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DK-2630 Taastrup
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Appendix 2:

Analysis of SVAF 2 short term tests, TI, 2023.

Pierre-Jean Delêtre

This report presents the results from the different test called *Short Term Tests* (STT) that were conducted on the heat pump of the district heating companies HOFOR, CTR and VEKS in relationship to the project SVAF phase 2 supported by EUDP (the Energy Technology Development and Demonstration Program (EUDP) of Danish Energy Board.

The goal of these tests is to measure the influence of different parameters of the heat pump on its coefficient of performance (COP).

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3.0: Normal operation

During the tests, although the control setup of the heat pump remains constant, some parameters vary and significantly affect the performance of the heat pump, for example the return temperature from the district heating network (DHN) or the source temperature.

In this section the plant in "stable" operation, i.e., without test, is observed, and the parameters that can influence the COP and the way they influence it are selected. The selected parameters allow to *recreate* a "normal COP", using a regression based on measurements, that can be compared to the "test COP", measured during the tests.

The COP (Coefficient of Performance) is defined by the ratio between thermal power provided and electrical power consumed by the heat pump and auxiliaries.

Electrical power

On the electrical side, the overall installation's consumption is almost exclusively determined by that of the heat pumps (HP). Indeed, as seen on Figure 1 the power share of the HP represents almost 95% of the total power consumption (average on the *normal operation* period).

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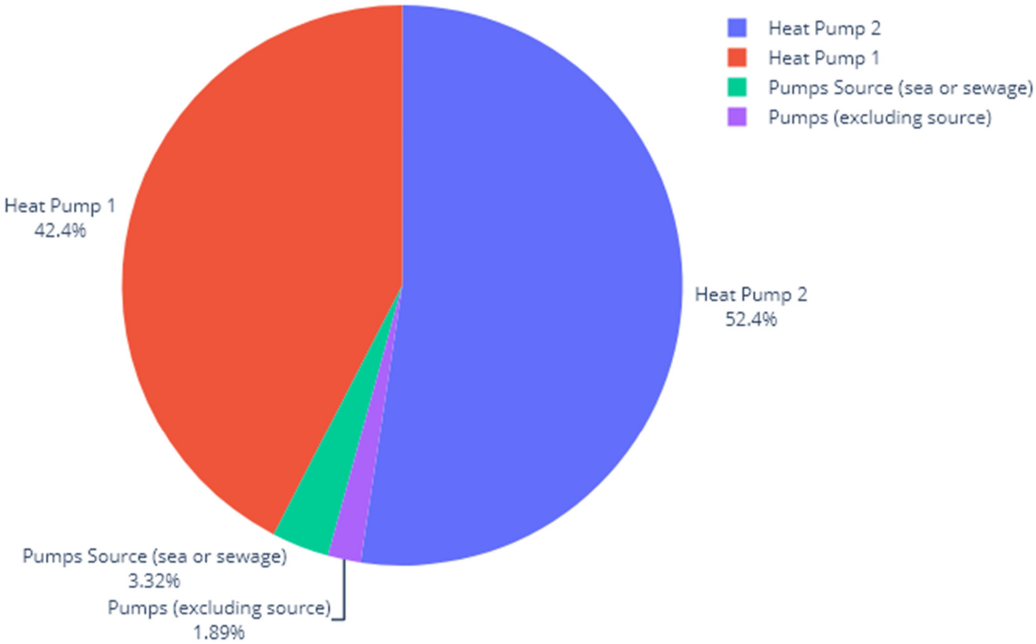

Figure 1: Power consumption on the installation

Inside the HP, it can be considered that the compressors represent the entirety of the power consumption, also because the power variations of the pumps stay relatively contained during operation.

The parameters that affect the compressors' power are therefore the ones of interest to express the electrical part of the COP. The typical variables used to express the compressors' work are the suction and discharge pressures as well as the compressor capacity.

Evaporation temperature

Figure 2 shows that the evaporation temperature (equivalent to the suction pressure) is highly correlated with the source temperature. The source temperature will therefore be used to calculate the regression.

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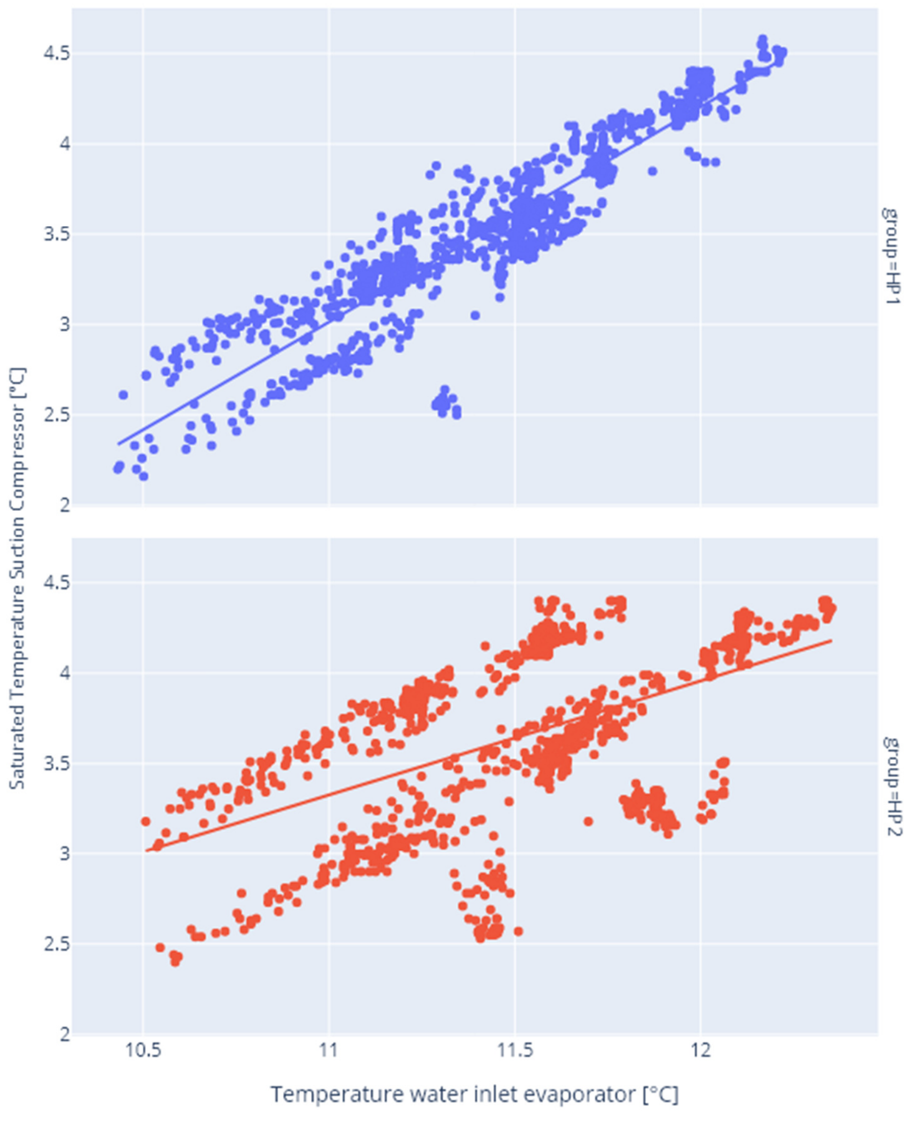

Figure 2: Dependance of the suction temperature on the source temperature

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Different *signals* can be seen, and they correspond to the different setpoints in the source flow in the evaporators. This flow has been slightly changed during the period considered, and therefore affects the relation between inlet temperature (water side) and evaporation temperature (refrigerant side). More details on this can be seen in section 3.5, where a test was conducted specifically on this parameter.

Condensation temperature

On the discharge side, the behavior is different for HP2 and HP1.

Heat Pump 1

For HP1, the condensing temperature is highly correlated to the temperature exiting the condenser of heat pump 1, as seen in Figure 3.

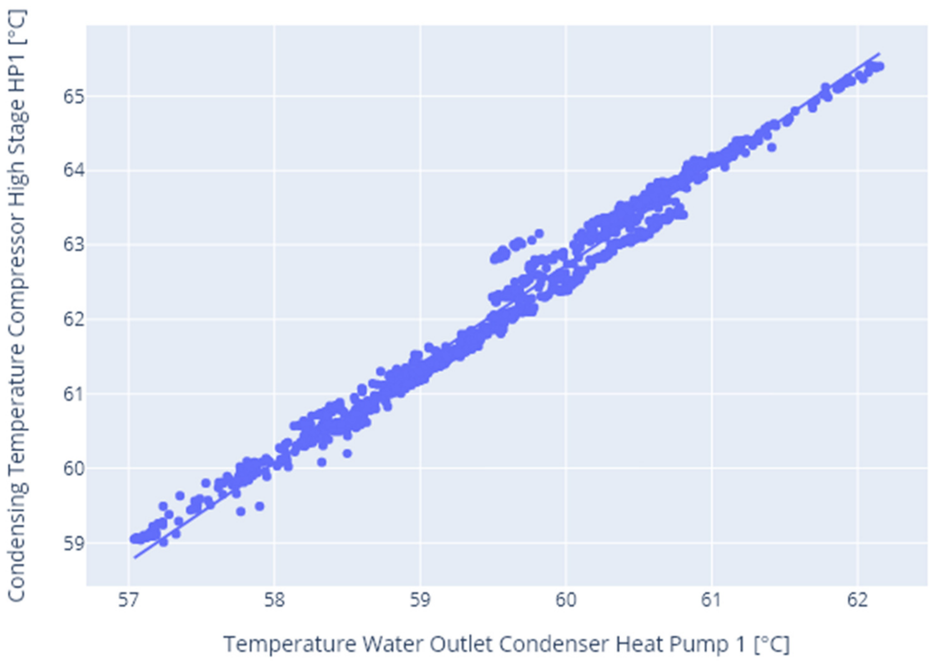

Figure 3: Dependance of the condensing temperature on the network return temperature (HP1)

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This temperature is a setpoint calculated using the average value between supply and return temperature on the district heating network. More specifically, there is a parameter that defines the share of the heat capacity produced by each heat pump, and it is set at 50% during the normal operation. In section 3.4 the variation of this parameter on the COP was studied.

Heat Pump 2

For HP2 the condensing temperature doesn't vary very much, since the supply temperature is a fixed setpoint. However, some variations still exist due to the changes of the load on the DHN side and the dynamics of the system.

Figure 4 presents the variations in the condensing temperature as a function of the variations in the supply temperature.

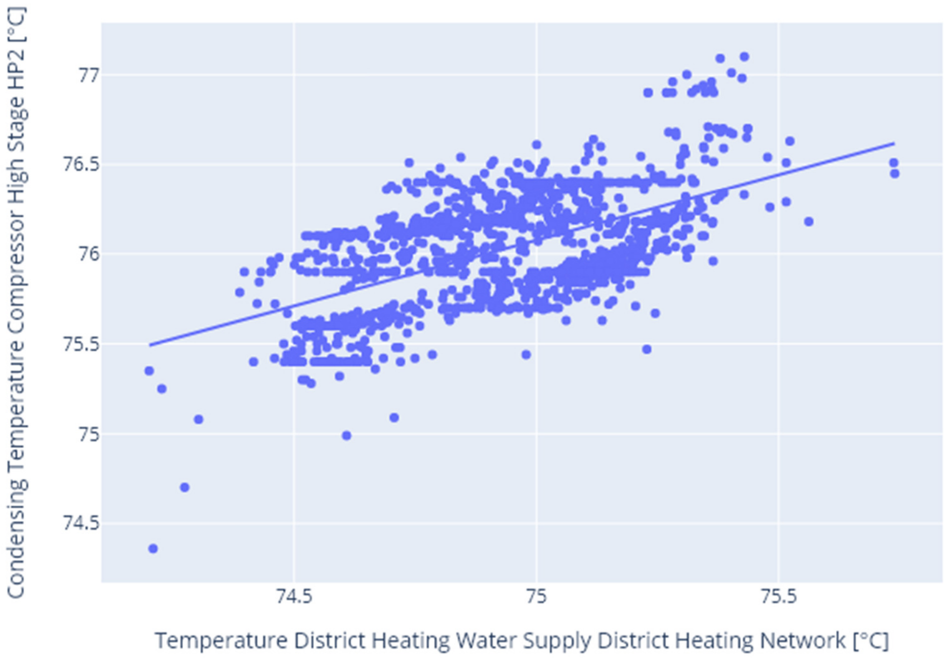

Figure 4: Dependance of the condensing temperature on the supply temperature (HP2)

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Capacity

The compressor speed, defining its capacity, is the last parameter defining the consumption of the compressors.

An example of the influence of the speed on electrical consumption is shown in Figure 5.

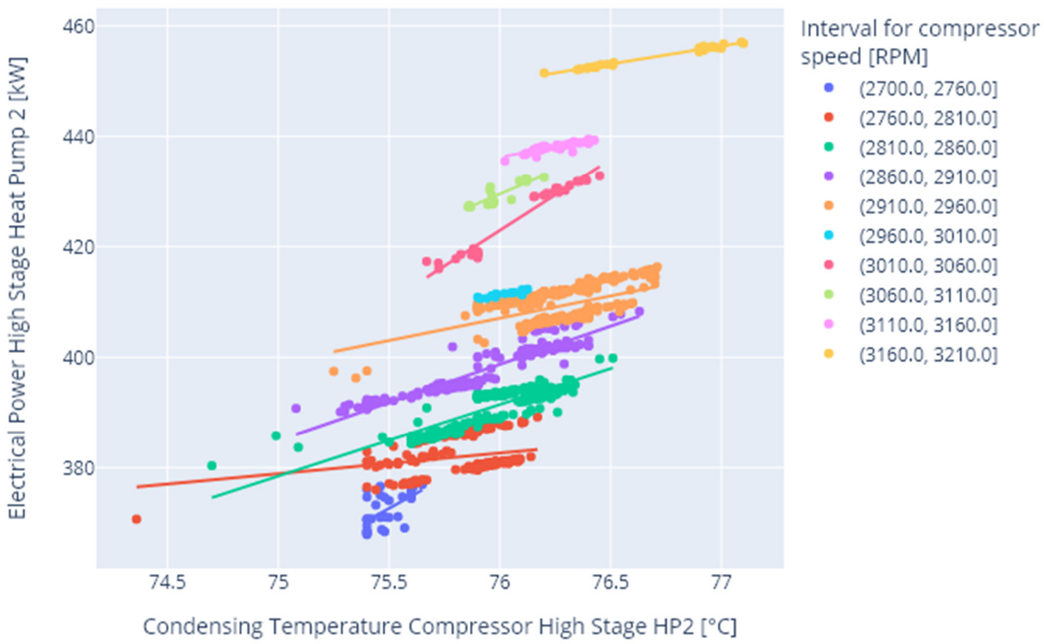

Figure 5: Influence of the speed and condensing temperature on the consumption of the compressor

Speed is an intrinsic parameter to the system, that is, an *output* or *reaction* of the installation, adjusting to other independent variables. It is assumed that the capacity of the compressor is represented by the load on the district heating side: the higher the load, the higher the compressor capacity.

The variables used for the electrical power regression will therefore be:

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- temperature of the source (water inlet evaporator)
- temperature of the water at the outlet of condenser for HP1
- temperature of the supply to the DHN (water outlet HP) for HP2
- heat load of the district heating network

Thermal power

On the sink side, the thermal power provided equals the load of the district heating network (DHN), and can be expressed as the product of the flow in the DHN multiplied by the temperature difference between return and supply.

The thermal power is used in the COP model but only as measurements data since it is considered as an independent variable. Indeed, the thermal power is defined by a setpoint, and the variations around the setpoint are caused by the network. The thermal part of the COP is therefore to some extend unknown and suffered. Only a regression for electrical power consumption is thus calculated.

It should be noted that even though data used are measurements data, the thermal power for each heat exchanger on the sink side is calculated using the water flow and the temperatures at the inlet and outlet. This can lead to some uncertainties due to the precision of the sensors, especially temperature sensors.

Regressions

Thanks to the regression for the electrical power and measurements for thermal power, an estimated COP can be deduced using Equation 1.

$$COP_{estim} = \frac{\dot{Q}_{measure}}{\dot{W}_{estim}} \qquad (1)$$

Figure 6, Figure 7 and Figure 8 hereunder show the measured COP as a function of the estimated COP, calculated thanks to the regression.

The line plotted is the identity (line of slope 1 and origin 0). The closer the points to this line, the better the approximation of the model.

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Figure 6: Comparison of measure and estimation of COP for HP1

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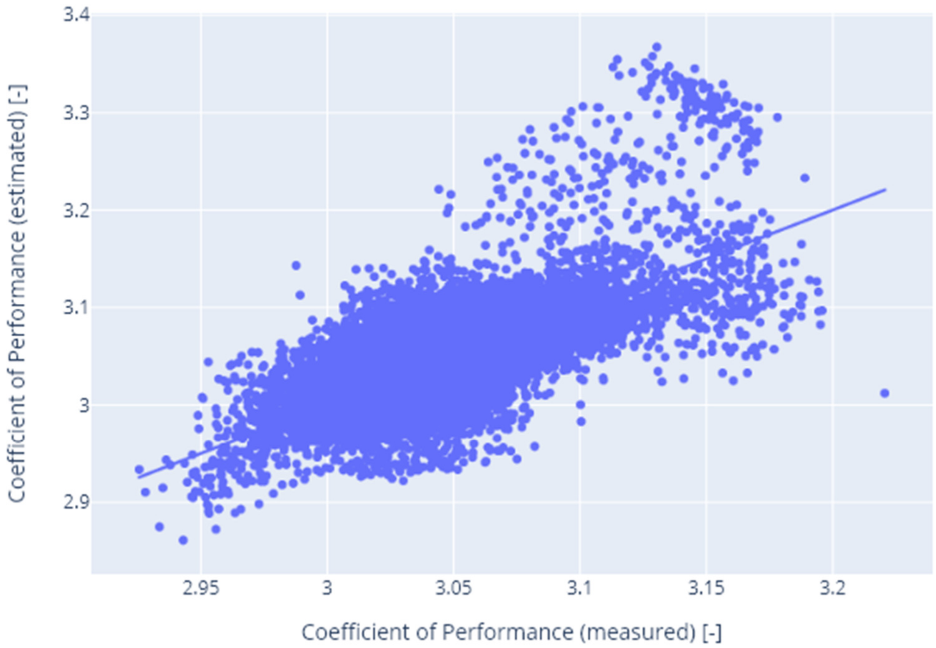

Figure 7: Comparison of measure and estimation of COP for HP2

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Figure 8: Comparison of measure and estimation of COP for both heat pumps

The approximation globally correct but the values of the COP measured have a “noise”, which makes it vary significantly from minute to minute.

These strong variations are caused by the fluctuations of the flows which happen extremely fast compared to the changes of the temperatures, and generate a noise in the thermal power data.

Figure 9 presents the variations of the flow at the same time as variations in the temperature on the district heating side. A similar phenomenon can be seen on all heat exchangers.

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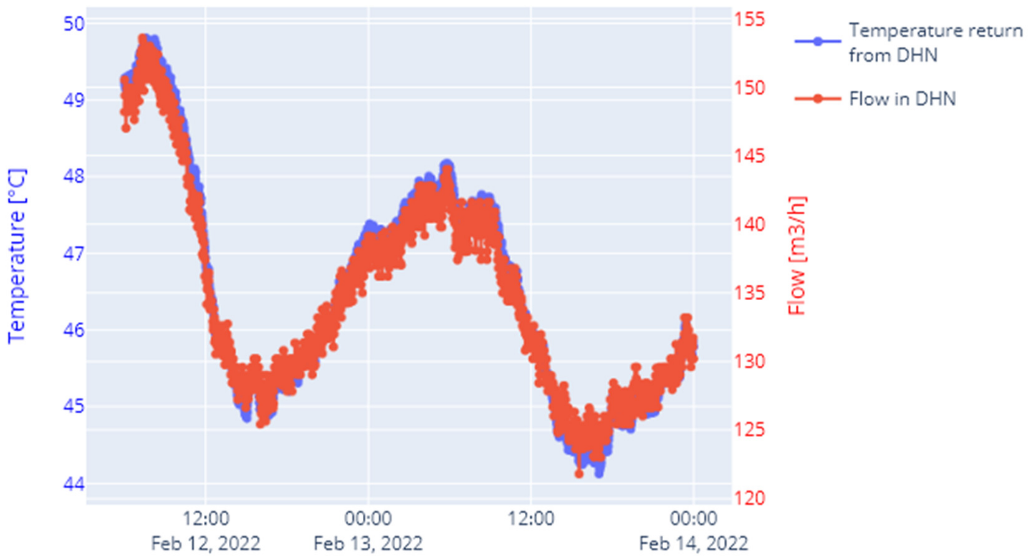

Figure 9: Fluctuations of the flow and temperature

On an aggregated signal (average on bigger time intervals for example), the noise of the flow is smoothened and the regressions are more performant, as shown on Figure 10.

