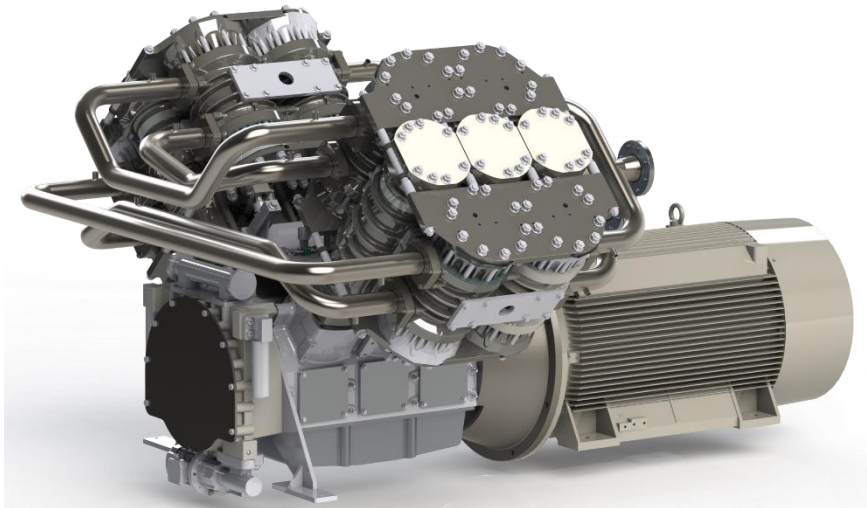


Fig. 2. HoegTemp heat pump

The HoegTemp is a 4-circuit piston compressor heat pump of the double-acting gamma configuration. Each

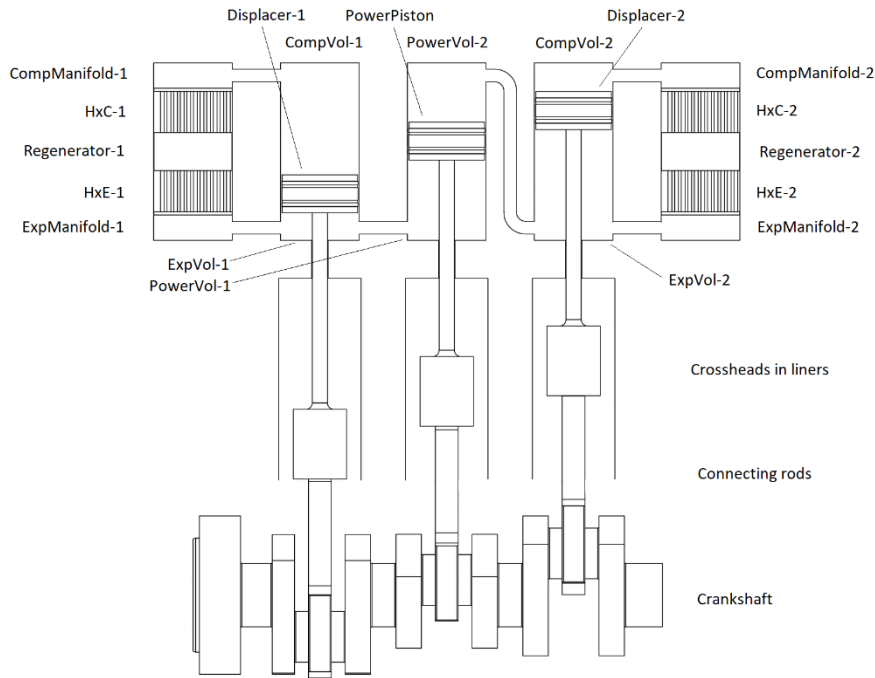


Fig. 3. A circuit pair

circuit consists of a pair of heat exchanger modules, a displacer cylinder, where a displacer piston pumps the working medium through the heat exchangers, and a power cylinder volume, where a power piston compresses and expands the working medium in the whole circuit. The power piston separates two circuits, hence the term double-acting. A pair of circuits is shown in Fig.3. The process components are also referred to in figures 8 and 9. The heat exchanger modules consist of a source heat exchanger and a sink heat exchanger, separated by a regenerative heat exchanger, where heat is stored in steel wire.

