Evaluation of a variable suction gas heat exchanger in a liquid chiller system using carbon dioxide as the refrigerant

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Abstract

In a refrigerated seawater (RSW) system using carbon dioxide (CO_2) as the refrigerant, a variable bypass valve was installed in front of a suction gas heat exchanger (SGHX). A simulation tool was developed and utilized to optimize the systems transcritical performance (COP) with respect to the gas cooler pressure and choke valve inlet temperature for cooling and combined cooling and water heating. The simulations indicate that the RSW system performance can be increased compared to running a system with a traditional non-variable suction gas heat exchanger, but only when the cooling water temperatures are high (above ~25°C).

Experimental testing of the cooling performance with transcritical gas cooler pressure show little or no improvement at cooling water temperature levels below 25°C compared to previous experiments. The optimum setting turned out to be maximum suction gas heat exchange (no bypass). Previous experiments also show that for low cooling water temperatures (20°C and below), the optimum gas cooler pressure is subcritical.

The introduction of a variable suction gas heat exchanger made it possible to also use the system for water heating. The optimum and only possible SGHX setting was to bypass the SGHX altogether, as heat exchange would cause overheating at the compressor outlet. In other words, if the current system is to be used for water heating, the possibility to fully bypass the SGHX was essential.

The concept of a variable SGHX appears not to beneficial in a water cooled RSW system, but the simulations indicate that it has potential in areas where air is used as the cooling medium, for instance commercial or mobile refrigeration. For cooling purposes, experiments and simulations show that a non-variable suction gas heat exchanger can reach near-optimum conditions in the RSW system when low cooling water temperatures are available. It is strongly recommended that a system to be used for simultaneous cooling and heating should have an improved design compared to the current setup, as this mode of operation shows low cooling capacity and poor energy efficiency.

1 Introduction

Table 1-1: Nomenclature

Symbol	Description	Unit
D	Diameter	m
D_h	Hydraulic diameter	m
h	Enthalpy, specific	$\frac{kJ}{kg}$
P	Pressure	Pa (absolute)
R	Thermal resistance	$\frac{K}{W}$
s	Entropy, specific	$\frac{kJ}{kgK}$
T	Temperature	$°C$
U	Velocity	$\frac{m}{s}$
v	Specific volume	$\frac{m^3}{kg}$
ε_{SGHX}	Suction gas heat exchanger efficiency	-
η_{is}	Isentropic efficiency	-
λ	Volumetric efficiency	-
μ	Dynamic viscosity	$Pa \cdot s$
υ	Kinematic viscosity	$\frac{m^2}{s}$
ρ	Density	$\frac{kg}{m^3}$

Leakage of refrigerants with high greenhouse warming potential (GWP) poses a threat to our environment, and one of the possible replacements is carbon dioxide (CO_2), which does not have any negative impact when leaked. There are challenges however, and these challenges will be addressed further in this master thesis.

In a refrigeration system using CO_2 the gas cooler pressure has to be controlled for transcritical operation. In traditional systems of this kind, a non-variable suction gas heat exchanger is specifically tailored to a specific operating condition. The optimum gas cooler pressure varies with the evaporation temperature and the cooling medium flow and temperature (Sarkar et al, 2004).

For systems undergoing large variations in operating conditions, it is impossible to design a suction gas heat exchanger (SGHX) that ensures optimum system efficiency for all conditions. In order to compensate for this, one can introduce a suction gas heat exchanger bypass, so that the SGHX efficiency (countercurrent heat exchanger efficiency) can be varied according to the current operating condition. The optimum gas cooler pressure then also has to consider the SGHX efficiency in addition to the evaporation temperature, cooling medium flow and temperature (Sarkar et al, 2004)

The main objective of this project was to investigate if there is any benefit of controlling both gas cooler pressure and suction gas heat exchanger efficiency in a RSW system, and create a tool for optimizing system efficiency using gas cooler pressure and choke inlet temperature as the controlled parameters.

2 The system

The system in question was originally built without a suction gas heat exchanger. The system was designed for transcritical operation with a high cooling water flow. Under-dimensioned evaporator outlet tubes caused liquid carryover (droplets in the compressor suction gas), which required the system to be run with reduced capacity. A suction gas heat exchanger was then installed, but at some conditions the compressor outlet temperature overheated. The last modification was the installation of a bypass solution, so that the suction gas heat exchanger efficiency can be modified depending on the operating condition.