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Figure 10: Comparison of measure and estimation of COP for HP (resampled with 10-minute average)

By using an averaged signal on each level for the different parameters of the tests, the approximation given by the regression gives a precision of COP of ±0.1, which corresponds to an error of ±5%.

This uncertainty is quite large and will make the model hard to use, since really big performance differences will be required to conclude on the effects of the tests.

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3.3: Intermediate saturated temperature COP sensitivity for HP1 and HP2

The parameter that is changed during this test is the intermediate (i.e., intercooler) pressure and therefore saturated temperature between the 2 stages of each HP. The GEA OMNI interface allows to change a value between 0 and 100% corresponding respectively to the minimum and maximum values in the range adjustable for this temperature. This will affect the share of the work made by one or the other of the compressors

The tests were conducted twice, in February 2022 and in January 2023, due to different conditions on the source side (evaporators in series or parallel).

First test (21/01/2022-28/01/2022)

Conditions of the test

The following conditions could be observed during the test:

1. The evaporators are in **series** (i.e., HP2 before HP1)
2. The source is the **sewage water**
3. Variations of temperature on the source side are **moderate** (around 11.5°C to 13°C)
4. Variations of temperature on the DHN side:
   - Return temperature: **important** (42.5°C to 48°C)
   - Supply temperature: **low** (74.5°C to 75.5°C)

Results

The tests were run alternatively on HP1 and HP2.

Heat Pump 1

For the heat pump 1, it seems impossible to stably work at high intercooler temperatures (load factor above 50%).

Indeed, when looking at the data, presented in Figure 11, temperatures fluctuate a lot and do not stabilize for tests values of 75% and 100%.

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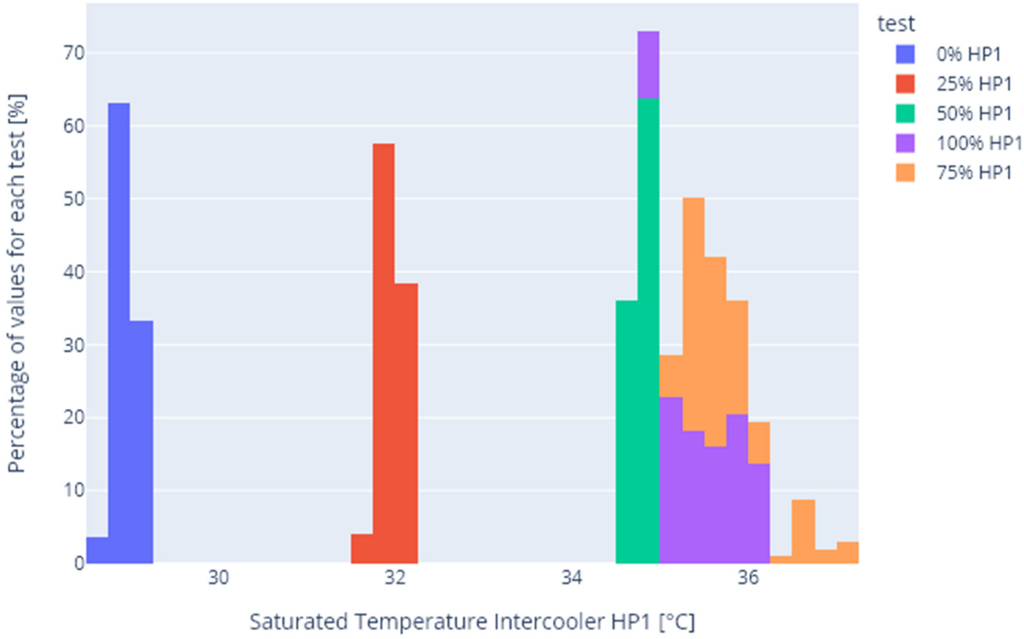

Figure 11: Repartition of the saturation temperatures in the intercooler for HP1

The range of values for tests with 75% and 100% is wider, indicating oscillations, while the average value is similar to that of the test with 50% (34.5 °C).

In those conditions, it is difficult to have a good overview of the effect of the load factor on HP1 for values above 50%.

Figure 12 presents the COP gain/loss between measurements and estimation by the regression.

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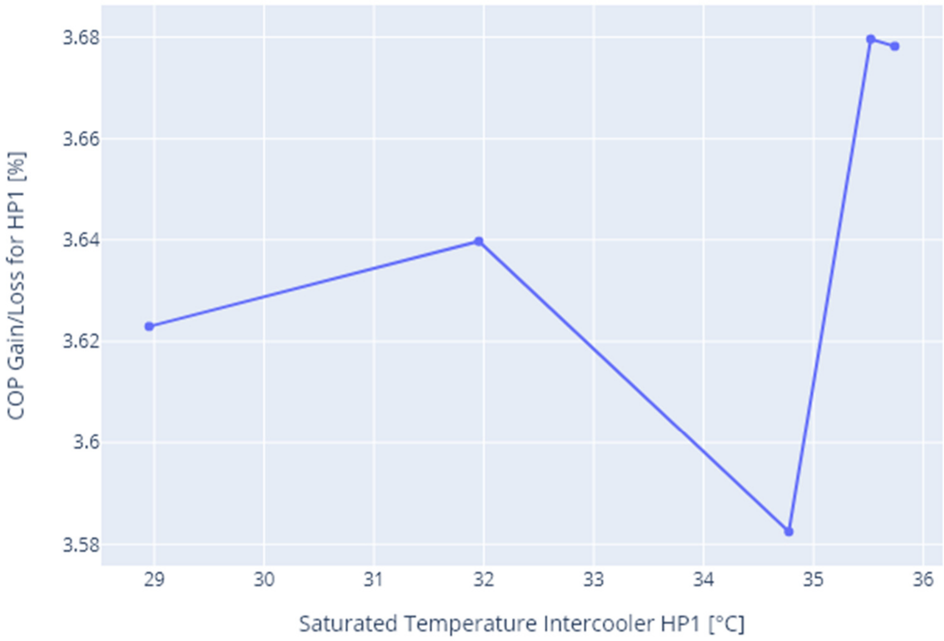

Figure 12: COP HP1 gain/loss between measured and expected

The numerical values for the heat pump 1 can be estimated and are presented in Table 1.

Table 1: COP HP1 gain/loss between measured and expected

Load factor	Intercooler temperature [°C]	Expected	Measured	Gain
0%	28.95	3.62	3.62	0.04%
25%	31.95	3.60	3.64	1.01%
50%	34.83	3.53	3.58	1.52%
100%	35.52	3.63	3.68	1.44%
75%	35.74	3.63	3.68	1.41%

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Due to the problem with high load factors, their corresponding average intercooler temperature (second column) is only slightly higher than that for the normal operation (50%) and the average value for 100% is lower than for 75%.

The COP changes between estimation and measurements are too small to conclude for this test.

Heat pump 2

For HP2 all the different load factors tested actually correspond to different saturated temperatures in the intercooler. Figure 13 shows the distribution of temperatures for the different tests conducted.

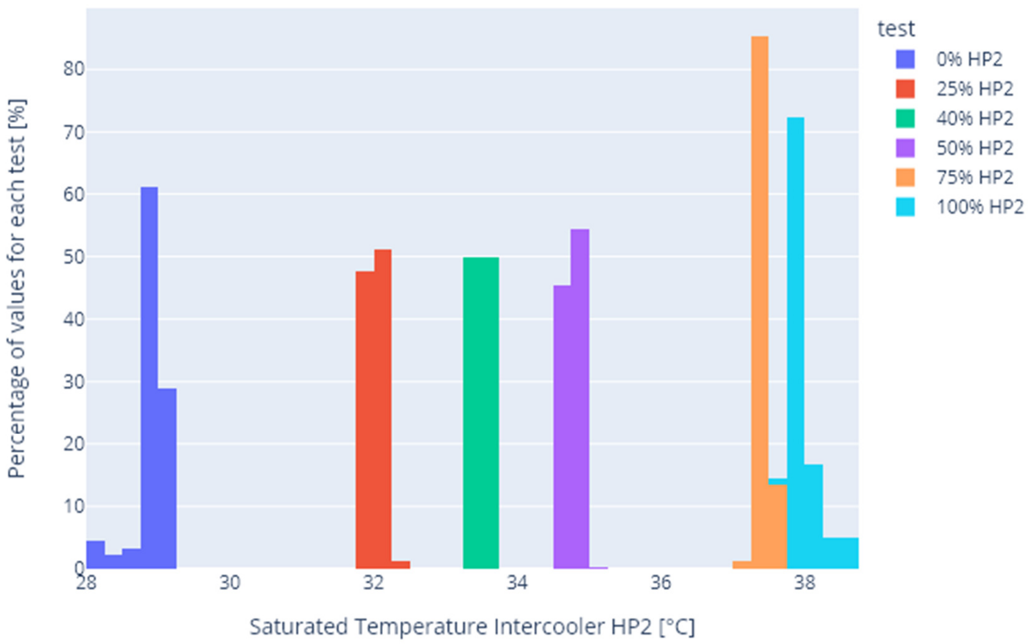

Figure 13: Distribution of the saturation temperatures in the intercooler for HP2

The difference between the measured COP and the expected COP (from the regression) is plotted in Figure 14.

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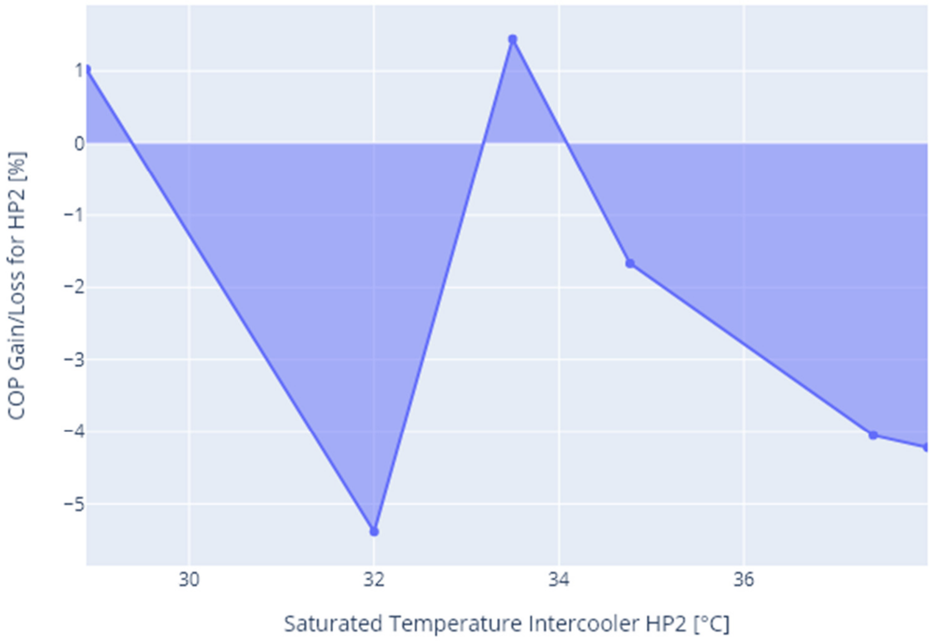

Figure 14: COP HP2 gain/loss between measured and expected

The numerical values are presented in Table 2.

Table 2: COP HP2 gain/loss between measured and expected

Load factor	Intercooler temperature [°C]	Expected	Measured	Gain
0%	28.89	3.06	3.09	1.02%
25%	32.00	3.26	3.09	-5.39%
40%	33.50	3.05	3.10	1.44%
50%	34.07	3.19	3.14	-1.67%
75%	37.40	3.28	3.15	-4.05%
100%	37.99	3.29	3.15	-4.22%

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The values are difficult to interpret, since the signal significantly changes for slight changes in the parameters.

Second test (21/01/2023-28/01/2023)

Results

Heat pump 1

Figure 15 presents the repartition of the saturated temperatures in the intercooler and the same phenomenon as for the first test can be observed: above 50%, the saturated temperature doesn't seem to increase.

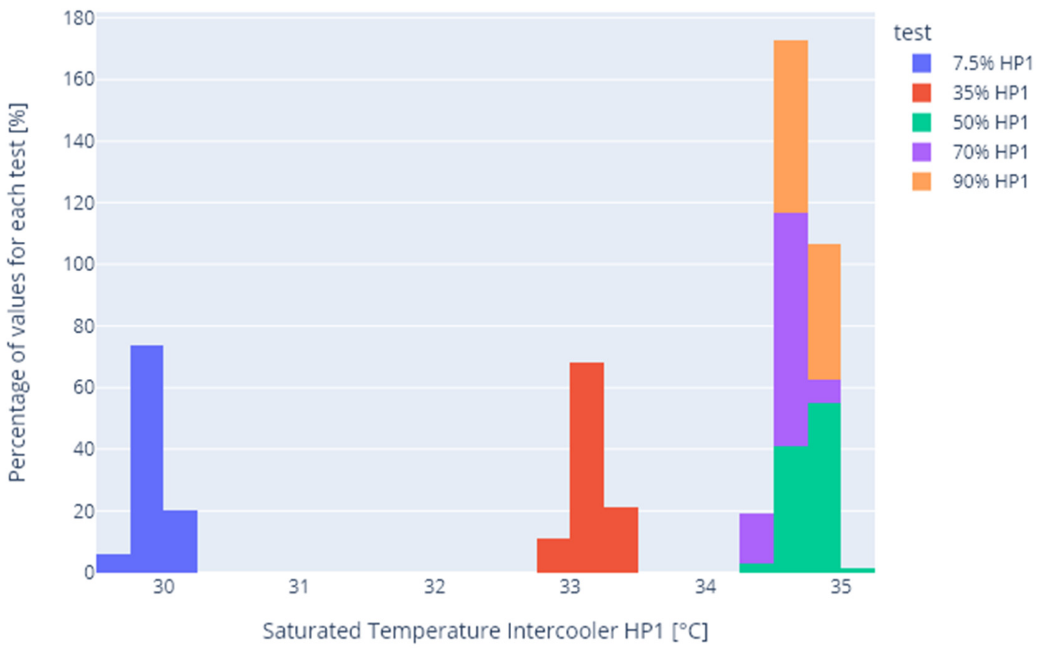

Figure 15: Repartition of the saturation temperatures in the intercooler for HP1

Figure 16 presents the COP gain/loss between measurements and estimation by the regression.

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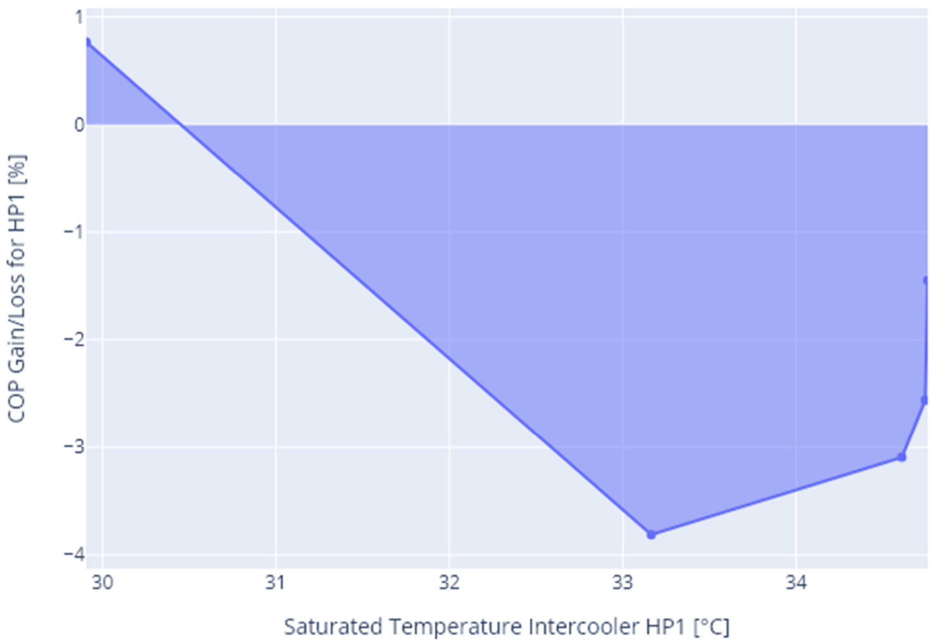

Figure 16: COP HP1 gain/loss between measured and expected

The numerical values are presented in Table 3.

Table 3: COP HP1 gain/loss between measured and expected

Load factor	Intercooler temperature [°C]	Expected	Measured	Gain
7.5%	29.91	3.74	3.77	0.77%
35%	33.16	3.85	3.71	-3.81%
70%	34.61	3.85	3.73	-3.09%
90%	34.74	3.82	3.73	-2.56%
50%	34.76	3.77	3.72	-1.45%

Once again, the differences between estimation and measurements don't allow to conclude.

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Heat pump 2

Figure 17 presents the distribution of the saturated temperatures in the intercooler for HP2 during the test.

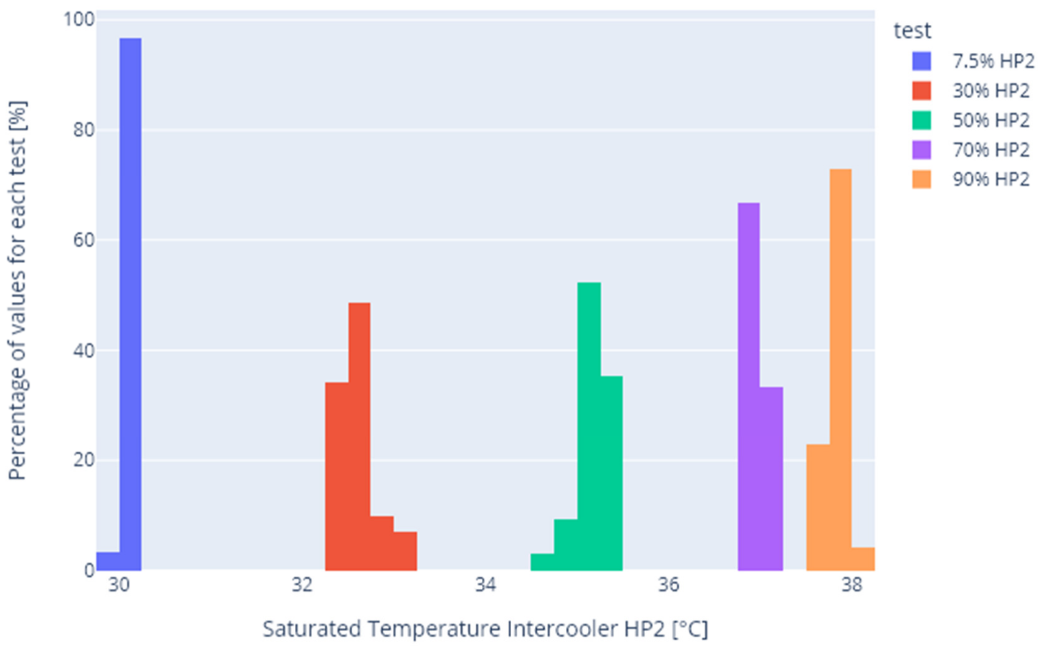

Figure 17: Repartition of the saturation temperatures in the intercooler for HP2

Figure 18 presents the COP difference between measured and expected COP for HP2.

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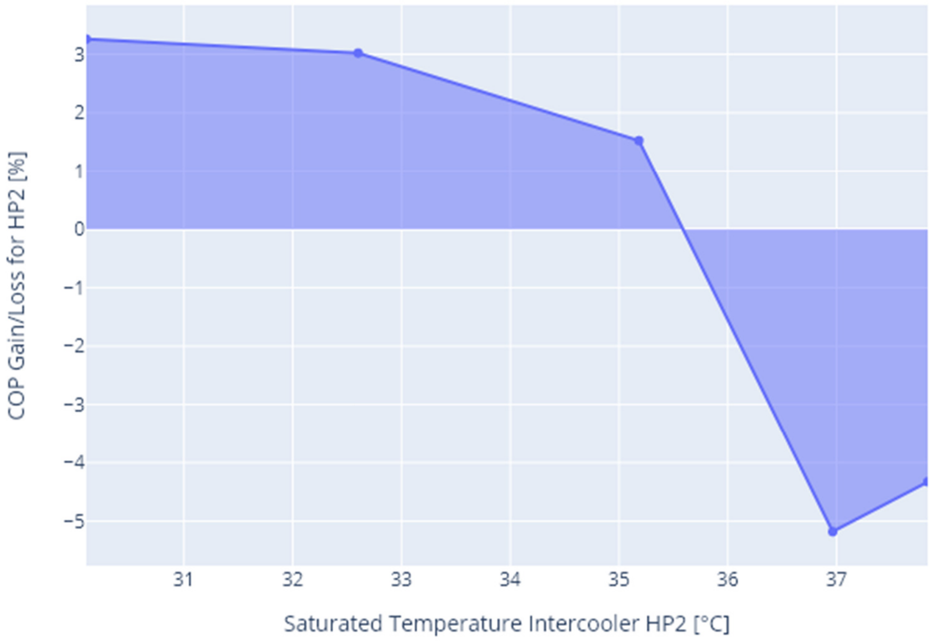

Figure 18: COP HP2 gain/loss between measured and expected

The numerical values associated with the graph are presented in Table 4.