The heat pump operates on the Stirling cycle, and due to the cylinder configuration, the two circuits of a circuit pair undergo slightly different processes, as shown in fig. 4.

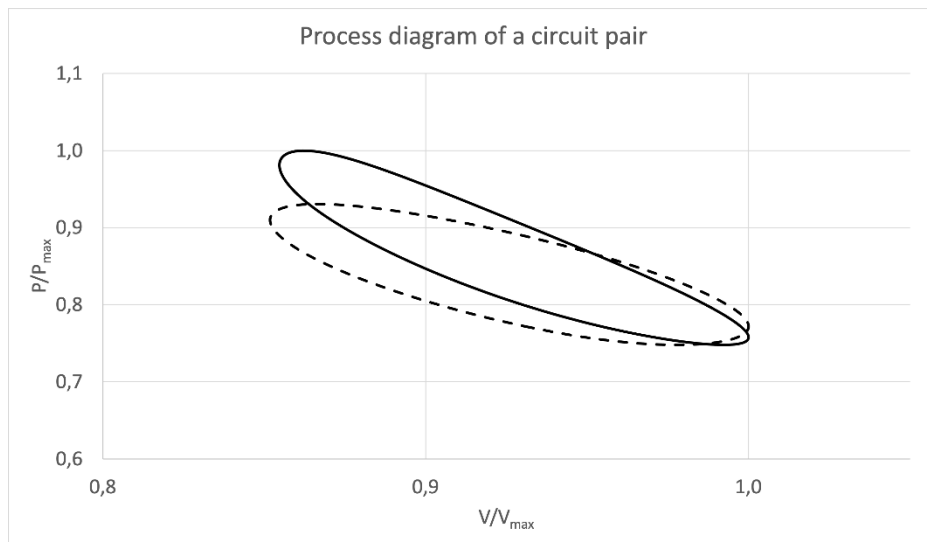


Fig. 4. Process diagram of a circuit pair

The maximum heating capacity of the current implementation of the HoegTemp heat pump is 400 kW. The heat can be generated at any temperature between 20°C and 200°C. The heat pump source can be at any temperature between -10°C and 120°C. If the source temperature is higher than the sink temperature, the heat

pump will work as a heat engine. In future implementations, the source temperature limit may be reduced to minus 250°C, and the sink temperature limit may be increased to +300°C.

Heat is transferred through closed water circuits, both for source and sink. Steam can be generated by a steam generator heated by the pressurized hot-water circuit. The heat pump has separate subsystems for oil lubrication and cooling, working medium handling and process control, diagnostics and data logging.