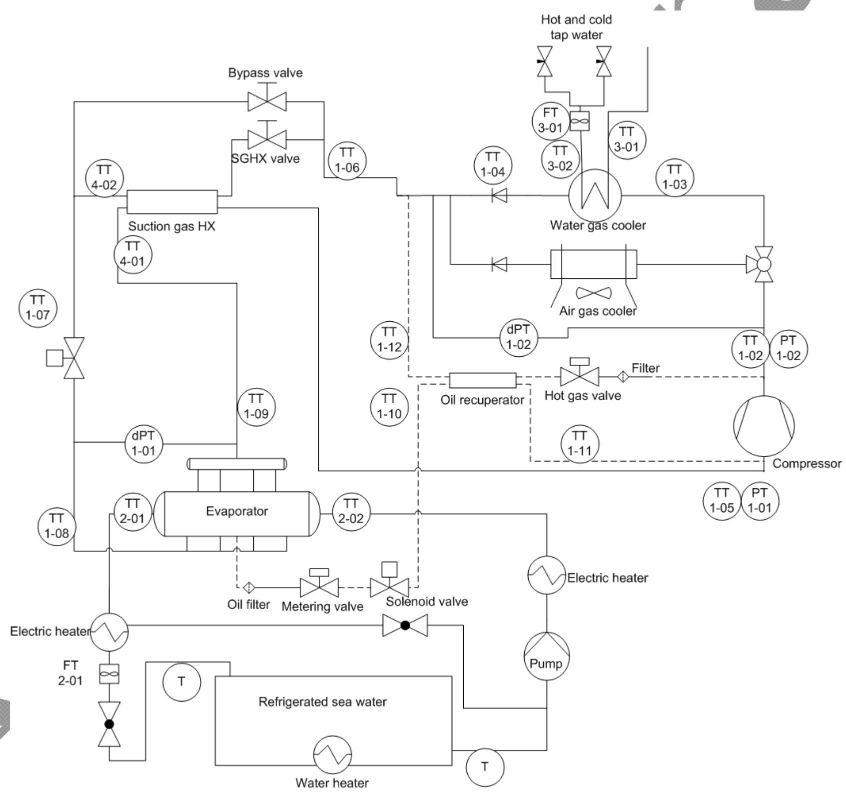


Figure 2-1: The system setup with measurement points

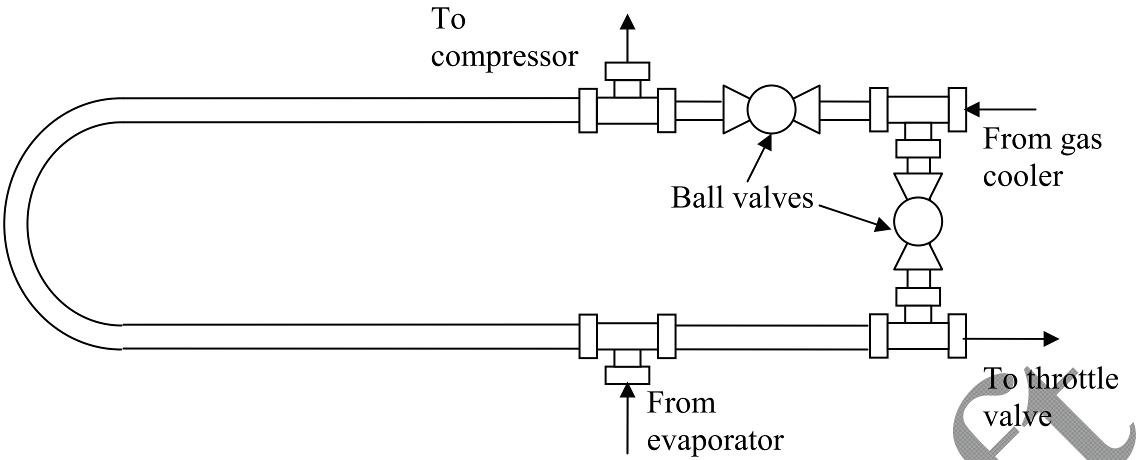


Figure 2-2: The modified suction gas heat exchanger. By opening the vertical ball valve and closing the horizontal ball valve one can bypass the suction gas heat exchanger.

The instrumentation consisted of

- 19 temperature measurements (thermocouples on tube walls, wrapped with alu-tape and insulation)
- 2 pressure transducers for absolute pressure measurements at the compressor inlet and outlet.
- 2 pressure difference measurements, across gas cooler and evaporator.
- 2 flow meters for measuring cooling water and brine (RSW) flow.
- The electric heaters had known power consumption, and the total heat input was logged manually.