Table 4: COP HP2 gain/loss between measured and expected

Load factor	Intercooler temperature [°C]	Expected	Measured	Gain
7.5%	30.10	2.98	3.08	3.26%
30%	32.60	2.98	3.07	3.02%
50%	35.18	3.01	3.06	1.53%
70%	36.96	3.19	3.03	-5.18%
90%	37.84	3.16	3.03	-4.33%

It seems from the values that the intercooler pressure should be as low as possible for HP2. However the uncertainty of the model limits the interpretation of the results.

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Conclusion

The use of the regression didn't help to evaluate the effects of the parameter on the COP of the heat pump. However some relevant observations could be made.

For HP1 there seems to be a upper limit to the temperature of the intercooler since for values of the load factor above 50% the intercooler temperature doesn't change much and cannot stabilize.

It can be noticed that the setpoint value for both heat pumps is similar (34.5 °C), while the pressure levels are different, especially on the condensation side. In theory, the optimal intermediate pressure for two stages is given by Equation 2.

$$P_{inter} = \sqrt{P_{low} \times P_{high}} \qquad (2)$$

According to this rule, the intermediate pressure for HP2 is optimal while that for HP1 is too high (a saturated temperature around 32 °C would be optimal). Graphs showing the values are presented in appendix.

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3.4: HP1 setpoint sensitivity on COP

In this test the influence of the setpoint temperature of the district heating water between the two heat pumps is studied. This temperature is the one exiting the condenser of HP1.

The value of this temperature cannot be set directly, since it is permanently recalculated depending on the return temperature. Nevertheless it can be influenced using the percentage of each heat pump in the total capacity: if the share of HP1 is higher, the temperature will be higher.

Results

Figure 19 shows the gain/loss of the COP for the overall heat pump (HP1 & HP2) subject to different temperatures out of the condenser of HP1.

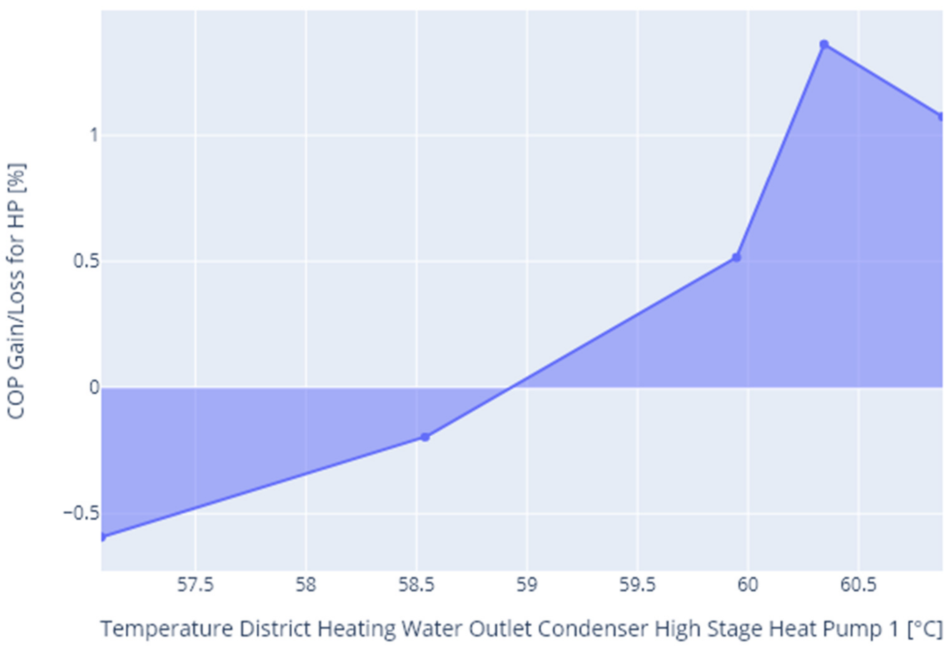

Figure 19: COP HP gain/loss between measured and expected

The numerical values are presented in Table 5.

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Table 5: COP HP gain/loss between measured and expected

Load factor	Condenser HP1 outlet temperature [°C]	Expected	Measured	Gain
44%	57.07	3.36	3.34	-0.59%
50%	58.54	3.34	3.34	-0.20%
53%	59.95	3.25	3.27	0.52%
55%	60.34	3.23	3.27	1.36%
57%	60.88	3.21	3.24	1.07%

It is difficult to conclude to a clear tendency because the changes observed are quite low (<1.5%).

It can be interesting to see the actual capacity share for HP1 corresponding to the setpoint parameter, the latter being a temperature, and therefore not necessarily matching capacities. Figure 20 presents the values measured.

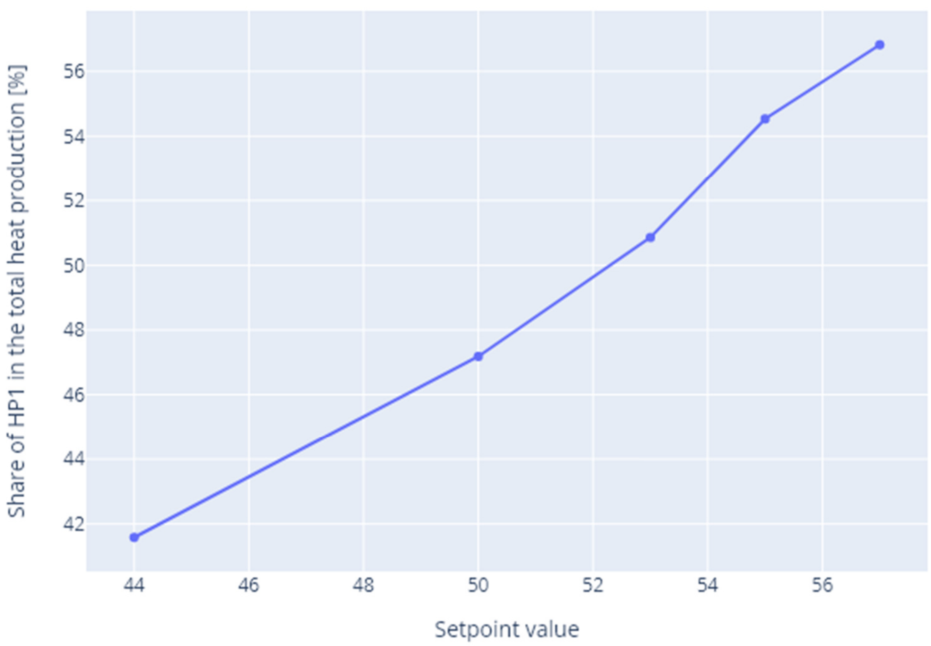

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Figure 20: Share of HP1 in the heat production relative to the setpoint value

Due to the position of the various heat exchangers of each heat pump, the setpoint value actually doesn't exactly match the actual heat capacity. A share of 50% of the production is measured for a parameter value around 52%.

Conclusion

The observed differences being low between measurements and calculation, it is difficult to conclude for this test. However, the tendency of the results shows that a parameter value of 55% seems to be giving the highest performance, meaning to have a higher temperature exiting the condenser of HP1.

Also, when looking at the actual share of thermal power from each heat pump, there seems to be a small distortion between the parameter (based on a temperature) and the measurements. If the load should be equally shared between the heat pump, a setpoint value around 52% or 53% should be used.

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3.5: Source flow sensitivity on COP

The purpose of the test is to get a graph over the sensitivity of the changes of flow on the source side on the overall COP.

To modify the flow, the temperature difference between inlet and outlet of the evaporators is adjusted: the higher the temperature difference, the lower the flow.

Results

The overall COP is estimated thanks to the regressions showed in section 3.0 and a estimated value for the pumps power based on the flows on the source side and the district heating side.

To make the comparison relevant, the flow on the source side for the regression in this test was taken equal to 620 m3/h, which corresponds to the setpoint value before the test was run (08.03.2023).

Figure 21 shows the gain/loss of the COP for the overall system (HP1, HP2 & pumps) as a function of the sewage water flow in the evaporators. The estimated COP is calculated for a flow of 620 m3/h.

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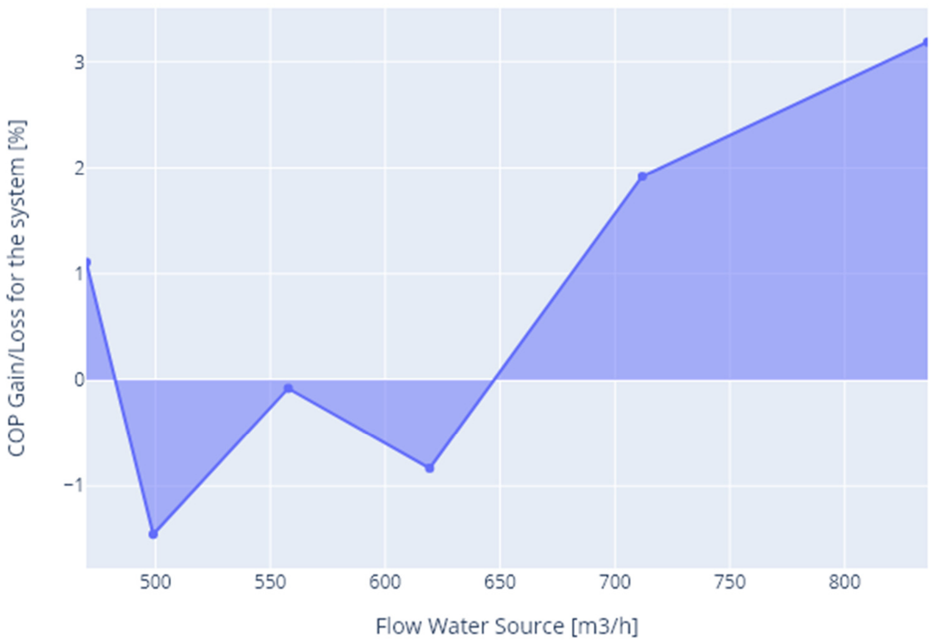

Figure 21: COP gain/loss for the system between measured and expected

Table 6 presents the numerical values.

Table 6: COP HP gain/loss between measured and expected

Temperature difference	Water flow in both evaporators [m3/h]	Temperature source [°C]	Expected	Measured	Gain
5.5K	469.87	11.35	3.11	3.14	1.11%
5K	499.02	10.61	3.14	3.09	-1.45%
4.5K	557.77	10.55	3.15	3.15	-0.08%
4K	619.33	10.79	3.13	3.10	-0.83%

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Temperature difference	Water flow in both evaporators [m3/h]	Temperature source [°C]	Expected	Measured	Gain
3.5K	711.92	10.78	3.08	3.14	1.92%
3K	836.11	10.85	2.94	3.03	3.18%

The potential gains observed between estimation and measure are inside the error margin so it is difficult to conclude based on these values.

In this test it can be considered that only the low stage power consumption (low stage compressors and pumps) is influenced since the intercooler temperature is fixed. Figure 22 presents the electrical consumption for the different equipment on the source side for the different flows studied.

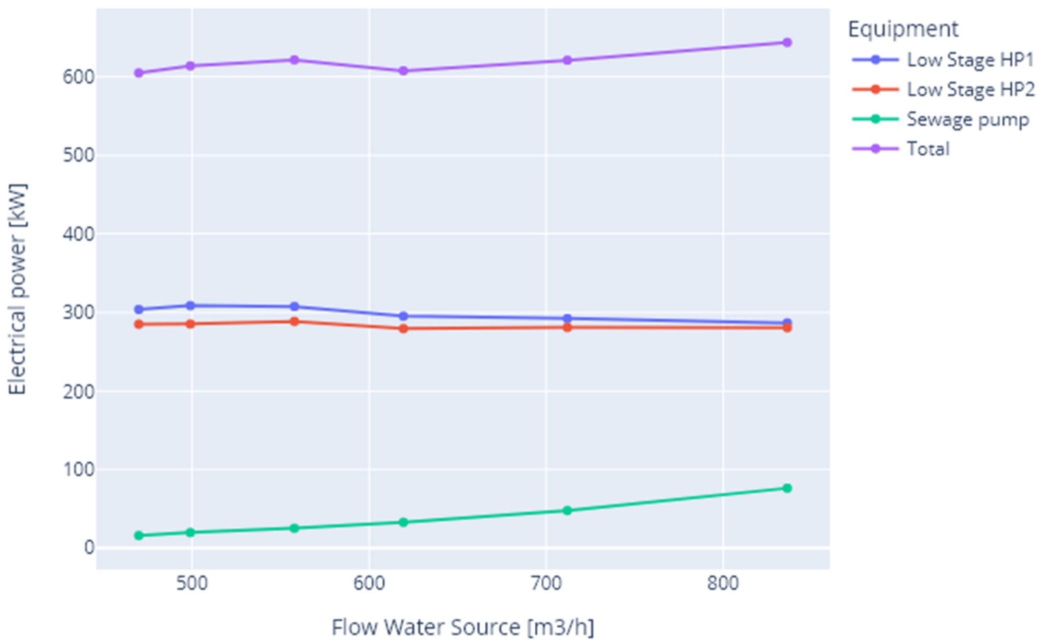

Figure 22: Electrical power on the source side (sewage pump and low stage compressors)

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The results should be interpreted with care because the source temperature is varying on the period and therefore affecting the consumption of the compressors. However, it can be seen that the total consumption is highly increased for higher flows, due to a much higher pump work.

The power consumption of the sewage water pump as a function of the source flow is presented in Figure 23.

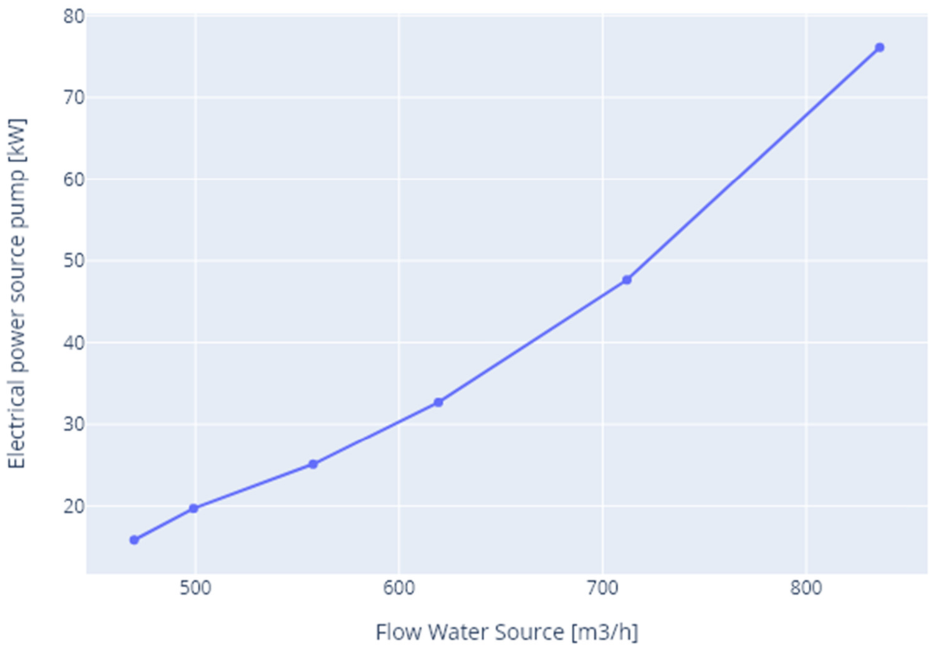

Figure 23: Electrical power of the source pump as function of the source flow

Between the reference at 620 m3/h and the maximal flow, the electrical consumption is more than doubled (32 kW to 76 kW), which strongly impacts the overall COP.

On the compressor side a higher water flow means a higher suction temperature. The temperature difference between the inlet of the evaporator and the evaporation temperature can help see the effect of the increased flow through the evaporators, and is presented in figure Figure 24.

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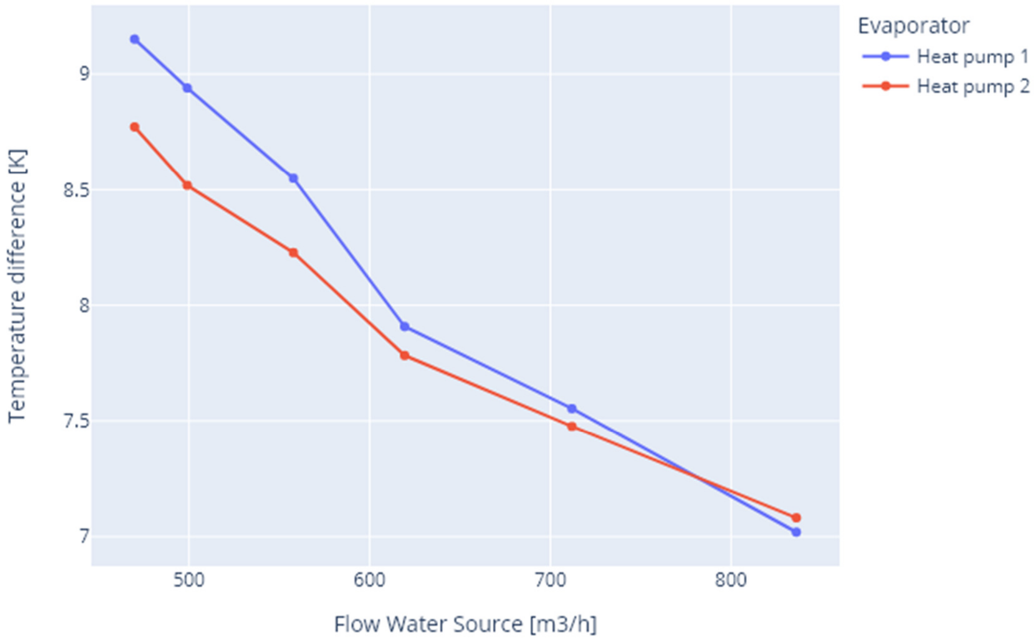

Figure 24: Temperature difference between water at the inlet of the evaporator and evaporation temperature

The gain between the lowest flow (470 m3/h) and the highest (840 m3/h) is about 2K: the evaporation temperature is 2K higher for the highest flow. This causes the electrical consumption of the compressors on the low stage of the heat pumps to drop but for high flows this reduction is lower than the increase in pump work, hence the increase in the total consumption seen in Figure 22.

More detailed graphs present the electrical consumption observed for variations in the evaporation temperature in the appendix.

Conclusion

The temperature changes on the source side make it difficult to find the optimal flow to minimize the combined consumption of both pumps and low stage compressors, but it appears that flows

Gregersensvej 2
DK-2630 Taastrup
Telefon 72 20 20 00

info@teknologisk.dk
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above 650 m3/h cause the pump power to increase much more than the compressors' power to decrease.

The optimal flow seems to be around 600 m3/h, which corresponds to a temperature difference of 4K between the inlet and outlet of the evaporators.

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3.7: HP1-LS: Desuperheater DH flow sensitivity on COP

During this test the influence of the water flow through the desuperheater (DSH) of the low stage of HP1 on the COP is studied.