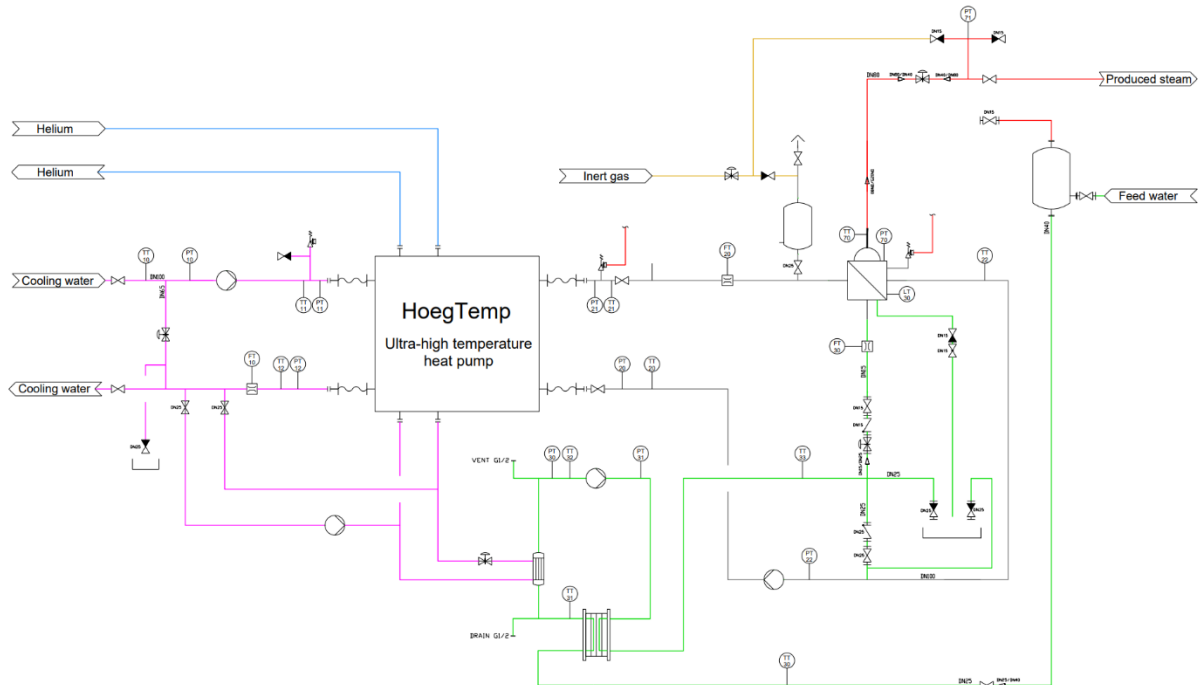


Fig. 5. P&ID of a steam generating installation

### 3. Pilot installation in biogas facility

Fig. 5 shows a P&ID of the steam generating installation. Heat is transferred to and from the heat pump through closed liquid circuits. Water is a very suitable heat transfer fluid at the temperatures of interest, but other fluids can also be used.

The cold or source side circuit (purple lines) uses pure water, and delivers heat from the external source by receiving and returning water at different temperatures. A shunt valve is used to recirculate a fraction of the water, thereby achieving two objectives:

- Control the heat pump inlet temperature
- Vary the mass flow  $\times \Delta T$  relationship independently from requirements set up by the external source

The hot side (sink side, grey lines) circuit uses water under pressure in a closed loop to transfer heat from the heat pump to a steam generator, without any direct contact with the external sink. The steam generator is a shell and plate design heat exchanger, where the hot side (hot side circuit) runs inside the plates, while the cold side (feed water, steam) runs between the outer shell and the plates. A pump is used to circulate the hot side circuit, pump work is predominantly caused by pressure loss in the heat exchangers of the heat pump.

Feed water (green lines) from an external source is pumped into the steam generator at the same rate that water is evaporated. Steam (red line) is finally delivered to the external heat sink at a pressure regulated by a control valve. Heat delivered from the hot side circuit is used for preheating of incoming feed water, evaporation of feed water and to a certain amount superheating of steam. It must be assumed that full crossflow conditions are not achieved in the steam generator.

### 3.1. Instrumentation and measurements

A complete overview of the system from the point of view of measurements is given in figure 6, with descriptions in table 1.