A more detailed instrumentation description can be found in Jakobsen et al (2007). All measurements where logged using a computer logging software.

Table 2-1: The result of installing the suction gas heat exchanger.

Conditions	Cooling capacity (kW)			COP (-)		
	Old	New	Change (%)	Old	New	Change (%)
50 bar, 10°C	16,4	17,2	4,8	1,8	2,0	8,8
60 bar, 10°C	34,4	34,7	0,8	3,1	3,3	6,6
70 bar, 10°C	24,7	37,7	52,7	3,0	3,2	7,9
80 bar, 10°C	26,2	36,7	40,0	2,3	2,9	25,4
	2007	2008		2007	2008	
60 bar, 15°C	26,1	23,9	-8,4	2,3	2,2	-2,8
70 bar, 15°C	30,6	34,0	11,4	2,5	2,9	15,8
80 bar, 15°C	27,8	34,8	25,4	2,3	2,7	17,9
90 bar, 15°C	26,9	33,4	23,9	2,0	2,4	22,6
70 bar, 20°C	21,9	32,5	48,3	1,8	2,7	52,7
80 bar, 20°C	29,8	33,3	11,7	2,1	2,6	20,8
90 bar, 20°C	27,7	31,6	14,3	1,9	2,3	22,0
80 bar, 25°C	28,0	31,5	12,4	2,0	2,4	21,5
90 bar, 25°C	26,6	30,5	14,8	1,7	2,2	27,3

3 The system model

The system model was based on experimental data which was already available, both with (experiments from 2008) and without the suction gas heat exchanger installed (experiments from 2007). The model was kept fairly simple, but as will be shown, the results are adequate

The component models are based on libraries made by SINTEF for calculating thermodynamic properties. The libraries had an Excel front-end, which made it logical to use Excel and macros to create a simple system model. For most cases, traditional model parameters where used, with the exception of pressure drops in the gas cooler and evaporator. All thermodynamic data are calculated using these libraries.

rnlib (a refrigerant library) was used for calculating water properties and finding dynamic viscosity for carbon dioxide. *co2lib* (a library for calculation of thermodynamic properties based on Span-Wagners equation of state for carbon dioxide) was used to find all carbon dioxide related properties except dynamic viscosity, and finally *htclib*, a library for calculation of convection coefficients, was used for heat transfer calculations for both carbon dioxide and water in the gas cooler model.

3.1 Compressor model

The compressor has a swept volume of 143.14ccm per revolution, giving a swept volume (V_{swept}) of 12.46 m^3/hour at full speed (1451 RPM). Some initial tests after installation indicated that the assumed isentropic and volumetric efficiencies previously used for calculation were not suitable, and it was decided to make new models based on previous measurements. The model uses the pressure ratio (p_r) as a variable to calculate volumetric (λ) and isentropic (η_{is}) efficiency. Both models are based on the work performed on the gas, calculated using the inlet and outlet properties, as well as the circulated mass of CO_2 based on the gas cooler heat balance.

The isentropic efficiency was calculated from previous experiments as

$$\eta_{is} = \frac{h_{\text{compr out,isentropic}} - h_{compr,in}}{h_{\text{compr out,measured}} - h_{compr,in}} \tag{3.1}$$

and the volumetric efficiency as

$$\lambda = \frac{\dot{m}_{co2,gc} \, v_{compr,in}}{\dot{V}_{swept}} \tag{3.2}$$

The final compressor model uses the following equations:

$$\dot{V}_{swept} = 0.000143139 \cdot \frac{RPM}{60} \left[\frac{m^3}{s} \right] \tag{3.3}$$

$$\dot{V}_{suction} = \dot{V}_{swept} \cdot \lambda \tag{3.4}$$

$$h_{discharge} = h_{suction} + \frac{(h_{isentropic} - h_{suction})}{\eta_{is}} \tag{3.5}$$

$$\eta_{is} = -0.0442775\ p_r^2 + 0.1636479\ p_r + 0.5233507 \tag{3.6}$$

$$\lambda = -0.1427691\ p_r^2 + 0.6725351\ p_r + 0.0003813 \tag{3.7}$$

$$p_r = \frac{p_{discharge}}{p_{suction}} \tag{3.8}$$

$$\dot{m}_{circulated} = \dot{V}_{suction} \cdot \rho_{suction} \tag{3.9}$$

$$\dot{W}_{compressor} = \dot{m}_{circulated}\left(h_{discharge} - h_{suction}\right) \tag{3.10}$$