Results

The thermal power in the desuperheater of HP1-LS is mainly affected by 3 variables:

- the return temperature from the DHN (see Figure 25)
- the speed of the compressor (mass flow of ammonia) (see Figure 26)
- the flow of water through the heat exchanger (parameter studied in this test)

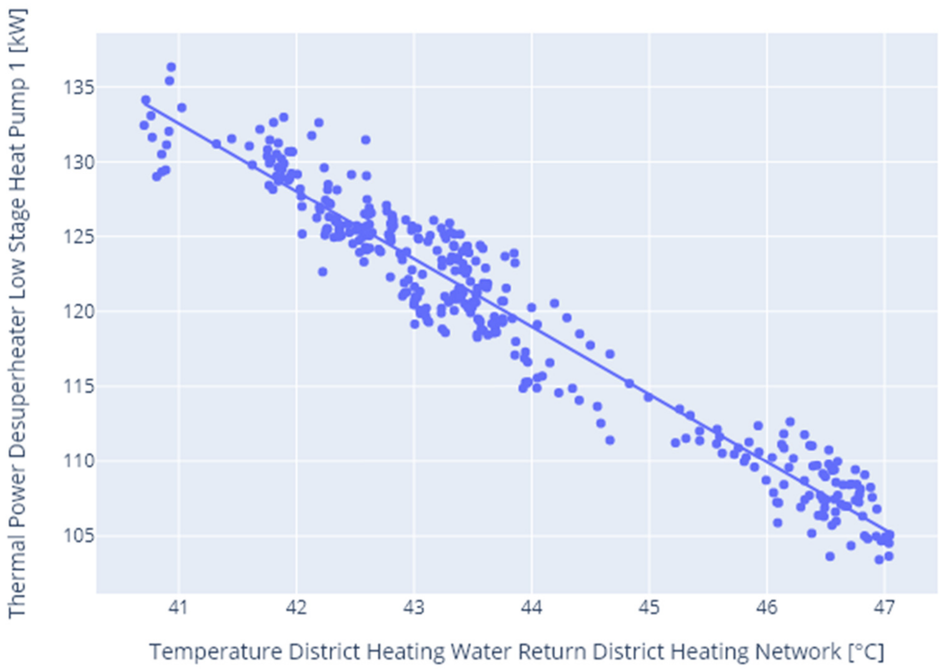

Figure 25: Thermal power of the desuperheater of HP1-LS: evolution with DH water temperature

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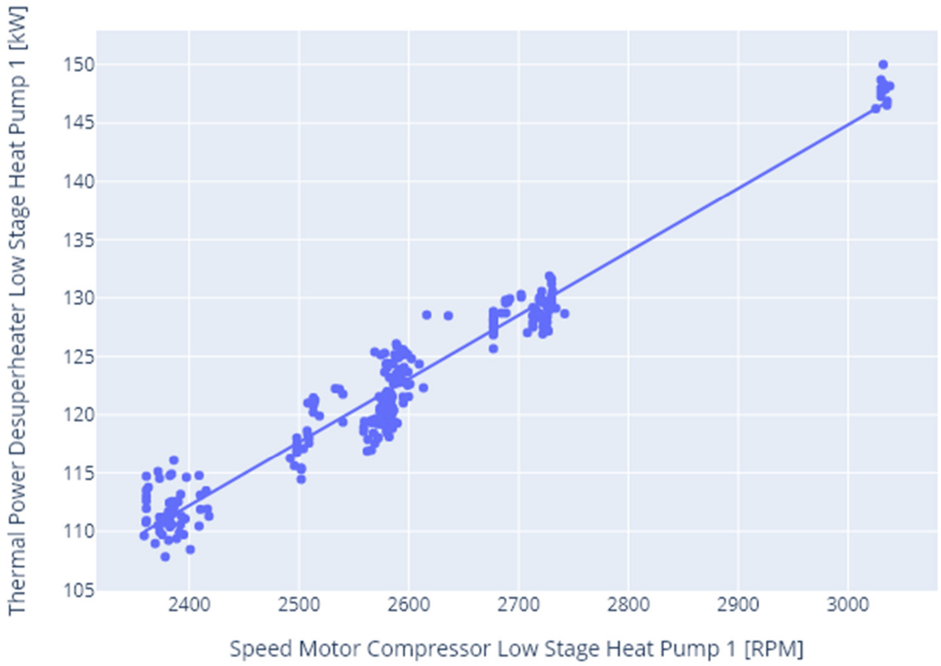

Figure 26: Thermal power of the desuperheater of HP1-LS: evolution with compressor speed (ammonia flow)

The discharge temperature on the ammonia side remains more or less constant, changing a little with the oil temperature, which is dependent on the water return temperature. This variable is therefore considered to have a negligible impact on the thermal power of the heat exchanger, especially since the ammonia is on gas phase, and therefore has a low calorific capacity.

Figure 27 shows the thermal power of the desuperheater of HP1-LS. The variations of the return temperature were low during the test, so the values didn't need to be filtered in different bins. The compressor speed, however, varied substantially and was therefore filtered to keep values between 2425 RPM and 2435 RPM.

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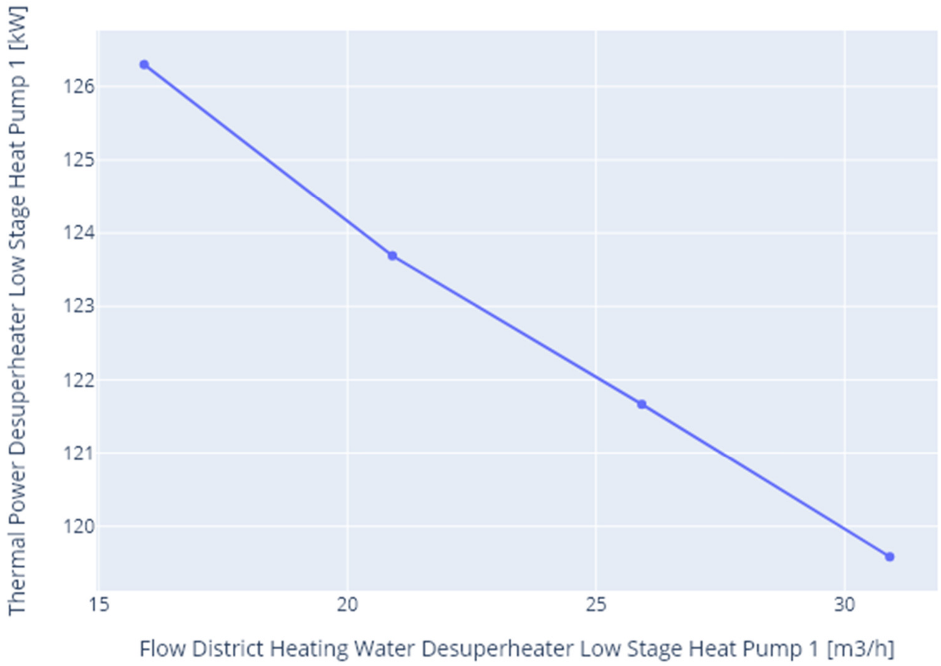

Figure 27: Thermal power of the desuperheater of HP1-LS: evolution with water flow through the heat exchanger

Surprisingly, the thermal power of the heat exchanger is higher for a lower flow. This result cannot be explained just by the changes in the flow, since a flow increase always causes the thermal power to increase, due to higher thermal transfer coefficients.

The difference between the highest and the lowest thermal powers is about 7 kW in the conditions of the test, which represents a variation of about 5%. This small variation could be explained by other factors than the flow such as slight changes in the water temperature at the inlet or mass flow rates on the ammonia side (speed were filtered but the mass flow can be slightly different due to different densities of the refrigerant with different pressures).

This gain is very small compared to the heat capacity of the heat pump and a gain in terms of COP would be impossible to detect, hence the approach chosen of looking up closely on the heat exchanger.

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The temperatures for the ammonia and the water at different positions of the heat exchangers are presented in Figure 28.

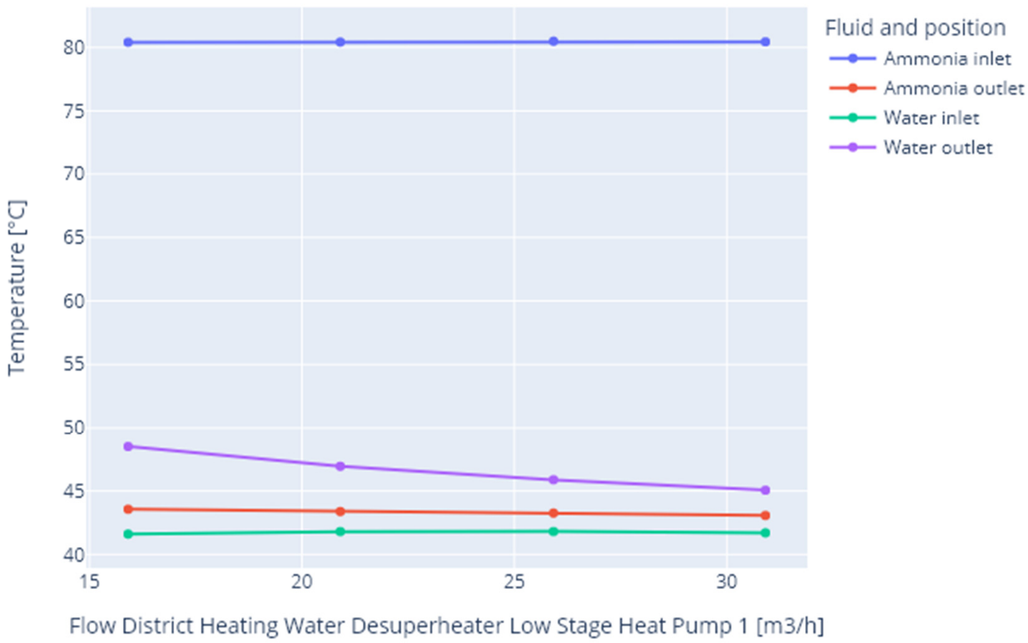

Figure 28: Average values for temperatures in the desuperheater of HP1-LS

It is interesting to notice that the temperature of the ammonia at the outlet of the heat exchanger doesn't decrease much with an increase in the water flow, due to a low pinch (temperature difference) on this end of the heat exchanger. It therefore seems that the flow in this heat exchanger should be lowered at least to 16 m3/h, and the flow gain used in another parallel heat exchanger (one of the 2 subcoolers for example).

Figure 29 presents the electrical power of the pump in relation to the flow through the desuperheater.

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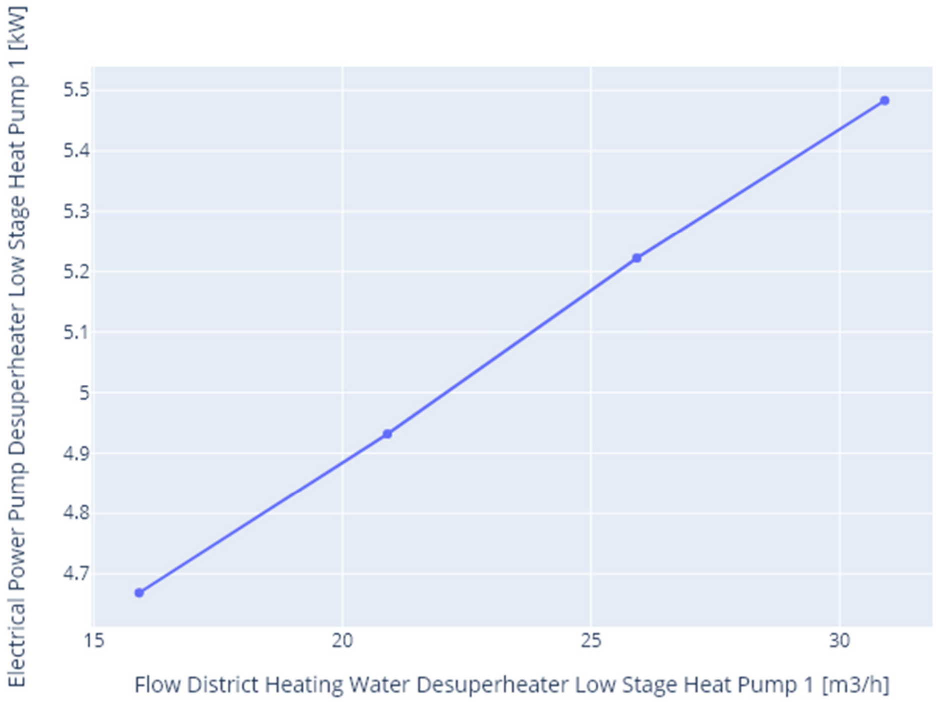

Figure 29: Evolution of the pump's power consumption with the flow through the desuperheater of HP1-LS

Reducing the flow in the heat exchanger generates a marginal gain of about 1 kWel in the consumption of the pump between the lowest and the highest value.

Conclusion

The studied parameter has a marginal influence on the performance of the installation, with a small change in the thermal power of the desuperheater.

The test results would point towards a reduction of the flow to 16 m3/h or even under: Indeed, this water flow could be redirected to another heat exchanger such one of the subcoolers (see the corresponding test 3.16), where it could potentially generate a gain of performance for the system.

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3.9: HP1-HS: Desuperheater DH flow sensitivity on COP

During this test the influence of the water flow through the desuperheater (DSH) of the high stage of HP1 on the COP is studied.

Conditions of the test

The following conditions could be observed during the test:

1. The evaporators are in **parallel**
2. The source is the **sewage water**
3. Variations of temperature on the source side are **low** (around 11°C to 11,5°C)
4. Variations of temperature on the DHN side:
   - Return temperature: **moderate** (44,5°C to 46,5°C)
   - Supply temperature: **very low** (almost constant 75°C)

Results

The gain/loss of COP for HP1 is plotted as a function of the flow in the DSH on Figure 30.

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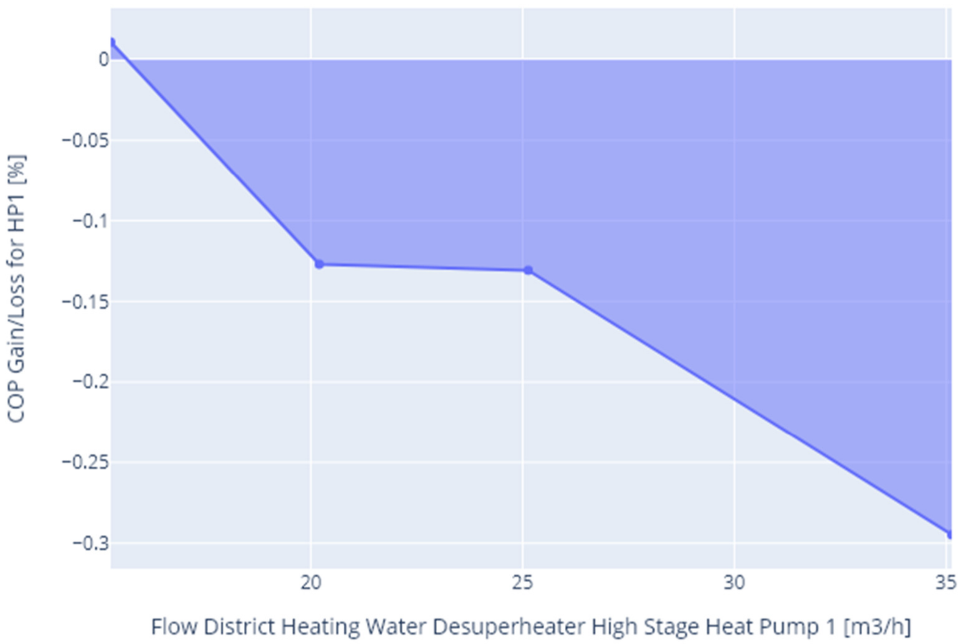

Figure 30: COP HP1 gain/loss between measured and expected

When looking at the graph, no real difference can be detected. The expected and measured COP are similar for all the values of the parameter.

The thermal power in the desuperheater of HP1-HS is mainly affected by 3 variables:

- the water temperature exiting the condenser of HP1 and entering the desuperheater (see Figure 31)
- the speed of the compressor (mass flow of ammonia) (see Figure 32)
- the flow of water through the heat exchanger (parameter studied in this test)

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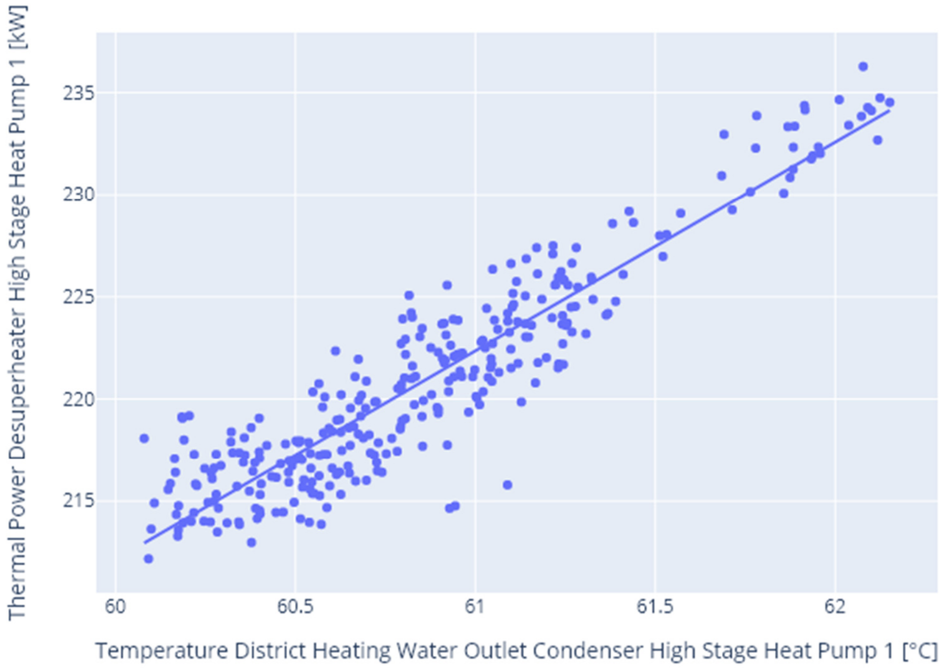

Figure 31: Thermal power of the desuperheater of HP1-HS: evolution with DH water temperature

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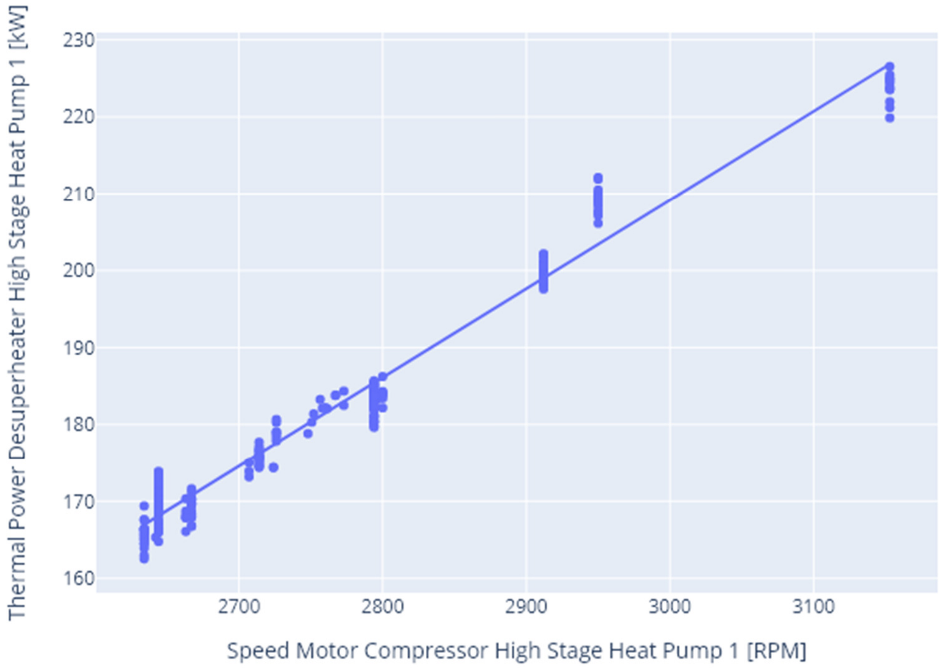

Figure 32: Thermal power of the desuperheater of HP1-HS: evolution with compressor speed (ammonia flow)

Figure 33 shows the thermal power of the desuperheater of HP1-HS.

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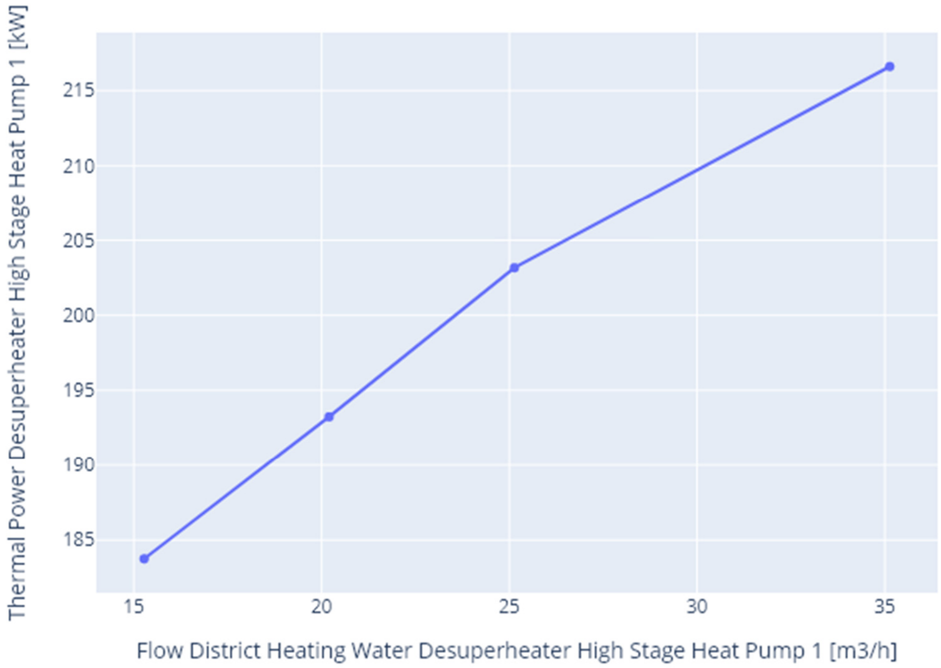

Figure 33: Thermal power of the desuperheater of HP1-HS: evolution with flow through the desuperheater of HP1

The thermal power is highest when the flow is the highest. Some variations of the return temperature during the test period were observed, complicating the analyzing, but the tendency is there and can still be seen on measurements with the same return temperature, as shown in Figure 34, where return temperatures have been filtered between 44.5 °C and 44.6 °C.