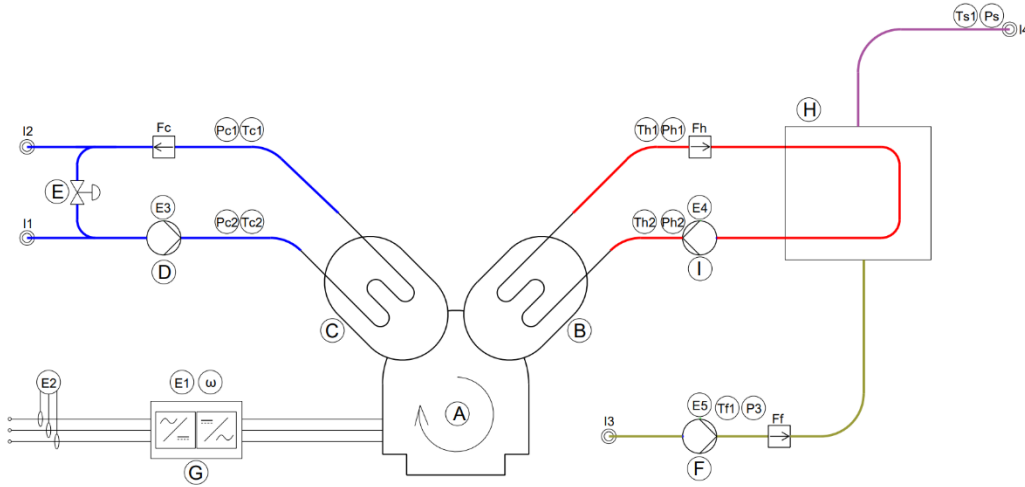


Fig. 6. Instrumentation

Table 1. Instrumentation

A	Heat pump	B	Heat pump hot side heat exchangers
C	Heat pump cold side heat exchangers	D	Cold side circulation pump
E	Shunt valve	F	Feed water pump
G	Variable frequency drive (VFD)	H	Steam generator
I	Hot side circulation pump		
I1	Tempered water from external source	Pc1	Pressure in cold side circuit after heat pump
I2	Tempered water return to external source	Pc2	Pressure in cold side circuit before heat pump
I3	Feed water from external sink	Ph1	Pressure in hot side circuit after heat pump
I4	Steam to external sink	Ph2	Pressure in hot side circuit before heat pump
Tc1	Cold side temperature after heat pump	P3	Pressure of feed water
Tc2	Cold side temperature before heat pump	Ps	Pressure of produced steam
Th1	Hot side temperature after heat pump	Fc	Flow in cold side circuit
Th2	Hot side temperature before heat pump	Fh	Flow in hot side circuit
Tf1	Feed water temperature after pump	Ff	Feed water and steam flow
Ts1	Produced steam temperature	ω	Heat pump speed
E1	VFD produced energy	E2	VFD consumed energy
E3	Hot side circulation pump consumed energy	E4	Cold side circulation pump consumed energy
E5	Feed water pump consumed energy		

### 3.2. Accuracy estimates

The main energy streams are measured as follows:

$$\dot{Q}_x = F_x(h(T_{x2}, p_{x2}) - h(T_{x1}, p_{x2})) \quad (1)$$

If the accuracy of an instrument is generically defined as  $V = \bar{V}(1 \pm \hat{V})$ , where  $\bar{V}$  is the apparent value,  $\hat{V}$  is the relative deviation and  $V$  is the actual value, the total accuracy for the calculated energy flow is

$$\dot{Q}_c = \bar{F}(1 \pm \hat{F}) (\bar{h}_2(1 + \hat{h}_2) - \bar{h}_1(1 + \hat{h}_1)) \quad (2)$$

which becomes

$$Q_c = \bar{F}\bar{h}_2(1 \pm \dot{F})(1 + \dot{h}_2) - \bar{F}\bar{h}_1(1 \pm \dot{F})(1 + \dot{h}_1) \quad (3)$$

The latter can be transformed into

$$Q_c = \bar{F}\bar{h}_2 \left( 1 + \sqrt{\dot{F}^2 + \dot{h}_2^2} \right) - \bar{F}\bar{h}_1 \left( 1 + \sqrt{\dot{F}^2 + \dot{h}_1^2} \right) \quad (4)$$

The following equation can be used instead of eqn. (4).

$$Q_x = \bar{F} \left( (\bar{h}_2 - \bar{h}_1) + \sqrt{\left( \bar{h}_2 \sqrt{\dot{F}^2 + \dot{h}_2^2} \right)^2 + \left( \bar{h}_1 \sqrt{\dot{F}^2 + \dot{h}_1^2} \right)^2} \right) \quad (5)$$

It is assumed there is a linear relationship between temperature and enthalpy for small temperature spans (i.e. the uncertainty range of the temperature sensor)

For the cold side circuit

$$\dot{Q}_c = F_c(h(T_{c2}) - h(T_{c1})) \quad (6)$$

For the hot side circuit

$$\dot{Q}_h = F_h(h(T_{h2}) - h(T_{h1})) \quad (7)$$

The coefficient of performance of the heat pump is calculated as

$$COP = \frac{Q_h}{W_{E1}} \quad (8)$$

The relationship with Carnot efficiency is calculated by the mean temperature of the water-side of hot and cold side heat exchangers

$$COP_{ideal} = \frac{T_{h1} + T_{h2}}{T_{h1} + T_{h2} - T_{c1} - T_{c2}} \quad (9)$$

## 4. Simulated performance

### 4.1. Main energy flows

Figure 7 shows the main energy flows in the heat pump. The electric power  $W_{el}$ , the recycled heat  $Q_L$ , and the useful heat  $Q_H$ . The main losses are electric losses in the variable frequency drive,  $Q_{VFD}$ , the electric motor,  $Q_{motor}$ , and friction,  $Q_{friction}$ .