3.2 Gas cooler model

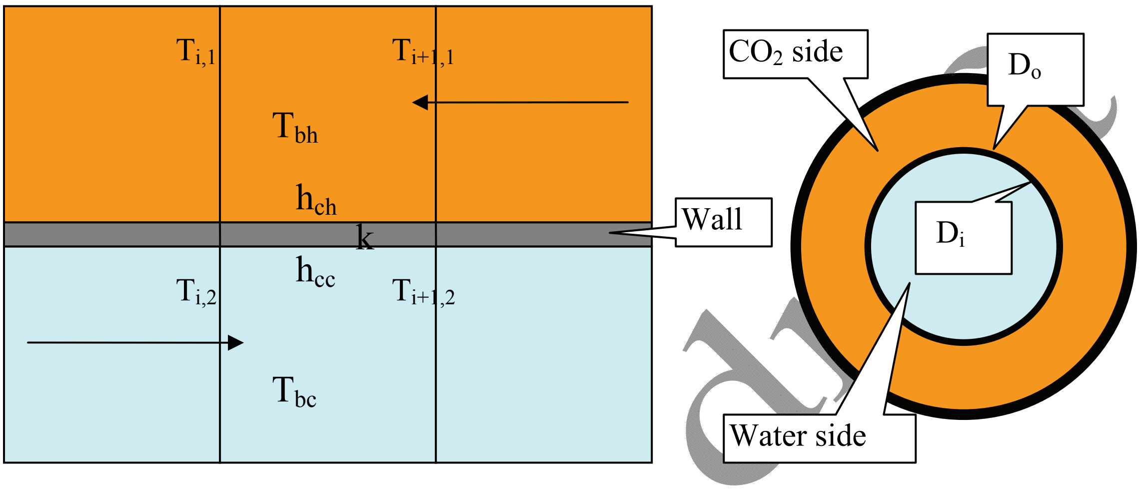


Figure 3-1: The finite elements, directions, symbols and indexing used in the gas cooler simulation.

$$\dot{Q}_{convective} = \frac{T_{high} - T_{low}}{R_{tot}} \tag{3.11}$$

$$R_{tot} = \frac{1}{A_{inner} h_{inner}} + \frac{L/n_{sections}}{2\pi k}\ln(D_o/D_i) + \frac{1}{A_{outer} \cdot h_{outer}} \tag{3.12}$$

$$\dot{Q}_{flow} = \dot{m}_{co2}\left(h_{co2,gc\ in} - h_{co2,gc\ out}\right) \tag{3.13}$$

$$Error = Q_{convective} - Q_{flow} \tag{3.14}$$

T_{high} and T_{low} are the bulk temperatures in the section on the high and low-pressure side of the gas cooler, calculated as the section average of the inlet and outlet temperature. h is the convective heat transfer coefficient, k is the thermal conductivity of the wall. D_i is the inner diameter of the small inside tubes (CO_2 tubes), D_o is the outer diameter. A_{inner} is the section area based on the inner circumference and section length, and A_{outer} is the section area based on the inner tube's outer circumference. The wall heat resistance equation (3.12) was based on equations from Incropera et al (2007c).

The first attempts to create a gas cooler model showed poor accuracy for low cooling water flow rates. The heat transfer for low water flow rates where underestimated. An explanation can be that in real life, the tube bends in the gas cooler induces more turbulence than a straight tube, and consequently the model underestimated the heat transfer for low cooling water flows, giving too high gas cooler outlet temperatures. In order to compensate for this, it was decided to introduce a factor that that narrows the effective tube cross sectional area, thereby increasing the convection coefficient. The factor decreases linearly until 600 l/h:

$$F_{reduction} = \max(1 + 0.001 \cdot (600 - massflow);1.0)\,, \quad \text{where } massflow \text{ is the cooling water flow in kg/hour} \tag{3.15}$$

At 10°C cooling water, 600 l/h corresponds to a Reynolds number (Re) of approximately 3000 in the gas cooler. The transition from laminar to turbulent flow is usually set to 2300, but fully turbulent conditions are not reached until Re is around 10000 (Incropera et al, 2007d), so a shift below Re of 3000 is understandable, and can be the cause of the need for such a modification. The Reynolds number was calculated using the equation

$$\mathrm{Re} = \frac{UD_h}{\upsilon}\,, \tag{3.16}$$

where U is the bulk water velocity, D_h is the hydraulic diameter for an annulus and ν is the kinetic viscosity.

$$dP_{GC} = (1.8469\ \dot{m}^2\ -\ 9.1012\ \dot{m})\ \cdot 10^3\ [\mathrm{Pa}] \tag{3.17}$$

3.3 SGHX model

In the 2008 experiments, the suction gas heat exchanger (SHGX) showed a quite stable U-value in the transcritical area, with values ranging from 1425.8-1572.0 W/m2K, with an average of 1479.63 W/m2K, based on the inner area. The average value was used to estimate the maximum performance of the SGHX. If the heat transfer exceeded this value, the result was discarded (see equation (3.21)).