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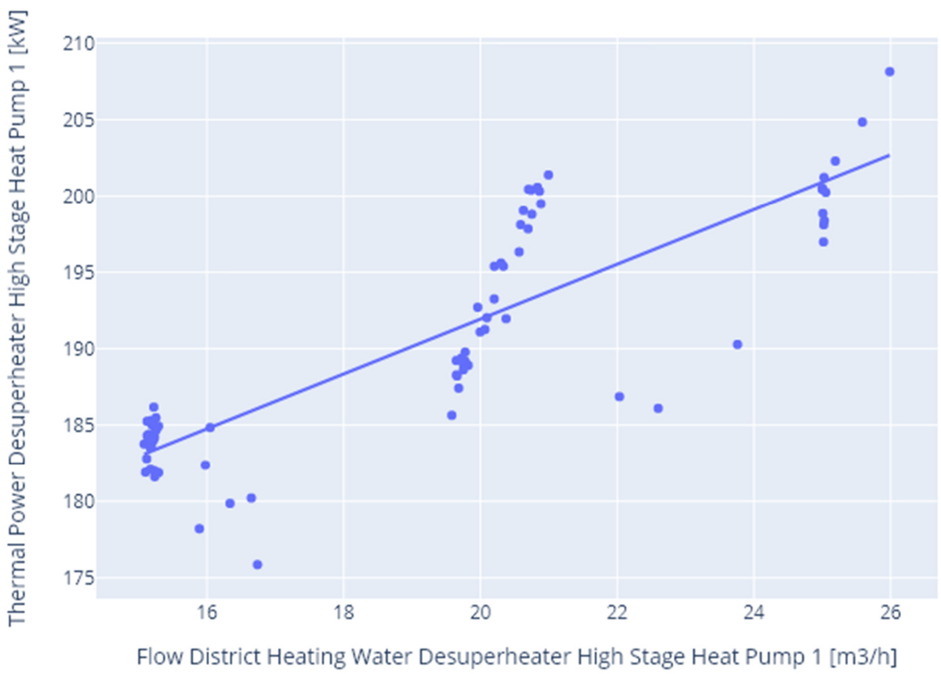

Figure 34: Thermal power of the desuperheater of HP1-HS: evolution with flow through the desuperheater of HP1 (constant inlet water temperature)

The temperatures for the ammonia and the water at different positions of the heat exchanger are presented in Figure 35.

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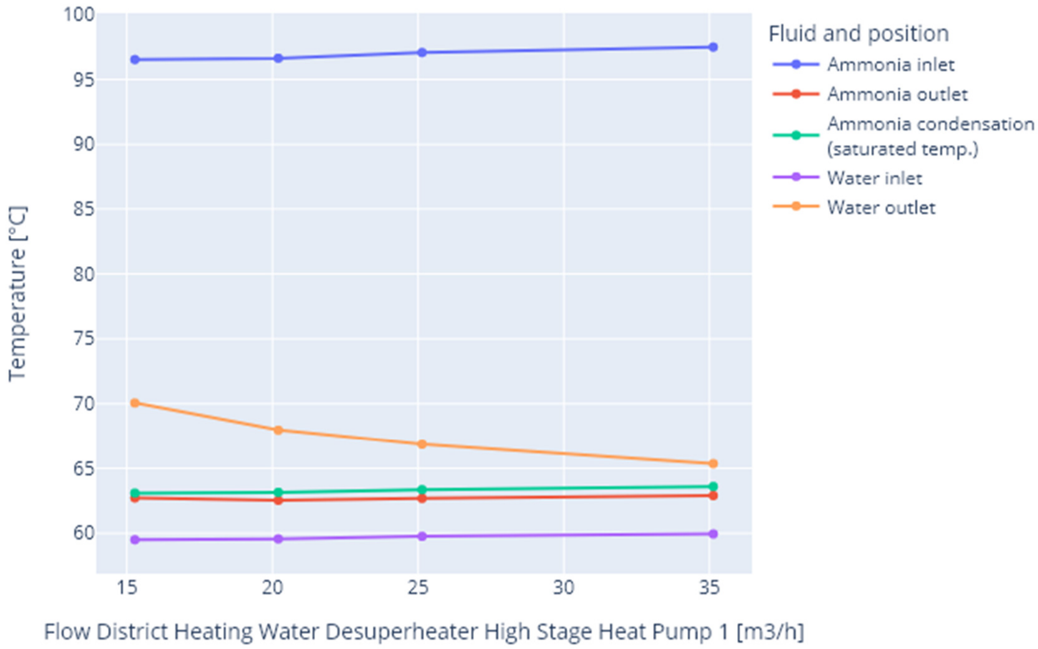

Figure 35: Average values for temperatures in the desuperheater of HP1-HS

It seems that the ammonia is starting to condense in the desuperheater, since the outlet temperature is rather constant and inferior to the calculated saturated temperature (measurement error in the temperature sensors can be the reason why values are not equal).

The water exiting the heat exchanger varies between 70 °C for the lowest flow down to 65 °C for the highest tested. This water is then mixed together with three other flows before entering the condenser of HP2:

- the outlet of the condenser of HP1 (one part of the flow bypasses the desuperheater)
- the outlet of the oil cooler of HP1-LS (in parallel with the desuperheater)
- the outlet of the oil cooler of HP2-LS (in parallel with the desuperheater)

These temperatures are shown in Figure 36.

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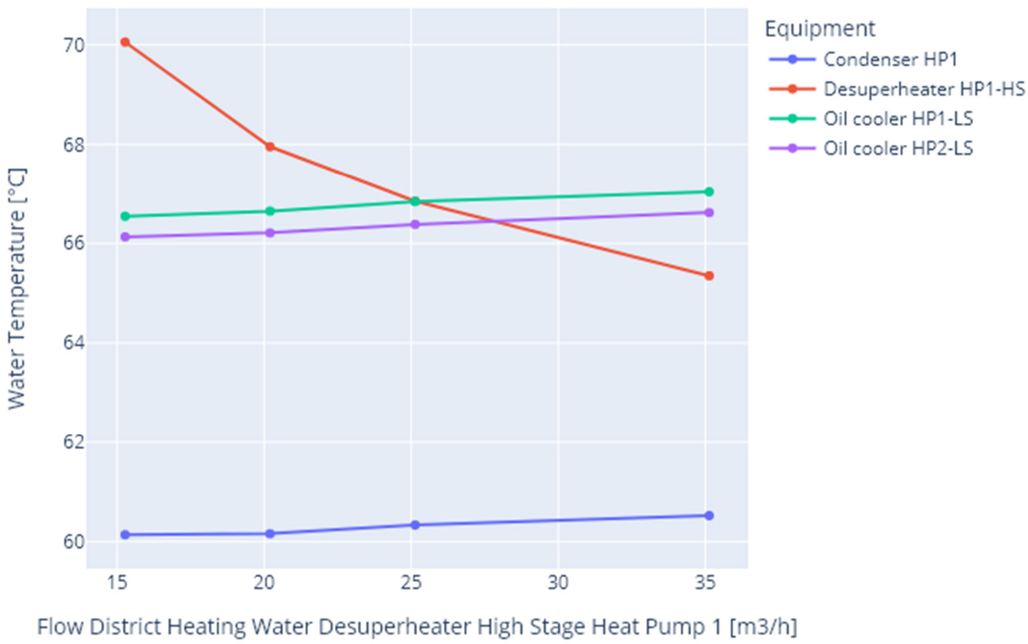

Figure 36: Average water temperatures in the condenser of HP2-HS

Thermodynamically speaking, the most efficient is to mix flows with a temperature that is the closest to each other, to reduce the irreversibilities. This approach would lead to have the highest flow tested through the desuperheater (35 m3/h), but at the same time to sensibly increase the flow in the oil coolers to also reduce the water temperature exiting them.

The electrical work required by the pump is plotted in Figure 37.

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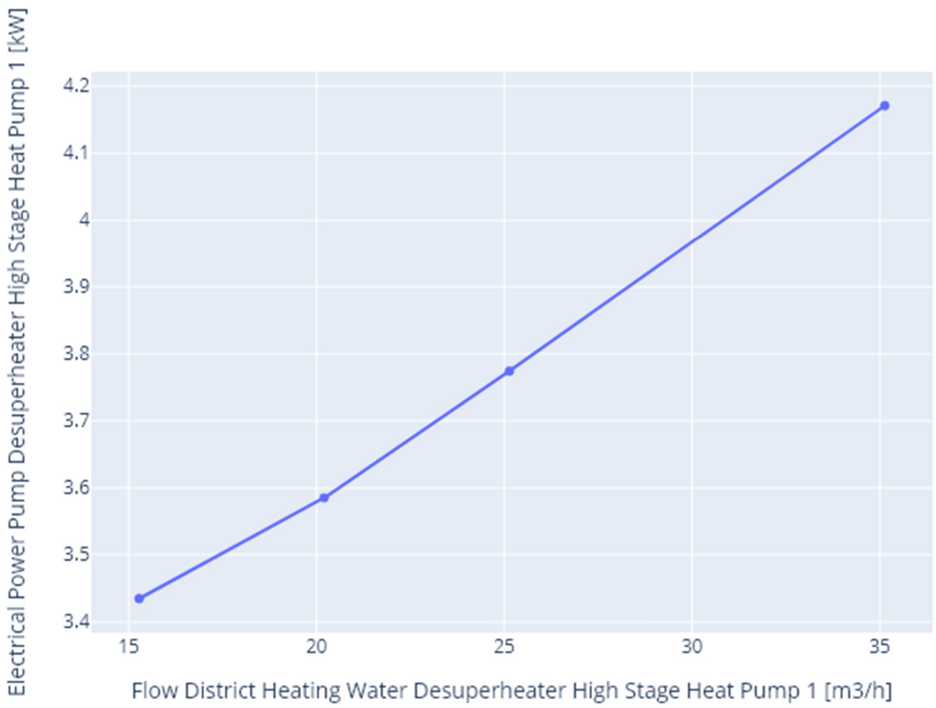

Figure 37: Evolution of the pump's power consumption with the flow through the desuperheater of HP1

The nominal power of the pump is 5.5kW, which means that there is still the margin to increase the flow through both the desuperheater and the oil coolers.

Conclusion

It is difficult to see the direct influence of the studied parameter on the performance of the installation.

The analysis of the heat exchanger thermal power and the water temperatures points towards having the highest possible flow through the desuperheater of HP1-HS (maximal value tested 35 m3/h).

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This increase should be performed together with a small increase in the water flow through the oil coolers of the low stage of both heat pumps, in order to have temperatures as close as possible.

Gregersensvej 2
DK-2630 Taastrup
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3.10: HP2-HS: Desuperheater DH flow sensitivity on COP

During this test the influence of the water flow through the desuperheater (DSH) of the high stage of HP2-HS on the COP is studied.

Conditions of the test

The following conditions could be observed during the test:

1. The evaporators are in **series**
2. The source is the **sewage water**
3. Variations of temperature on the source side are **very low** (11,2°C to 11,6°C)
4. Variations of temperature on the DHN side:
   - Return temperature: **low** (42,5°C to 44,5°C)
   - Supply temperature: **low** (74,5°C to 75,5°C)

Results

The gain/loss of COP for HP1 is plotted as a function of the flow in the DSH on Figure 38.

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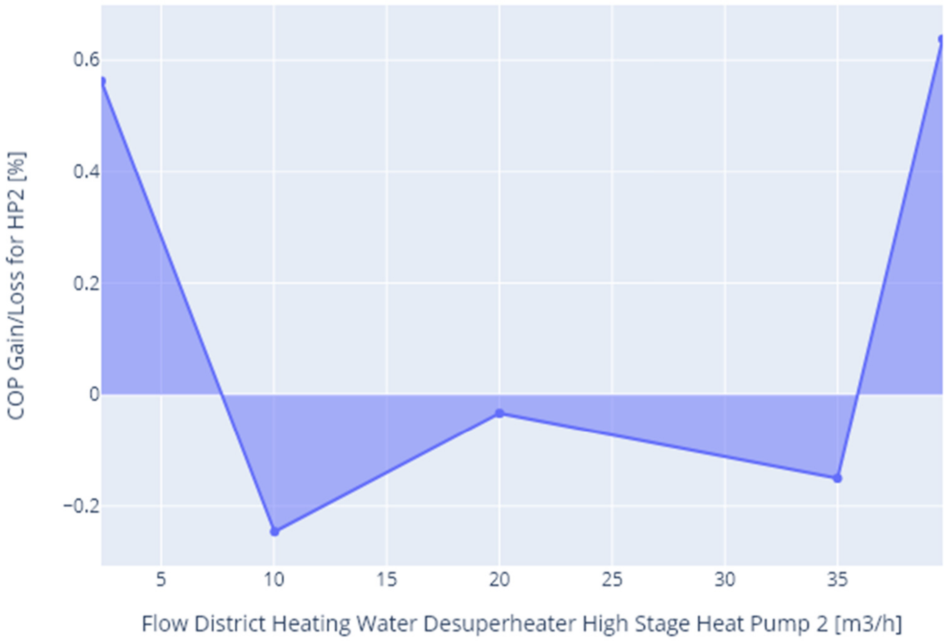

Figure 38: COP HP2 gain/loss between measured and expected

The COP of HP2 seems very sensibly impacted by the flow in the desuperheater: the variations are too small to conclude about a correlation similarly to the test 3.9 uniquely from the comparison with the regression.

The thermal power in the desuperheater of HP2-HS is mainly affected by 3 variables:

- the water temperature exiting the condenser of HP2 and entering the desuperheater (see Figure 39)
- the speed of the compressor (mass flow of ammonia) (see Figure 40)
- the flow of water through the heat exchanger (parameter studied in this test)

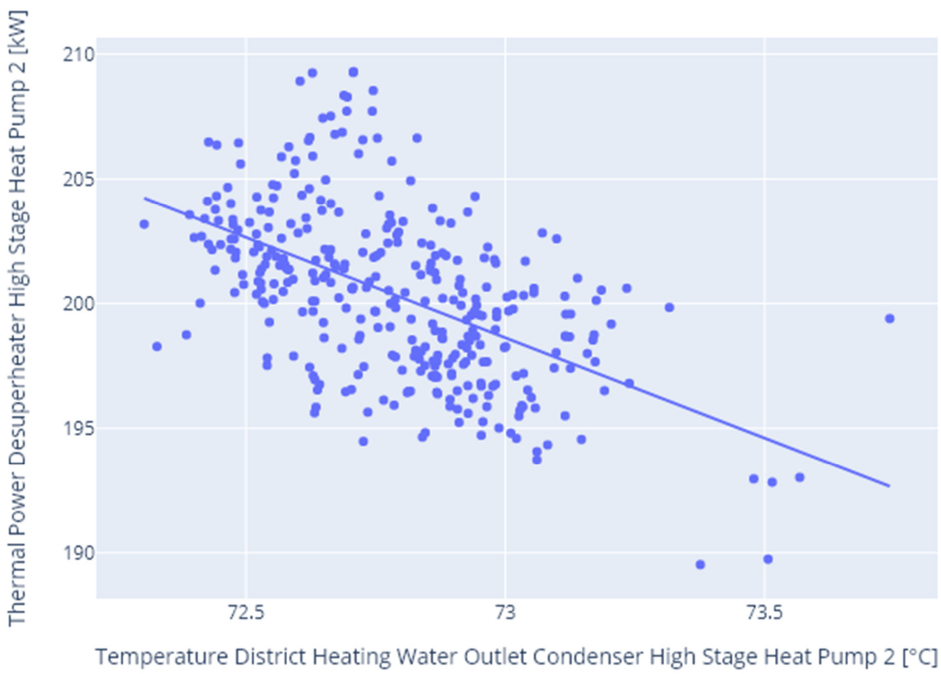

Figure 39: Thermal power of the desuperheater of HP2-HS: evolution with DH water temperature

The correlation is relatively hard to see, but the tendency is that the higher the temperature of the water in the desuperheater, the lower the thermal power of the desuperheater.

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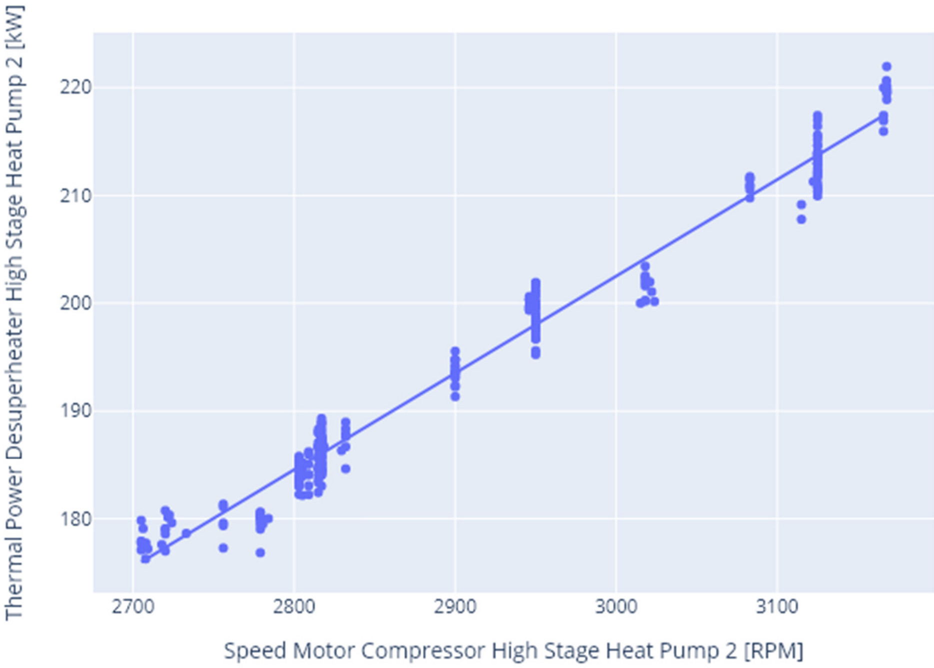

Figure 40: Thermal power of the desuperheater of HP2-HS: evolution with compressor speed (ammonia flow)

The thermal capacity of the desuperheater is plotted as a function of the flow in Figure 41.

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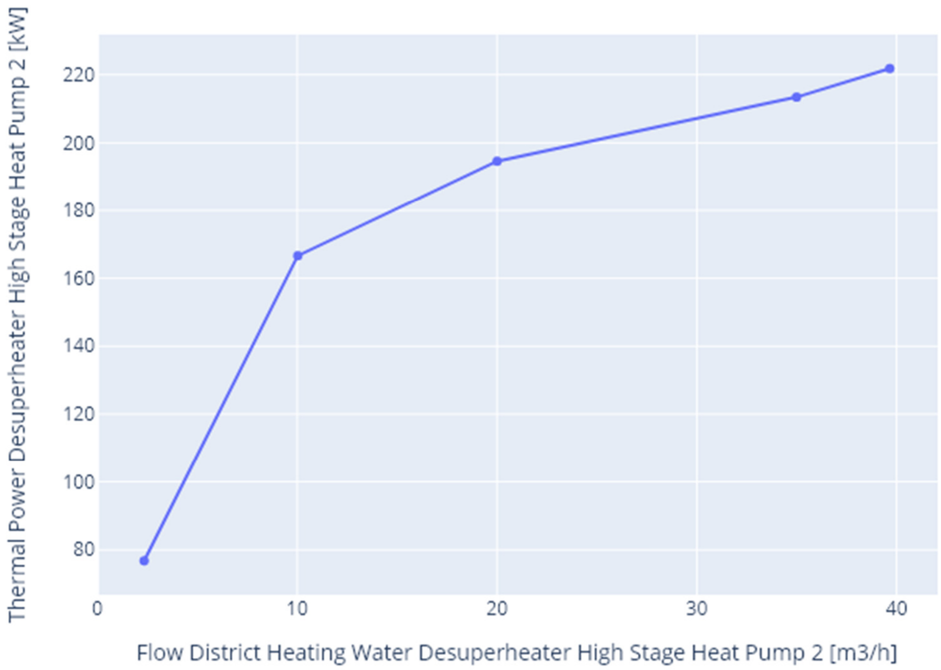

Figure 41: Thermal power of the desuperheater of HP2-HS: evolution with flow through the heat exchanger

The thermal power out of the desuperheater is higher for higher flows, and the difference is really important between the minimal and maximal flows. The effect of the flow is much stronger in this test by comparison with the similar test for HP1.

This can be explained by the fact that the regulation of the condenser temperature in HP2 is affected by the power of the desuperheater (temperature sensor for the control situated after the high stage oil coolers and desuperheater of HP2). The condensing temperature is therefore increased to ensure the respect of the setpoint (see Figure 42) and the water temperature entering the desuperheater is increased, reducing the heat capacity.