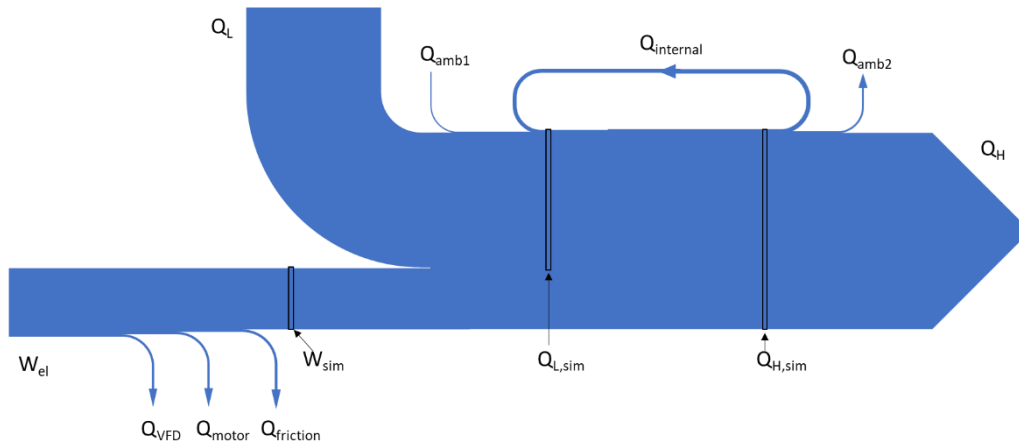


Fig. 7. Sankey diagram

$Q_{motor}$ , friction in the bearings, crosshead, piston ring seals and piston rings,  $Q_{friction}$ , internal heat transfer from hot to cold volumes,  $Q_{internal}$ , and heat exchange with the ambient,  $Q_{amb1}$  and  $Q_{amb2}$ .

#### 4.2. Simulation model

During the development of the process a simulation model of the engine was developed and refined as the design evolved. The simulation model was made in Sage, which is a software package specially designed to simulate Stirling engines. Sage is developed by David Gedeon [1]. The development and use of a simulation model for an earlier heat pump, is described in detail in a previous paper by Tveit et al [2]. Sage is a 1-dimensional finite element model, where each component is divided into a small number of elements (typically 3 to 9 elements each). Inputs to the model are the principal dimensions of each gas volume in terms of cylindrical equivalents, such as diameter, length, number of tubes, Bernoulli factor  $k$ , surface area and wall thickness, and the operating conditions such as shaft speed, average pressure and surface temperatures. The model calculates all heat flows and  $pV$  work. Internal heat transfer may be modelled, but the authors have chosen to model such losses outside of the simulation model.

The root model consists of three pistons, connected to a crankshaft, defined by geometry, the average charge pressure of each circuit, and the circuit model, shown in fig.8. As can also be seen in fig.3 Displacer-1 separates CompVol-1 and ExpVol-1, Displacer-2 separates CompVol-2 and ExpVol-2, and PowerPiston separates PowerVol-1 and PowerVol-2 in the circuit model ‘GammaCircuits’.

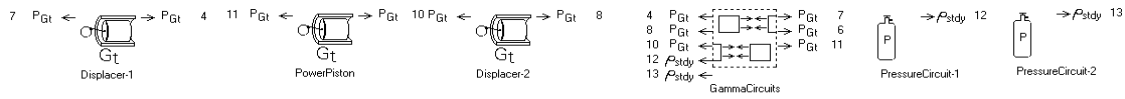


Fig. 8. Main elements of the simulation model implemented in Sage

The circuit model (fig.9) consists of two similar circuits, each with a compression volume and an expansion volume, connected by the heat exchangers and ducts, both circuits connected to each side of the power piston, via gas ducts. The gas flow is denoted with numbers: for Circuit 1, the connections are 6-7-20-21-9-10-29-30, and the components are shown principally in fig.3.