Heat transfer from the high pressure side in the SGHX was calculated as follows:

$$\dot{Q}_{\max} = \dot{m} \cdot \min((h_{in} - h_{out})_{hot}, (h_{out} - h_{in})_{cold} \tag{3.18}$$

$$\dot{Q}_{transferred} = \dot{Q}_{\max} \cdot \varepsilon_{SGHX} \tag{3.19}$$

$$h_{out} = h_{in} - Q_{transfer} / m \tag{3.20}$$

In order to check for a viable solution (obtainable in the actual system), a validity check for suction gas heat exchanger was introduced:

$$\dot{Q}_{transfer} <= \Delta T_{LMTD} \cdot A_{IHX} h_{\mathrm{experimental}} \tag{3.21}$$

where $h_{experimental}$ is the average heat transfer coefficient and the logarithmic mean temperature difference (Incropera et al, 2007b):

$$\Delta T_{LMTD} = \frac{(T_{SGHX,HPin} - T_{SGHX,LPout}) - (T_{SGHX,HPout} - T_{SGHX,LPin})}{\ln \dfrac{(T_{SGHX,HPin} - T_{SGHX,LPout})}{(T_{SGHX,HPout} - T_{SGHX,LPin})}} \tag{3.22}$$

3.4 Evaporator model

$$dP_{evap} = (2 \cdot 10^7 \cdot V_{satflow}^2 - 26466 \cdot V_{satflow}) \cdot 10^3 \ [\mathrm{Pa}] \tag{3.23}$$