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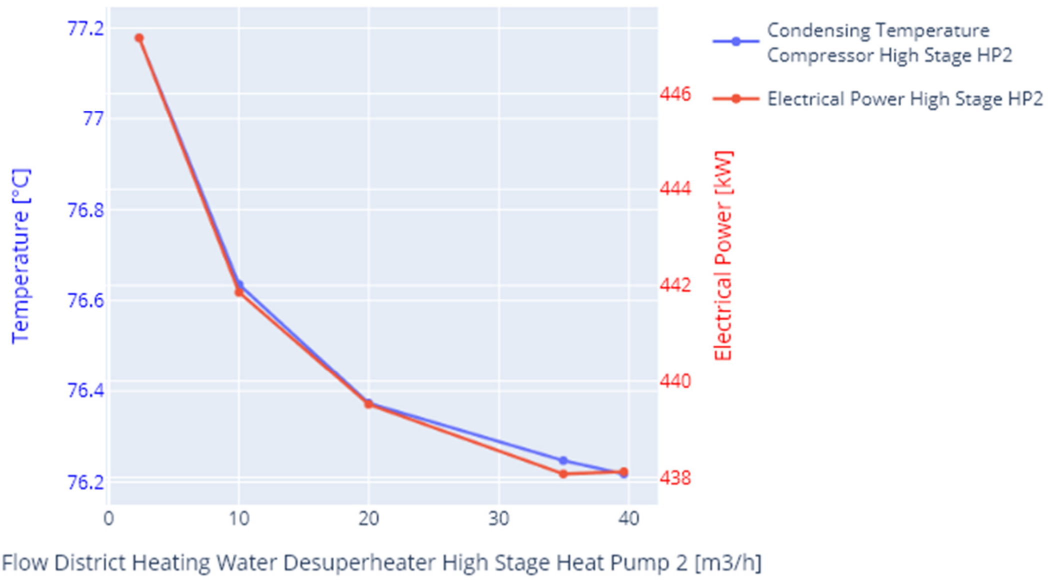

Figure 42: Dependance of the saturated temperature and power consumed on the flow in DSH HP2-HS

This increase in the condensing temperature of HP2-HS causes an increased electrical consumption, since the compressor needs to work at a higher pressure, reducing its performance.

It can be noted that the capacity of the compressor of HP2-HS was constant equal to 92% during the test, and is therefore not responsible for the extra consumption.

In the meantime the extra flow in the desuperheater causes an extra electrical work by the pump, as seen in Figure 43.

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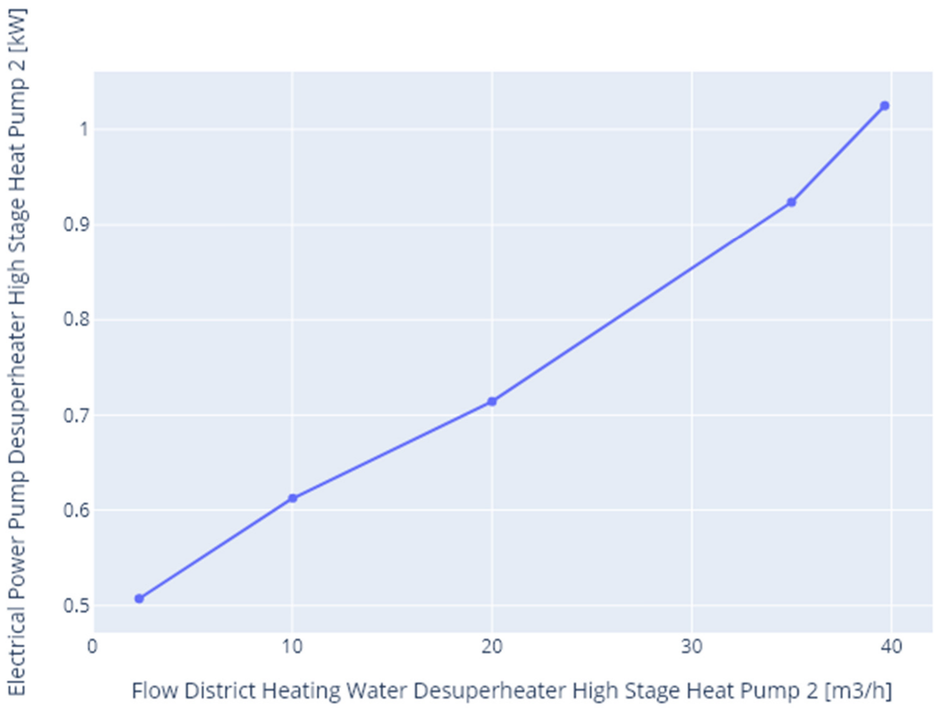

Figure 43: Evolution of the pump's power consumption with the flow through the desuperheater of HP2

The power consumed by the pump to circulate more flow is increased by less than 1 kW compared to a gain of about 9 kW on the compressor. The trade-off seems therefore in favor of the highest flow through the desuperheater. The maximal value achievable seems to be 40 m3/h, the pump power being 1.1 kW.

Conclusion

This test incentivizes to have the highest possible water flow circulating in the DSH of HP2, since it allows to lower the condensing temperature of the compressor, and thereby its electrical consumption.

The maximal achievable value seems to be 40 m3/h, for which the pump runs at 1 kW.

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Telefon 72 20 20 00

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3.14: HP2-HS: Oil cooler flow sensitivity on COP

During this test, the influence of the water flow through the oil cooler of the high stage of heat pump 2 (HP2) on the Coefficient of Performance (COP) is studied.

Conditions of the test

The following conditions could be observed during the test:

1. The evaporators are in **parallel**
2. The source is the **sewage water**
3. Variations of temperature on the source side are **low** (11.2°C to 11.6°C)
4. Variations of temperature on the DHN side:
   - Return temperature: **low** (44°C to 46°C)
   - Supply temperature: **low** (74.5°C to 75.5°C)

Results

The gain/loss of COP for HP2 is plotted as a function of the flow in the oil cooler on Figure 44.

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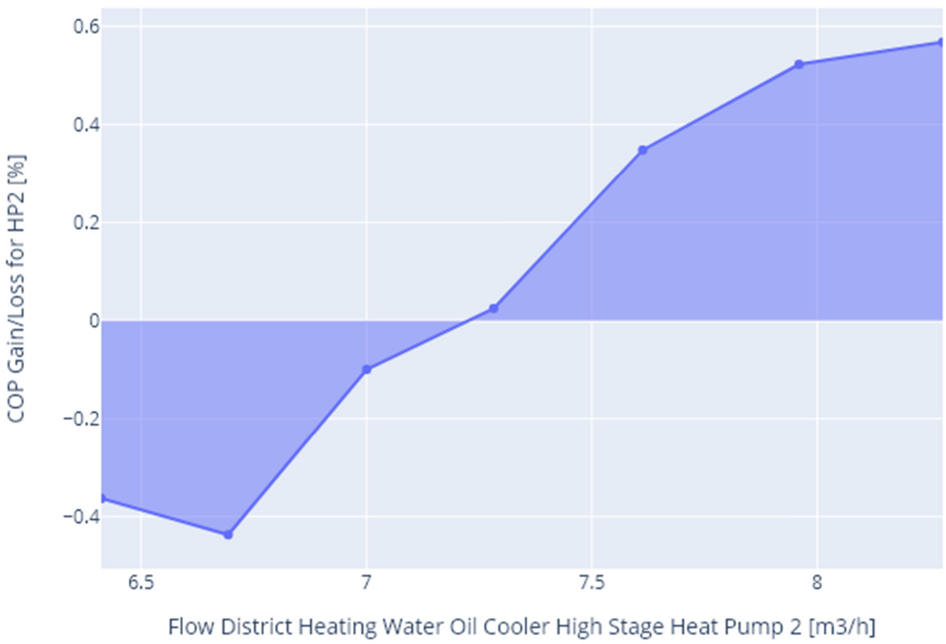

Figure 44: COP HP2 gain/loss between measured and expected

No significant difference between the different flow values can be detected. The precision of the regression of the COP doesn't allow to conclude on the effect of this parameter.

Figure 45 shows the calculated thermal power of the oil cooler of HP2-HS.

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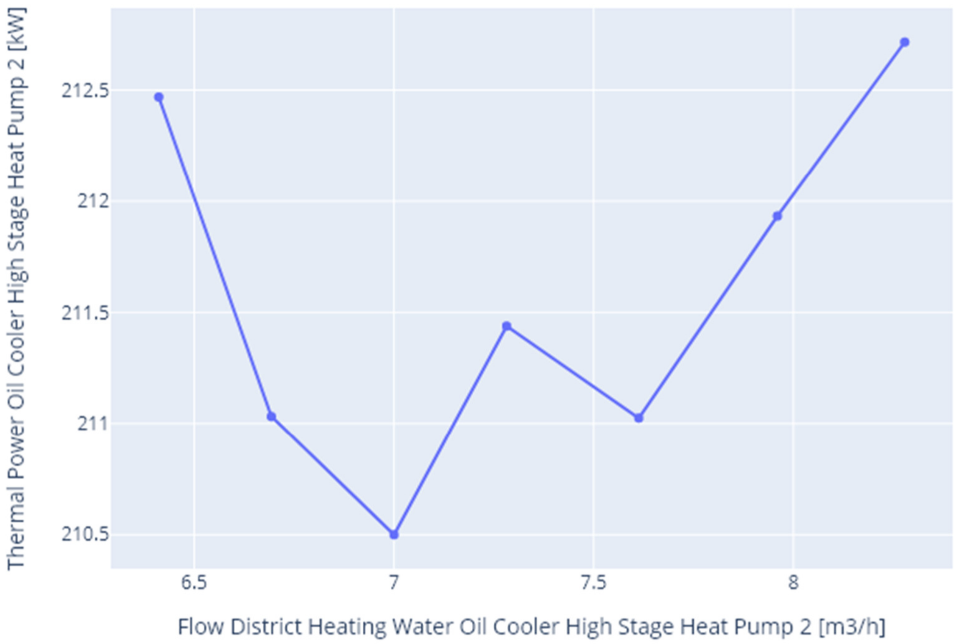

Figure 45: Thermal power of the oil cooler of HP2-HS: evolution with flow through the heat exchanger

No variation of the thermal power of the heat exchanger can be seen.

A higher flow in the oil cooler has however an influence on the oil temperature, as shown in Figure 46

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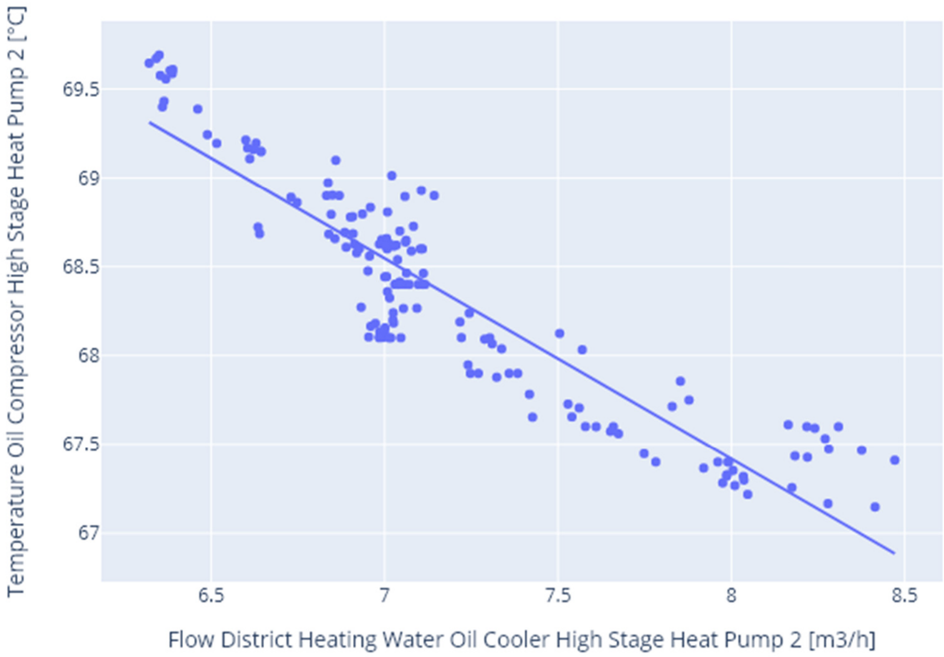

Figure 46: Oil temperature in the compressor of HP2-HS: evolution with flow through the heat exchanger

The difference between the minimal and maximal flows is little (about 2 K), but the extra electrical consumption for the pump is negligible (see Figure 47). The reduction of the oil temperature above a flow of 8 m3/h is not significant and this value therefore seems optimal.

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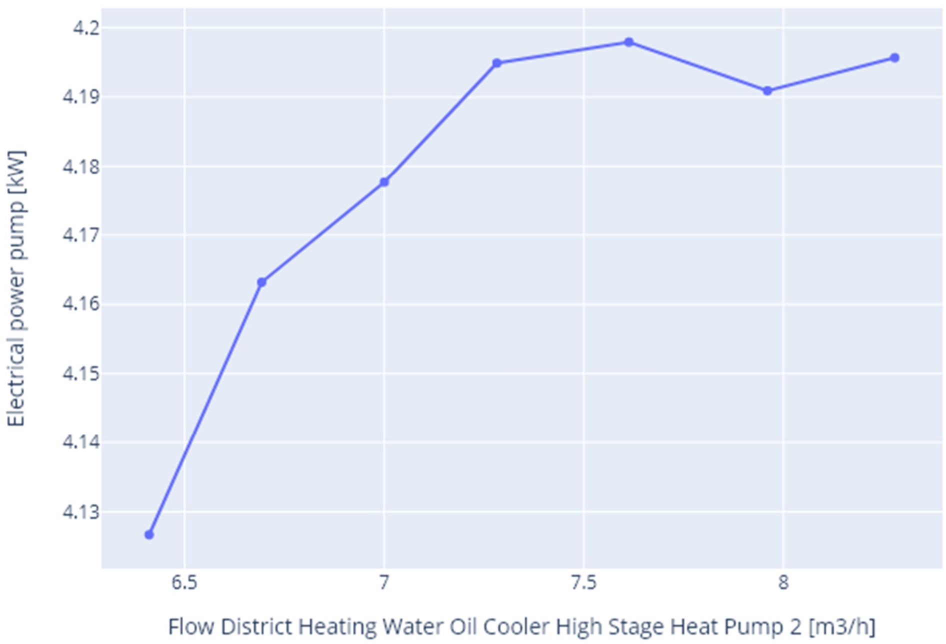

Figure 47: Evolution of the pump's power consumption with the flow through the oil cooler of HP1

Theoretically, the lower the oil temperature, the lower the compression work, since the oil is used to cool down the ammonia during compression. Practically it is really difficult to see this effect on the measurements with such small changes.

After this oil cooler the water is mixed together with three other flows:

- the water at the outlet of the oil cooler of HP1-HS
- the water at the outlet of the desuperheater of HP2-HS
- the water at the outlet of the condenser of HP2 (one part bypasses the heat exchangers)

These temperatures are shown in Figure 48.

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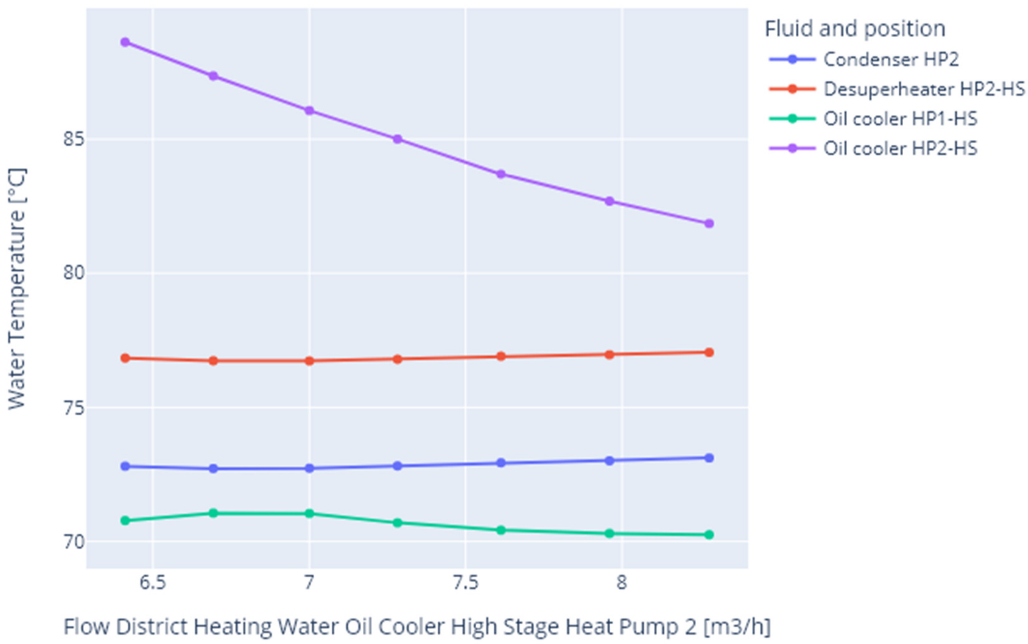

Figure 48: Average water temperatures mixing before the supply

The water temperature exiting the oil cooler of HP2-HS is much higher than for the other flows, especially when the flow through the heat exchanger is at the lowest (6 m3/h). Moreover, the water temperature out of the other oil cooler is lower than the supply value of 75 °C. It would therefore be relevant to decrease the water flow through the latter and redirect it through the oil cooler of HP2-HS (making sure that the oil temperature is not too high). These observations also encourage towards an increase of the flow through the oil cooler.

The changes in temperatures at the outlet of the other threee heat exchangers are caused by a change in the return temperature, as shown in Figure 49.

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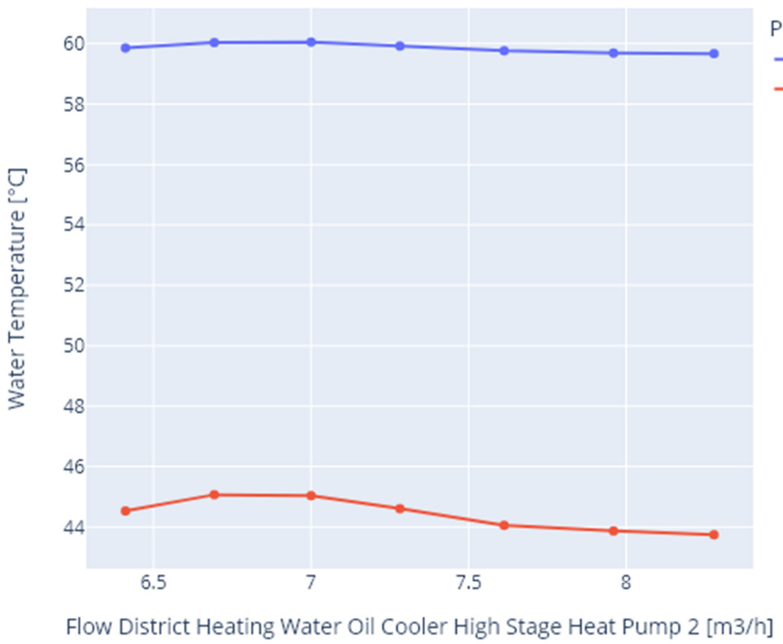

Figure 49: Average water temperatures mixing directly before the supply to the DHN

Conclusion

The studied parameter seems to have very little or no influence on the performance of the installation, at least under the conditions of the test, since the heat exchanger capacity remained the same.

The water flow in the oil cooler nevertheless influences the oil temperature, and the latter should be as low as possible to cool the compression down. The optimal flow value for the oil cooling seems to be 8 m3/h since above this value the oil remains about the same temperature.

When looking at the other heat exchangers in parallel with the oil cooler HP2-HS, the temperature levels of the water support a maximal increase in the water flow, and possibly removing flow from the oil cooler of HP1-HS, which was unfortunately not tested.

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DK-2630 Taastrup
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3.16: HP2-HS: Subcooler DH flow sensitivity on COP

During this test, the influence of the water flow through the subcooler (SC) of the high stage of heat pump 2 (HP2-HS) on the COP is studied.

Conditions of the test

The following conditions could be observed during the test:

1. The evaporators are in **parallel**
2. The source is the **sewage water**
3. Variations of temperature on the source side are **low** (11°C to 11.6°C)
4. Variations of temperature on the DHN side:
   - Return temperature: **moderate** (43.6°C to 46°C)
   - Supply temperature: **low** (74.5°C to 75.5°C)

Results

The gain/loss of COP for HP2 is plotted as a function of the flow in the DSH on Figure 50.