The simulation model takes into account certain losses, e.g., gas friction losses in the tube heat exchangers, hysteresis losses between gas and wall surfaces, but not the mechanical losses and heat losses by convection and radiation of the exposed surfaces. Internal heat transfer between parts are not included in the model either.

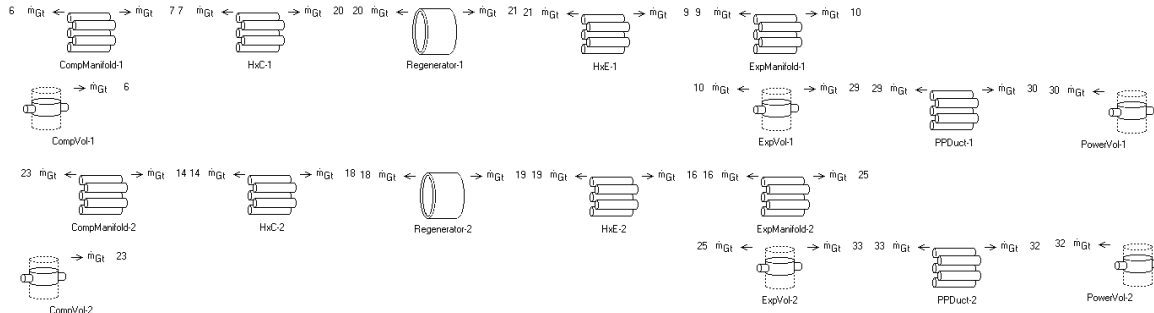


Fig. 9. Circuit model as implement in the software Sage

#### 4.3. Decoupled losses

A model for decoupled losses, was developed, along the methods described by Lundqvist [3]. The main loss factors are defined as:

- Losses in the electric drive train:  $Q_{VFD}$ ,  $Q_{motor}$
- Mechanical losses outside the cylinders:  $Q_{friction}$
- Mechanical losses inside the cylinders:  $Q_{friction}$
- Direct heat transfer from hot to cold heat pump volumes:  $Q_{internal}$
- Heat loss to the ambient:  $Q_{amb1}$ ,  $Q_{amb2}$

Internal heat transfer has been modelled as linear conduction, with temperature gradient and cross section. External heat transfer has been modelled as free convection and radiation.

Mechanical friction has been simulated by Daido Bearings, the bearing manufacturer for the rotating bearings. The crossheads and piston rod seals have been modelled like linear bearings, and the dry piston guide rings and piston rings, have been treated as linear dry bearings. The efficiency of the electric motor and VFD has been found from the specifications.