$$V_{satflow} = \dot{m} \cdot v_{satgas}(t_{evap}) \tag{3.24}$$

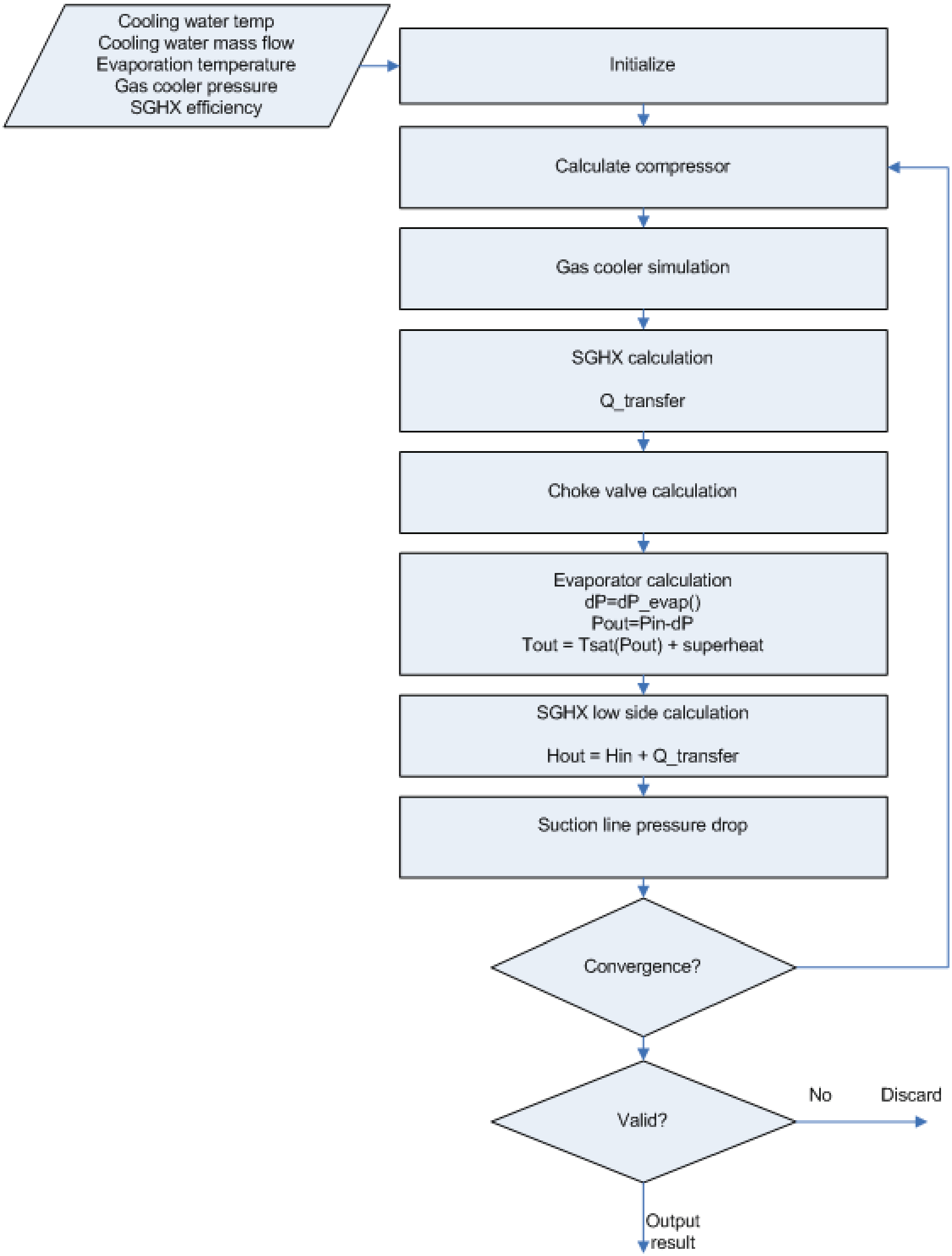


Figure 3-2: The iterative approach for solving the system.

4 Optimization results

The optimization process is focused on transcritical operation, as this has particular use for high pressure control. The goal of this master thesis is not to find the perfect optimization algorithm; the purpose here is to see if the concept of a variable SGHX can be beneficial. The simulations where run with a number of combinations of pressure and SGHX efficiency, and then automatically choosing the setting combination which provided the highest COP. As the SGHX efficiency is not directly measurable in a real system, the throttling valve inlet temperature was selected as the controlled parameter in addition to the gas cooler pressure.

4.1 Goal function

The system simulations calculate all properties of the system. The goal function outputs the COP (neglecting brine pump work), and by running combinations of gas cooler pressure and SGHX efficiency the highest COP for each cooling water flow, cooling water temp and evaporation temperature (later referred to as operating conditions) was selected.

The system COP was calculated using the cooling capacity and compressor work,

$$COP = \frac{h(T_{evap,out}, P_{evap,out}) - h(T_{choke}, P_{gc,out})}{h(T_{compr,out}, P_{compr,out}) - h(T_{compr,in}, P_{compr,in})} \qquad (4.1)$$

If the system did not converge or the converged result violated any of the restrictions set (exceeding the maximum compressor outlet temperature (130°C) or the heat transfer was higher than obtainable in the SGHX), the COP was set to zero.

For water heating (combined cooling and heating) it was necessary to use a trial and error approach on the cooling water flow in order to reach the desired outlet temperature, since the highest COP for a given cooling water flow not necessarily meant that the correct water temperature had been reached.

4.2 Proof of concept

In order to show the use of the variable SGHX concept, a series of simulations was run with a theoretical suction gas heat exchanger with a maximum efficiency of 0.9; in other words, the SGHX heat transfer restriction was temporarily removed from the system simulation model. The simulations were run at full cooling water flow (5000 kg/h) with inlet temperatures ranging from 10 to 50 °C. Simulations where also run with constant SGHX efficiencies of 0.3 and 0.6 in order to plot the differences. The compressor outlet temperature was limited to 130 °C.

Table 4-1: Combinations used for proof of concept

Variable	Low	High	Step size	# Steps
Pressure [bar]	74	110	2	19
SGHX efficiency [-]	0	0.9	0.1	10
Total combinations				190