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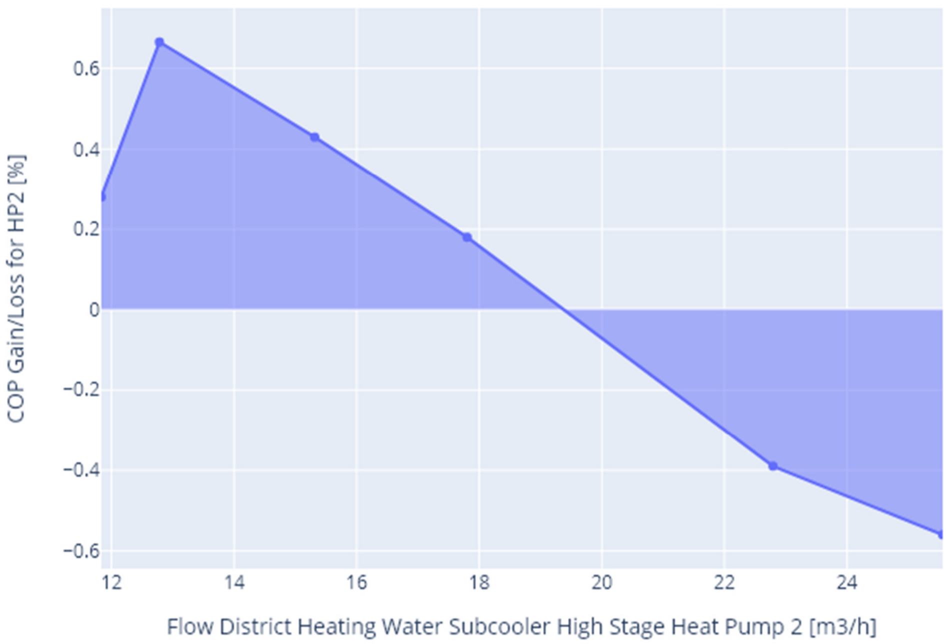

Figure 50: COP HP2 gain/loss between measured and expected

The variations of the COP observed are minimal, although a tendency seems to appear: the higher the flow in the subcooler, the higher the performance.

When looking at the thermal capacity of HP2, as shown in Figure 51, it seems to be higher with a higher flow, but dropping for the highest flow values.

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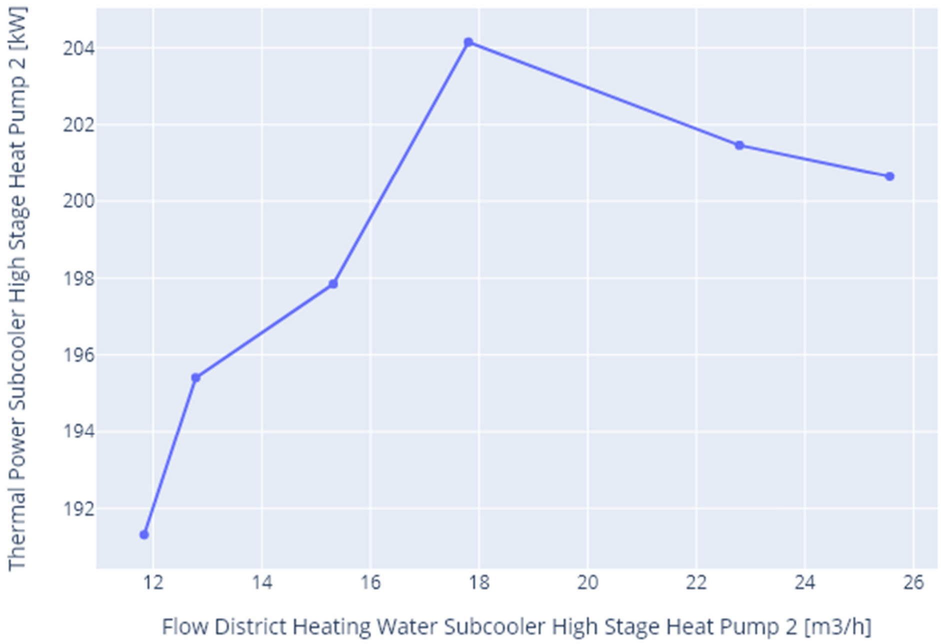

Figure 51: Thermal power of the subcooler of HP2-HS: evolution with flow through the heat exchanger

When looking closer at the data by taking into account the return temperature of the district heating water (see Figure 52), it can be seen that it has a big impact on the thermal capacity of the subcooler.

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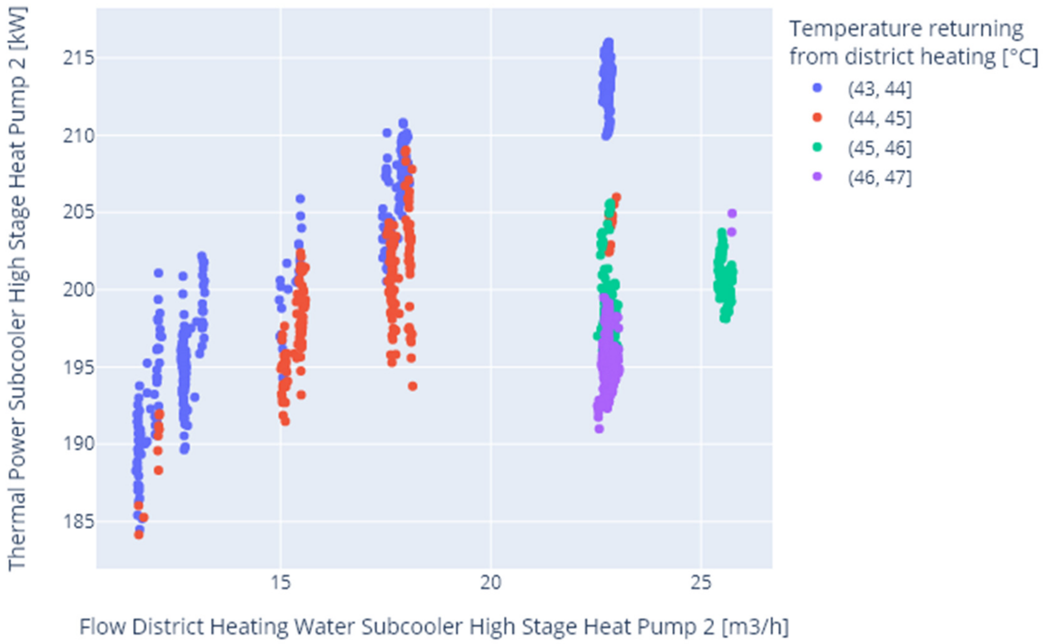

Figure 52: Influence of the return temperature on the thermal power of the subcooler of HP2-HS

The higher the return temperature, the lower the capacity of the subcooler. It makes sense since the subcooler is the first heat exchanger through which the water circulates. The return temperature affects therefore greatly the temperature difference between inlet and outlet of the subcooler, thus its capacity.

The temperatures for the ammonia and the water at different positions of the heat exchanger are presented in Figure 53, together with the condensation temperature in HP2.

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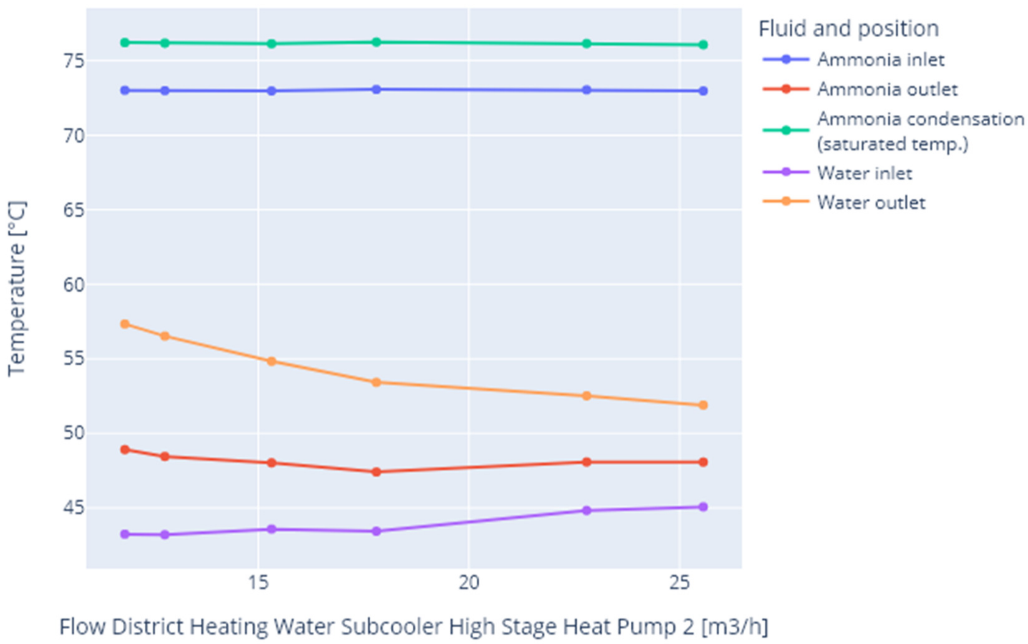

Figure 53: Average values for temperatures in the subcooler of HP2-HS

From these temperatures it can be seen that the limitation for the capacity increase heat exchanger would be at the water inlet and the ammonia outlet, since the pinch is relatively low for the highest flows (3 K).

The extra work required by the pump is negligible compared to the thermal capacity gain, as seen in Figure 54: +0.7 kWel compared to up to +30 kWth in the subcooler.

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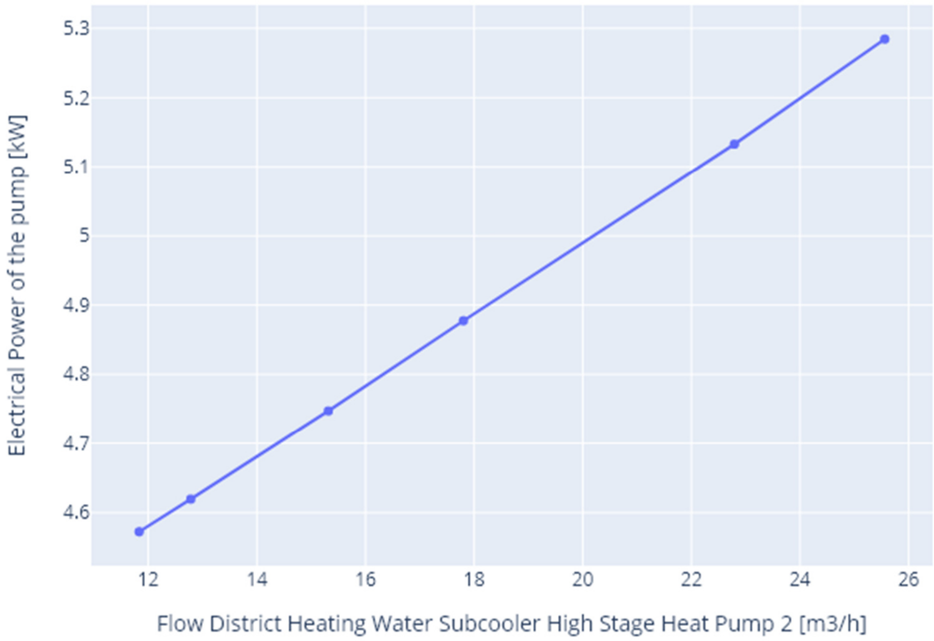

Figure 54: Variation of the pump's consumption with the flow in the subcooler of HP2

The nominal power of the pump being 7.5 kW, it would be interesting to test higher values of flow in the subcooler to potentially find an optimal flow. Indeed, the original value (in normal operation) was about 23 m3/h, while the highest value for the test was 25 m3/h.

The conclusion of the test with the desuperheater of HP1-LS was that the flow in the desuperheater could be reduced, this flow could therefore be used in the subcooler of HP2.

Conclusion

The increased capacity on the subcooler for higher flows incentivizes for a high value of the flow in the subcooler. 25 m3/h was the maximal value tested, but the nominal power of the pump is higher than the maximal power reached during this test. Additionally some of the water flow currently used for the desuperheater of HP1-LS could be used in the subcooler instead, opening the possibilities for even higher flows.

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If such an increase in made, it would be interesting to look at the pinch of the heat exchanger, meaning the difference between inlet of the water and the outlet of the ammonia.

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3.17: HP1: Liquid level in liquid separator for evaporator

The tested parameter is the liquid level in the liquid separator of HP1. The idea behind this test is that a lower level would cause a lower circulation ratio in the evaporator and therefore less pressure drop. The compressor of the low stage would therefore be able to work at a higher pressure, increasing the COP.

Results

The gain/loss of COP for HP1 is plotted as a function of the liquide level in the separator on Figure 55.

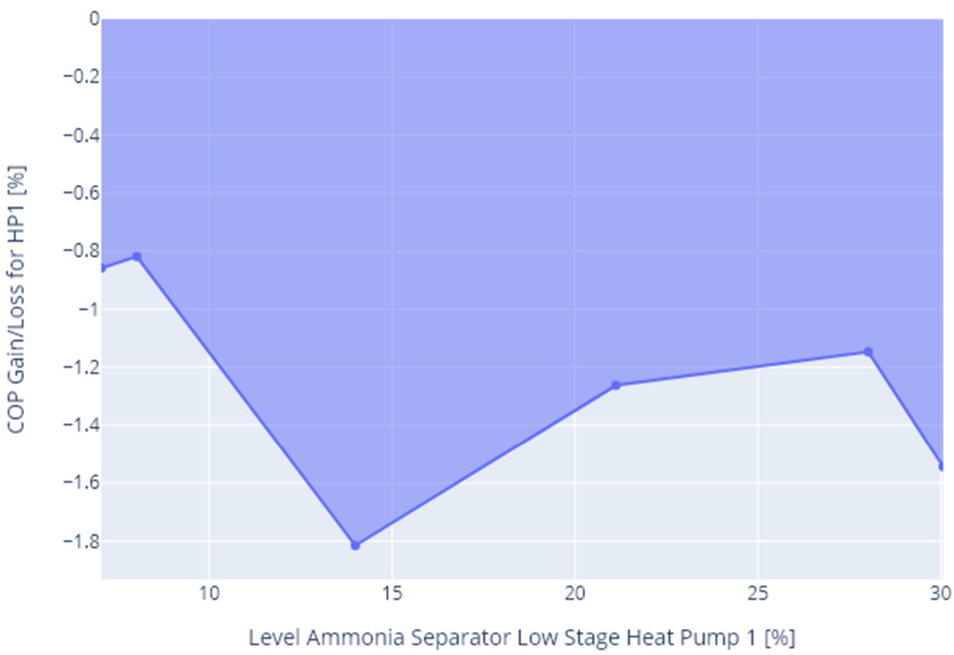

Figure 55: COP HP1 gain/loss between measured and expected

No clear tendency seems to appear and the variations in the COP are too little to be able to conclude about the benefits of the parameter directly on the performance.

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One can look at the variations in the pressure in the evaporator, which was the idea behind this test.

To do so, the temperature difference between the water entering the evaporator and the evaporating temperature (directly derived from the pressure) is calculated. This value is plotted against the level in the separator on Figure 56. The thermal load of the district heating, which can also affect the suction pressure on the low stage, was almost constant during the test (4.0 MW to 4.1 MW), so it is considered that its effect is negligible.

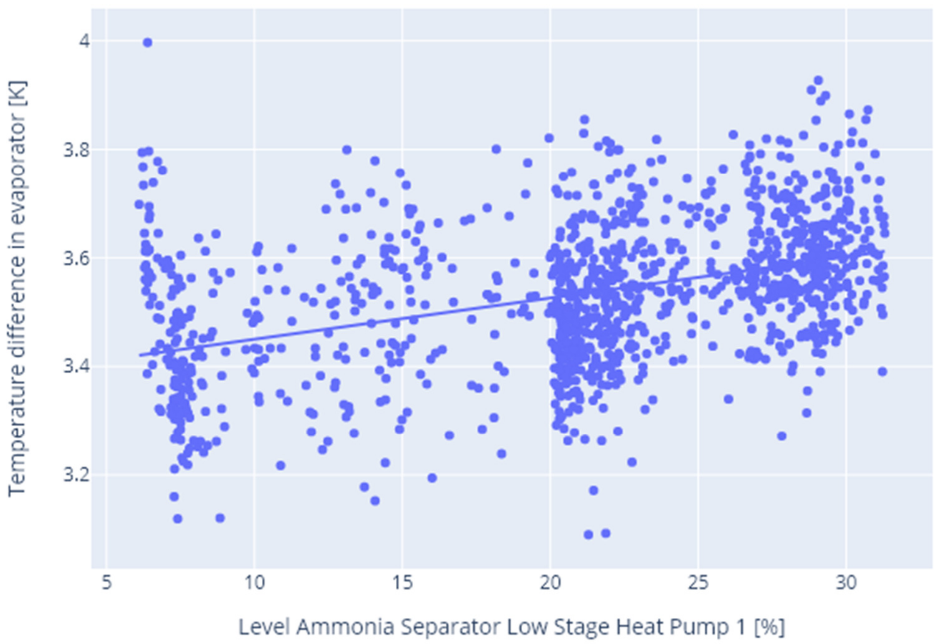

Figure 56: Influence of the liquid level in the separator on the temperature diffence in the evaporator

Once again it is difficult to conclude with these results. The variations are very low and could therefore be related to different changes in the system than the studied parameter. Nonetheless the light tendency that can be seen is that the higher the level, the higher the temperature difference, thus the lower the compressor suction pressure, meaning a lower performance.

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Conclusion

It seems that a lower liquid level in the separator could sensibly increase the evaporating temperature and therefore provide a better COP.

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3.18: HP2: Liquid level in liquid separator for evaporator

The same test as test 3.17 is conducted for HP2.

Results

The gain/loss of COP for HP2 is plotted as a function of the liquide level in the separator on Figure 57.

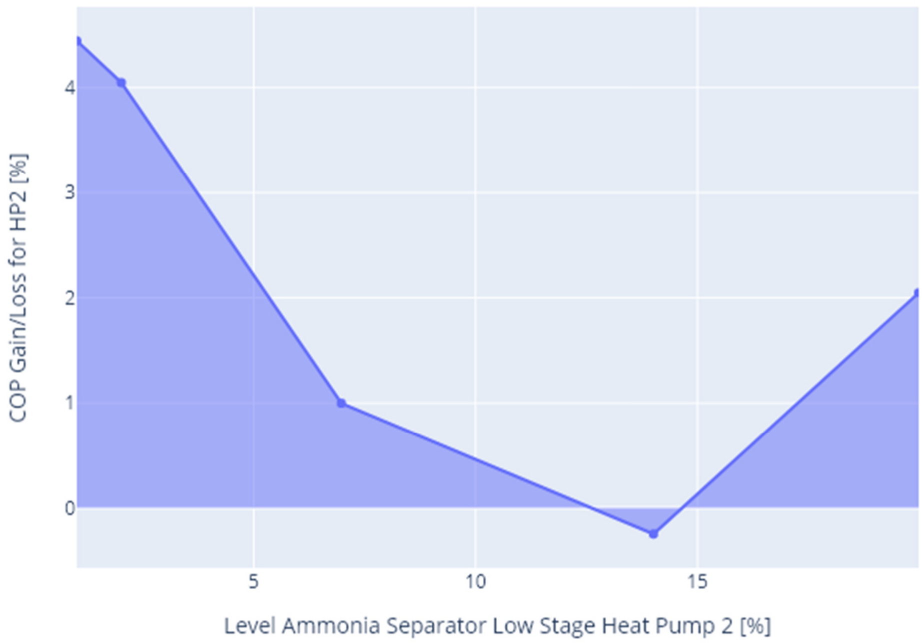

Figure 57: COP HP2 gain/loss between measured and expected

No clear tendency seems to appear and the variations in the COP are too little to be able to conclude about the benefits of the parameter directly on the performance.

The temperature difference in the evaporator is shown as a function of the level in the separator on Figure 58.

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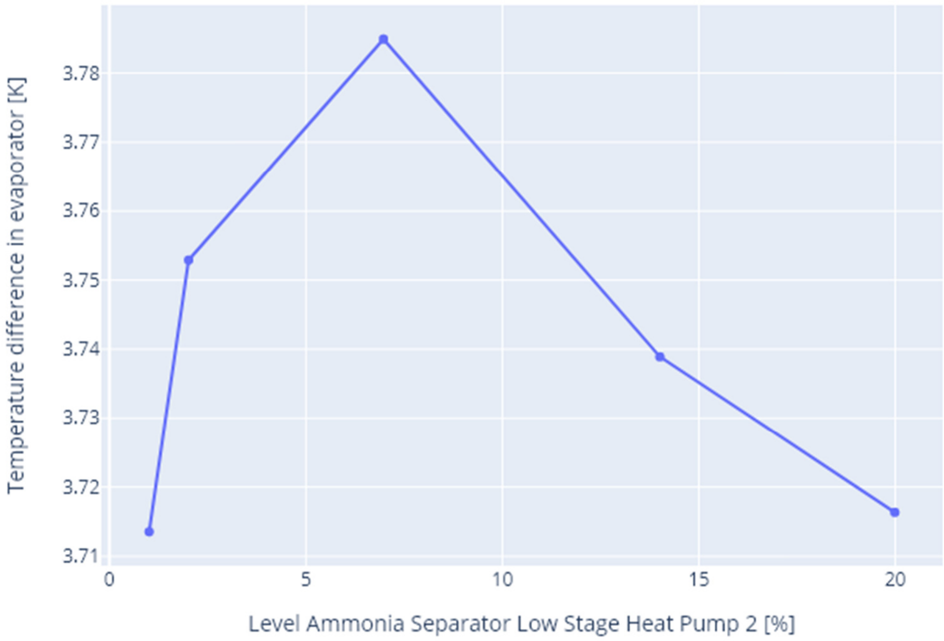

Figure 58: Influence of the liquid level in the separator on the temperature diffence in the evaporator

The variations are too small to be able to find a correlation between the effect of the liquid level and the evaporation pressure.