#### 4.4. Simulated performance at expected operating conditions

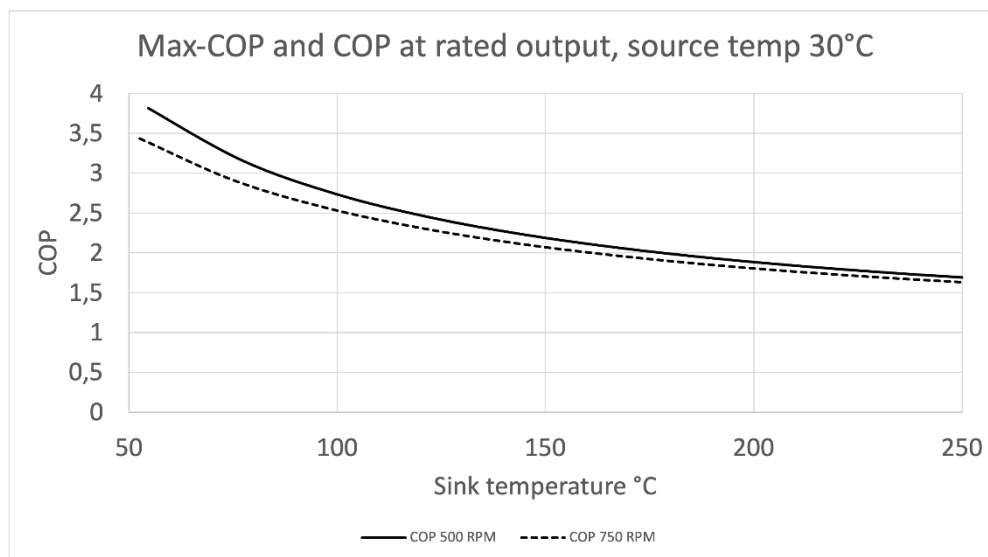


Fig. 10. Simulated COP for the HoegTemp heat pump

Fig. 10 shows the simulated performance of the process, simulated with source temperature 30°C (average between in and out flow temperatures of the water circuit, 35°/25° would be typical) and sink temperatures ranging from 60°C to 250°C (defined as average between in and out flow temperatures of the water circuit,  $\Delta T$  10-20° would be typical), at a shaft speed of 500 RPM, which is very close to the simulated maximum COP of the heat pump, and at a shaft speed of 750 RPM, which is the rated operating point, with a heat output,  $Q_H$ , of 400 kW across the sink temperature range.

## 5. Testing at the biogas facility

During operation, the available heat source temperature varies with the ambient temperature, and with the plant operation. The installation allows a reduced source temperature, as long as it is above 5°C to avoid freezing. The maximum expected source temperature is 50°C in the summer.

The plant needs saturated steam at minimum 2 barG, but the test cell is designed to allow heating of the steam generator with higher temperature hot water, or even a higher delivery pressure, that may be throttled to 2-3 barG. The piping and systems have been designed for 16 barG and 200°C. For testing purposes, it is possible to test the heat pump with sink temperatures below 133°C (2 barG), by cooling the steam generator with cold water.

During early testing, the available source temperatures will be 5-15°C, and heat generation will be tested at 130°C to 180°C. Steam will be generated at both 2-3 barG, and at up to 7 barG.

The installation is not ready at the time of writing, so measurements are not available yet.

## 6. Comparison of simulated results to other high-temperature heat pumps

The simulated performance of the Enerin HoegTemp heat pump is compared to other high-temperature heat pumps in figure 11. For reference, the performance of the stirling cycle heat pump from Single-Phase Power was documented by Høeg et al [4], the performance of heat pumps described in an overview by Bless et al [5], and the peak performance points of cooling machines with natural refrigerants, presented by NTNU [6], have been included. As the Carnot COP and also the COP of a stirling cycle heat pump, depends on the temperature ratio between sink and source temperatures, the chart is presented in terms of temperature ratio rather than the

more common temperature lift. A temperature lift from 5° to 105°C corresponds to 1.36 K/K, while a temperature lift from 70° to 170°C corresponds to 1.29 K/K, and the heat pumps would have different COPs, and different share of Carnot COP.

$$\text{Temperature ratio} = \frac{T_{\text{sink,in}} + T_{\text{sink,out}}}{T_{\text{source,in}} + T_{\text{source,out}}} \tag{10}$$

In order to compare the relative efficiency of the heat pump to other types of heat pumps and cooling machines, the cooling COP relative to the cooling COP of a Carnot heat pump, is plotted against the ratio of sink temperature to source temperature, in fig.12. In the authors' opinion, this ratio more accurately assess the relative efficiency of different cycles, especially at high temperature lifts (large temperature fractions).