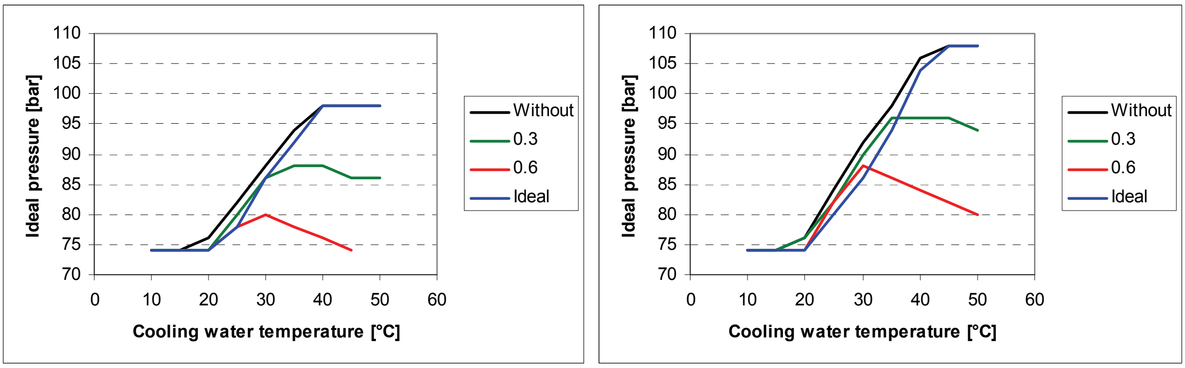


Figure 4-1: The ideal high side pressure (absolute, bara) at evaporation temperature -6°C (left) and -2°C (right)

As seen in Figure 4-2, the COP can deviate far from the optimum with different cooling water temperatures for a given SGHX efficiency. The figure also shows that the SGHX efficiency has very little influence on the system performance for low cooling water temperature. This confirms the statement by Sarkar et al (2004): *"...the performance of internal heat exchanger has a minor influence on system optimization at low and moderate gas cooler exit temperatures*". The concept of a suction gas heat exchanger is therefore expected to only give minor improvements in system efficiency when in cooling mode (high cooling water flow).

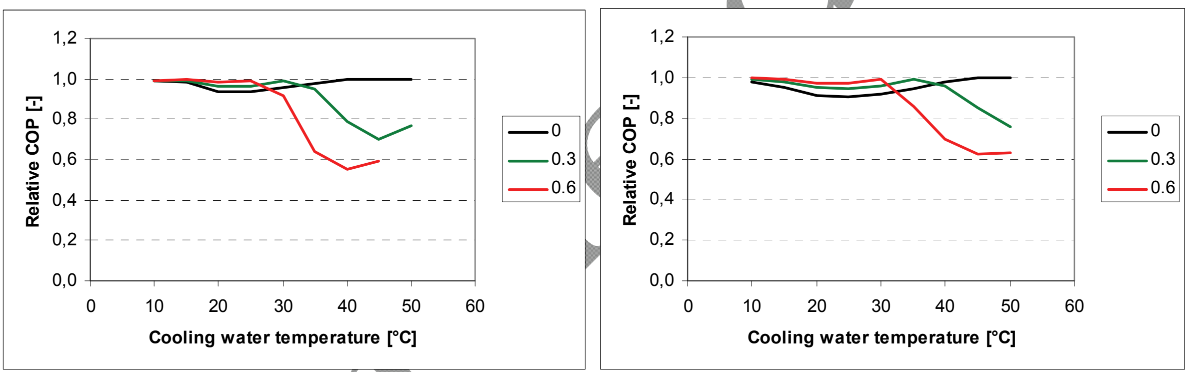


Figure 4-2: Relative COP using different SGHX efficiencies for evaporation temperature -6°C (left) and -2°C (right). The relative COP using the ideal SGHX efficiency is unity (1). A relative COP of 0.8 means the setting gives a COP of only 80% compared to the optimum setting.

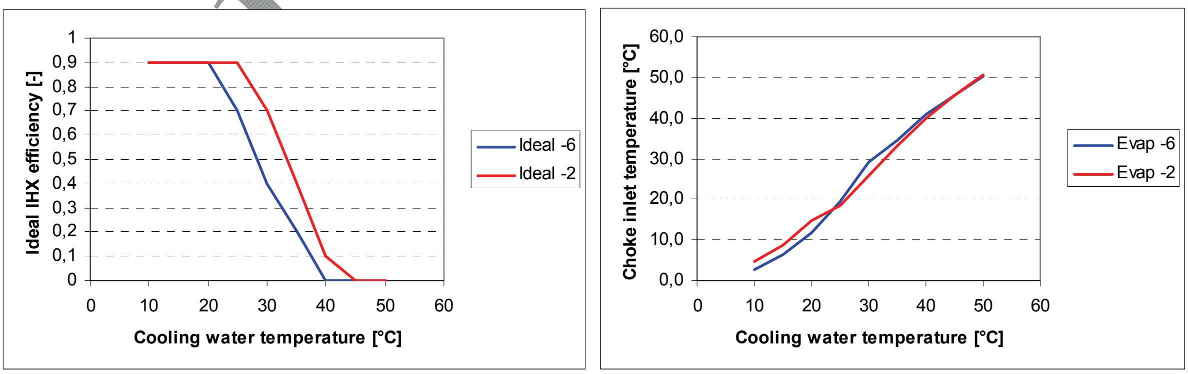


Figure 4-3: The ideal IHX (SGHX) efficiency (left) and the corresponding choke valve inlet temperature (right) as a function of cooling water temperature for different evaporation temperatures.