Conclusion

The results of the test unfortunately don't allow to conclude on the effect of the parameter on the COP.

However, following the results for the test on HP1, the liquid level should be as little as possible in the liquid separator.

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General conclusion and considerations after the tests

Although a clear performance tendency was difficult (or impossible) to detect in the test results due to constant changes in the running conditions, some conclusions can be drawn from the short term test program.

All the parameter changes of a test should be run during a relatively stable period, otherwise the changes of performance caused by the parameter cannot be seen. To do so, the return temperature of the DHN could for example be maintained at the same temperature during the test. Another possibility would be to run the tests on several days, with the change of parameter lasting longer, to make sure to find comparable conditions for all the parameter changes. The problem with this strategy lays in the fact that it's much harder to surveil the installation and its response to the changes during several days instead of a few hours, especially when the plant is running outside regular working hours.

The idea of a regression for the electrical consumption of the compressor is in theory a good way to estimate gains or losses during the tests when changes occur. However the difficulty to get a precise model, even with measurements data, compromises its use. Indeed the imprecision of the model being bigger than the changes potentially observed during the tests, it made the analysis with the regression complicated. Some tracks could be followed to improve the model, such as the use of more "training" data (limited in the current study due to low running time of the plant) and more homogeneous data (here very long period with changes of equipment and setpoints in the system).

Parameters to change according to the test results

The thorough analysis of the data allowed to better understand the running conditions of the heat pump and to see potential (minor) improvements detailed in the following table.

Test	Max gain for the system	Recommendation
3.3-1	Difficult to conclude	Decrease the intercooler temperature to the theoretical optimum for HP1: 32 °C
3.3-2	Difficult to conclude	Decrease the intercooler temperature to the theoretical optimum for HP1: 32 °C
3.4	Difficult to conclude	Increase the setpoint at 53% to balance more the heat production of each heat pump
3.5	Flow around 600 m3/h	Change the setpoint at the evaporators to about 4 K

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Test	Max gain for the system	Recommendation
3.7	Lower flow through the desuperheater HP1-LS	Decrease the flow through desuperheater of HP1-HS to 16 m3/h or lower
3.9	Highest flow through desuperheater HP1-HS	Increase the flow through desuperheater of HP1-HS to 35 m3/h or above
3.10	Highest flow through desuperheater HP2-HS	Increase the flow through desuperheater of HP2-HS to the maximum possible (40m3/h)
3.14	Highest flow through oil cooler of HP2-HS	Increase the flow through oil cooler of HP2-HS
3.16	Highest flow through subcooler HP2	Increase the flow through subcooler HP2-HS to the maximum possible (25 m3/h or above)
3.17	Difficult to conclude	Keep the liquid level in separator low, around 5% for example
3.18	Difficult to conclude	Keep the liquid level in separator low, around 5% for example

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Telefon 72 20 20 00

info@teknologisk.dk
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Appendix

Some graphs that were not presented in the body of this documents were plotted here since they could provide extra insights on the installation.

Slide capacity on low stage compressors

During the test 3.3 on intercooler temperature, the slide capacity of the compressor changes with the change in temperatures

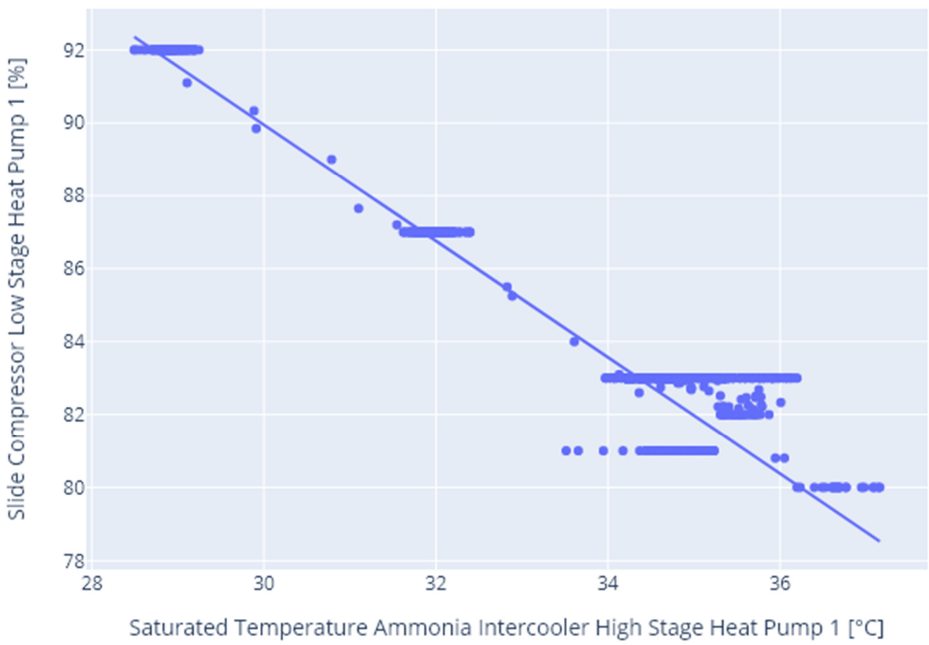

Figure 59: Slide capacity as a function of the intercooler temperature for LS-HP1

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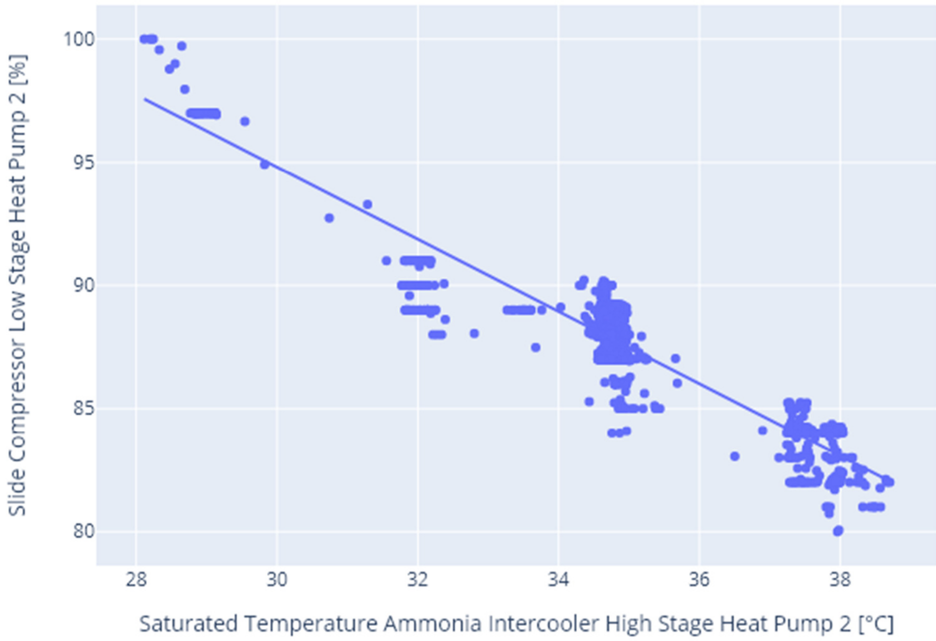

Figure 60: Slide capacity as a function of the intercooler temperature for LS-HP2

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COP for each heat pump stage as function of pressure ratio

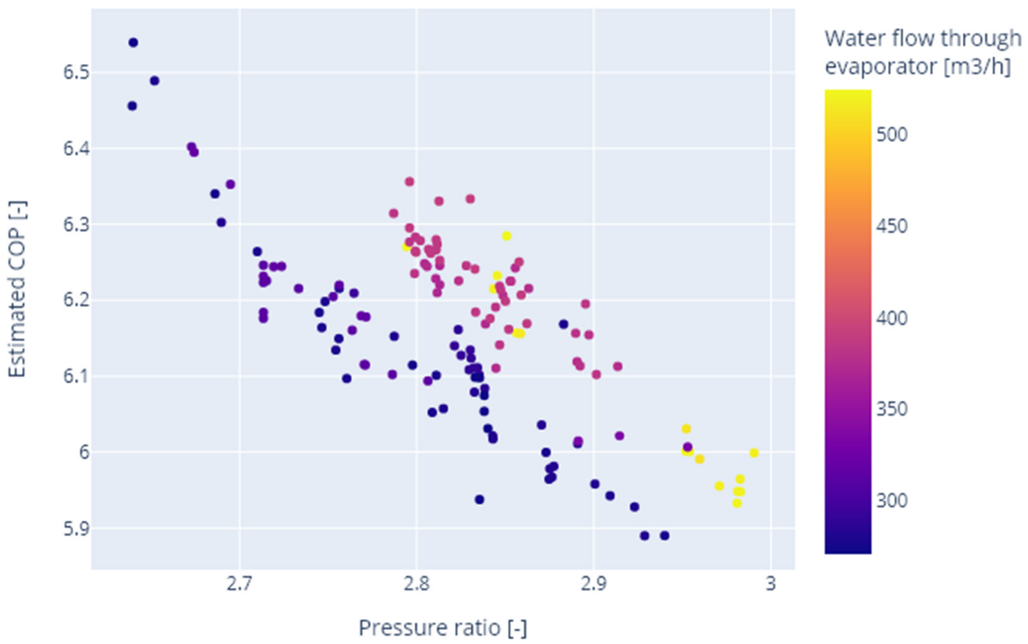

Figure 61: Coefficient of Performance as a function of the pressure ratio for the compressor of HP1-LS

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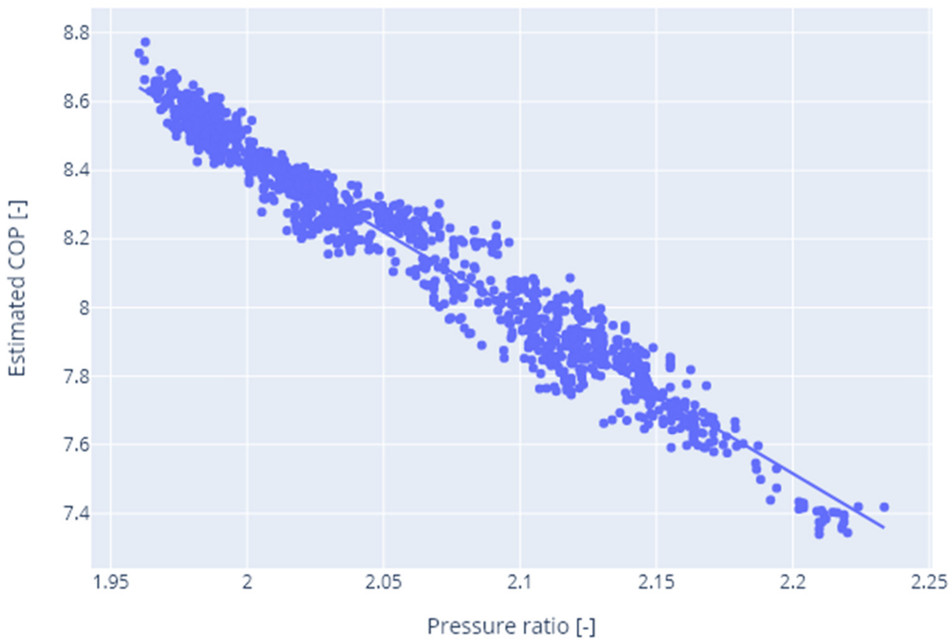

Figure 62: Coefficient of Performance as a function of the pressure ratio for the compressor of HP1-HS

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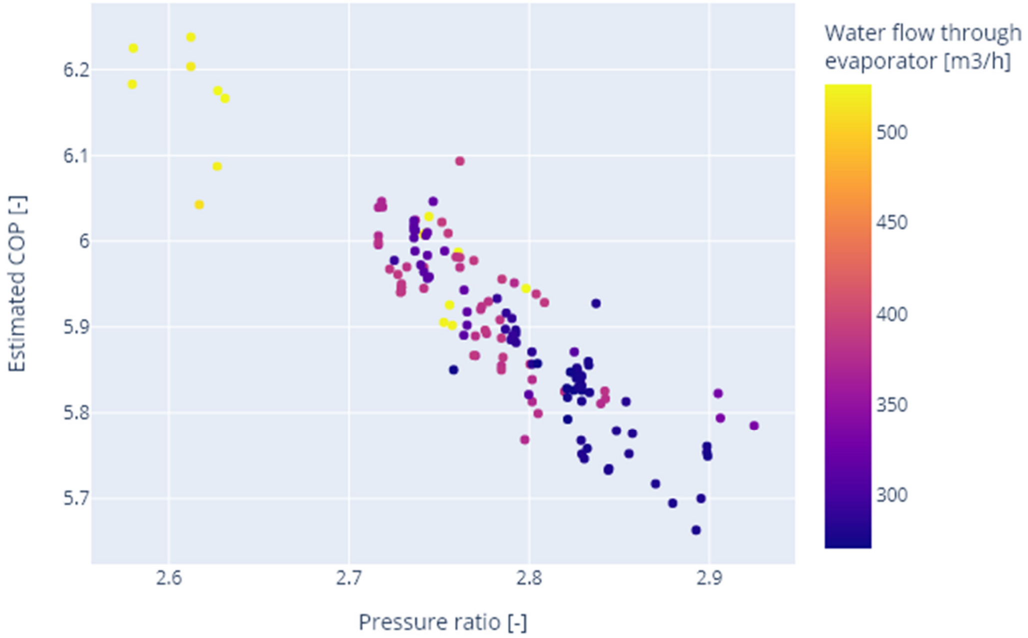

Figure 63: Coefficient of Performance as a function of the pressure ratio for the compressor of HP2-LS

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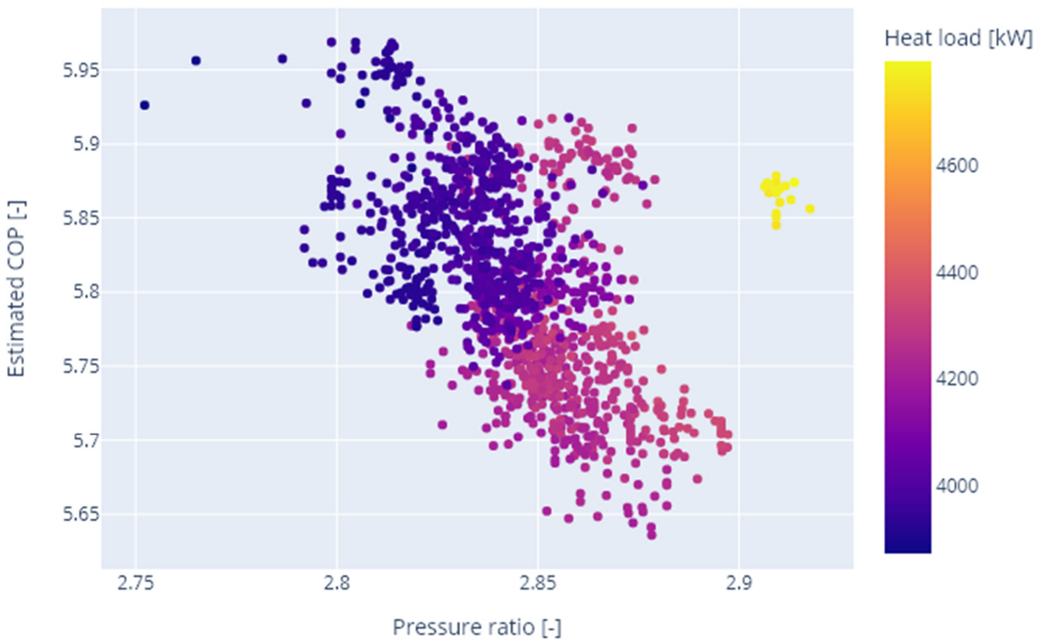

Figure 64: Coefficient of Performance as a function of the pressure ratio for the compressor of HP2-HS

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Low stage work as function of evaporation temperature

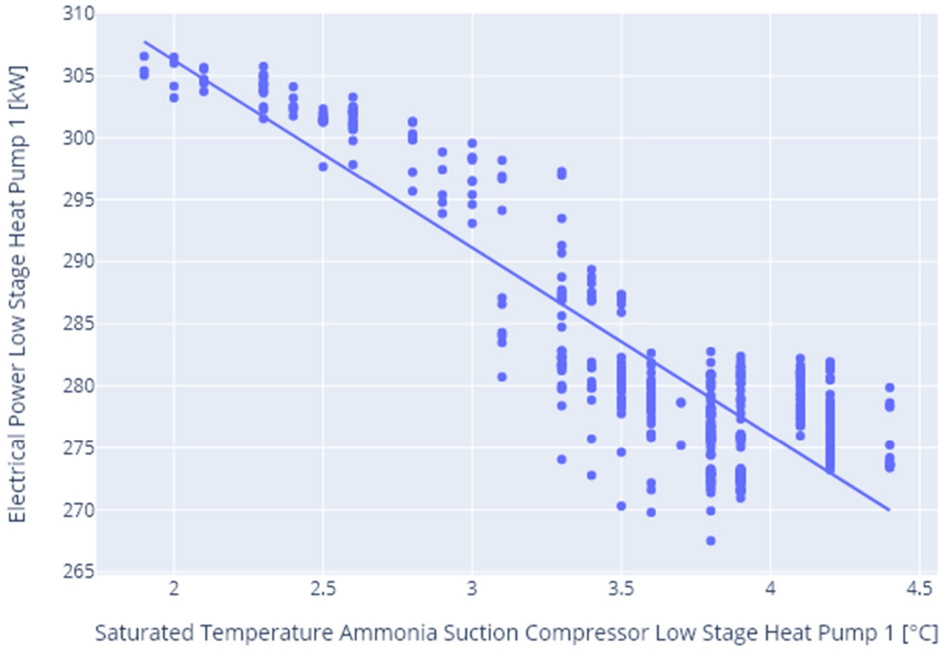

Figure 65: Compressor work as a function of the evaporation temperature for HP1-LS

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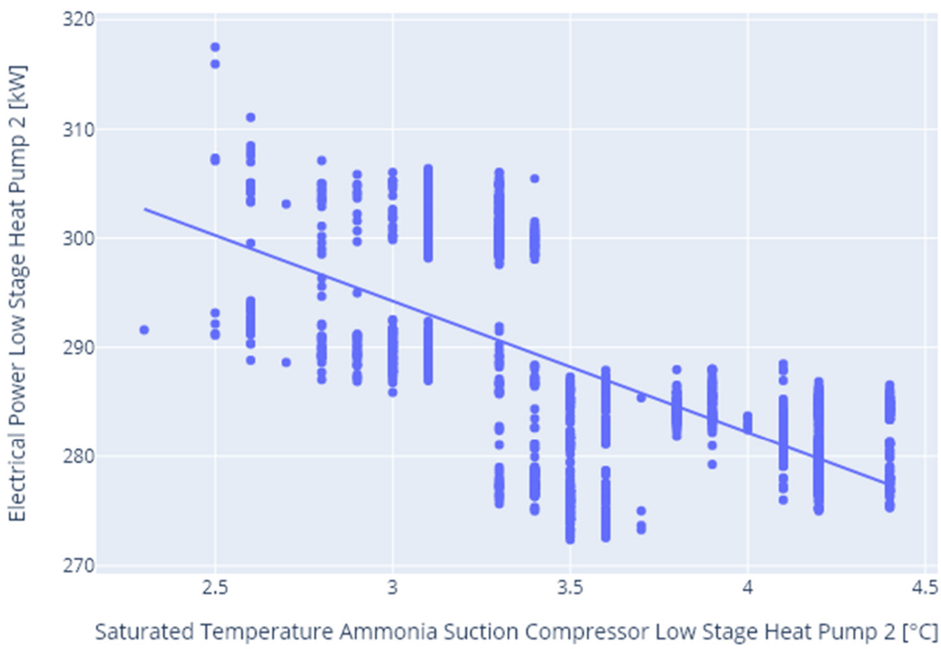

Figure 66: Compressor work as a function of the evaporation temperature for HP2-LS