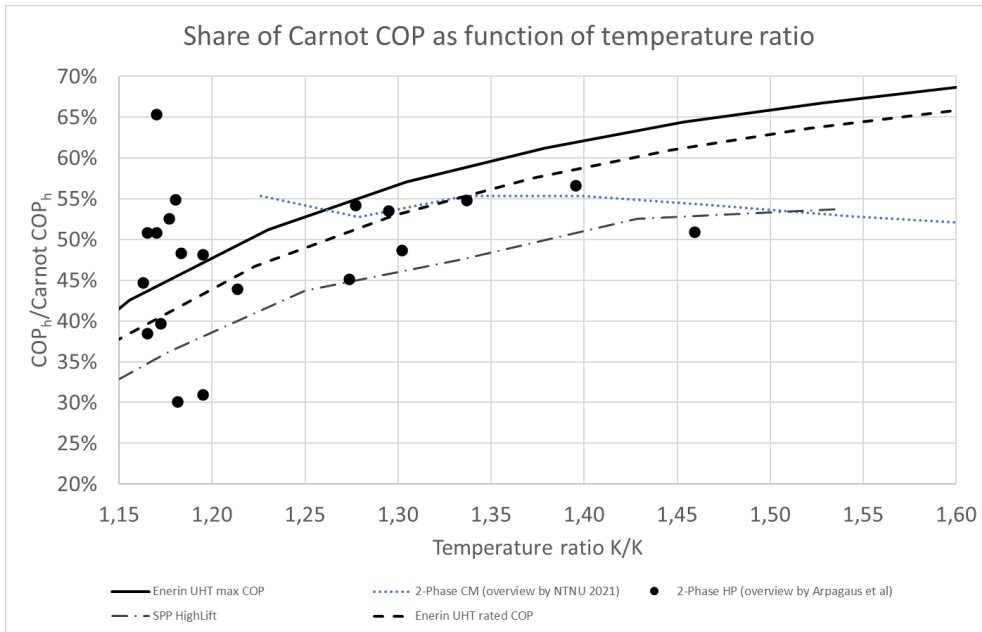


Fig. 12. Share of Carnot COP as function of temperature ratio

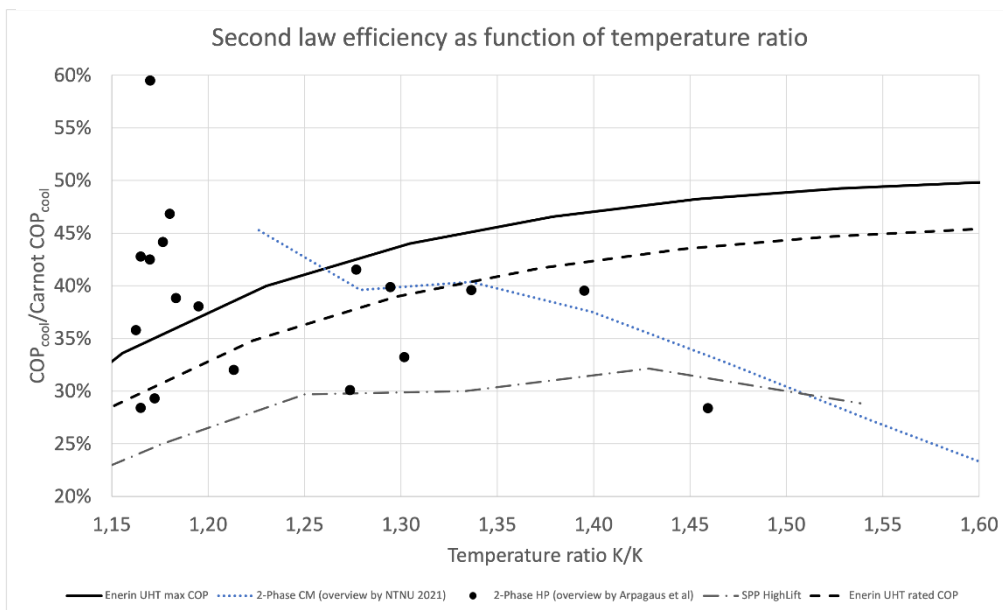


Fig. 11. Second law efficiency of the HoegTemp heat pump, compared to vapour compression heat pumps

## 7. Conclusions

The comparison shows that stirling-cycle heat pumps may be more efficient than vapour compression heat pumps when the ratio of sink to source temperature is above 1.3 K/K. That could be 114°C heating from a 25°C source, or a circuit at 162°/154°C to make 4 bar steam, with a source of 65°/50°C waste heat.

## Acknowledgements

The R&D project has been part-funded by Innovation Norway, and the inter-municipal infrastructure company IVAR has provided the test installation at its biogas facility at Grødaland.

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