4.3 Cooling results

For cooling, an evaporation temperature of -6°C was selected based on previous experimental results. The variable combinations were can be found in Table 4-2. Figure 4-4shows the ideal pressure and choke valve inlet temperatures for 5000 kg/h cooling water flow at varying evaporation temperature for different cooling water temperatures.

Table 4-2: Variable combinations for each operating condition (cooling water temp, cooling water flow, evaporation temperature) used for batch simulations.

Variable	Low	High	Step size	# Steps
Pressure [bara]	74	110	2	19
SGHX efficiency [-]	0	0.6	0.1	7
Total combinations				133

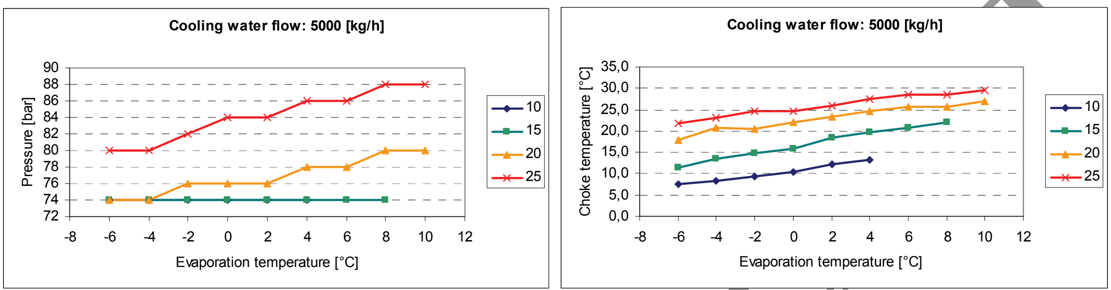


Figure 4-4: The optimum pressure (left) and optimum choke inlet temperature (right) as a function of the evaporation temperature, at different cooling water temperatures.

4.4 Heating results

Water heating was previously not possible (2008 experiments) using the installed suction gas heat exchanger because of overheating at the compressor outlet.The optimization results show that no internal heat exchange is desired for hot water production, and that the maximum pressure before overheating is 108 bar at -2 °C (the evaporation temperature was kept constant at -2°C), and this pressure remains constant for all the low water flows. Note that the goal was set to produce the desired hot water temperature at the highest possible COP. Higher COP was possible for the given flow rates, but these resulted in lower hot water temperatures. If one were to use a higher SGHX efficiency, the pressure would have to be reduced significantly in order to prevent overheating at the compressor outlet (Figure 4-1), which would have reduced the system efficiency.

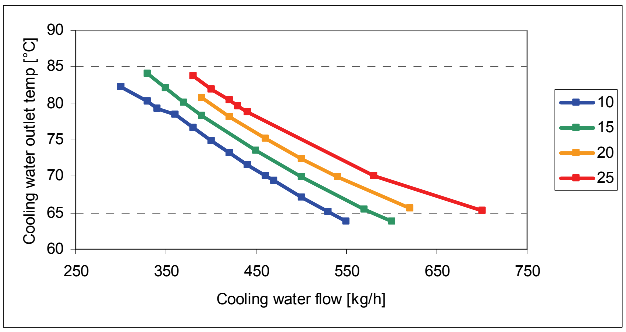


Figure 4-5: The outlet temperatures for different cooling water flows and cooling water temperatures.

5 Experimental results

The system has manual control of gas cooler pressure (via the choke valve), cooling water temperature and flow, SGHX bypass, compressor speed and heat input. This made it demanding to keep constant system input. The measurements where logged for 20-30 minutes with one minute intervals in order to get good average values.

The experimental performance was calculated based on the circulated mass flow of carbon dioxide in the system. The flow rate was calculated using heat balance over the gas cooler. The refrigeration capacity was calculated as if the evaporation outlet was saturated or superheated, depending on temperature. Thus, COP values may be slightly exaggerated, as liquid out of the evaporator does not contribute to the cooling capacity.

Table 5-1: Experiment matrix.

	Cooling water temperature			
Goal	**10°C**	**15°C**	**20°C**	**25°C**
Cooling at medium cooling water flow rate		X	X	X
Cooling at maximum water flow rate	X	X	X	X
Combined cooling and hot water, 65°C outlet	X	X	X	X
Combined cooling and hot water, 70°C outlet	X	X	X	X
Combined cooling and hot water, 80°C outlet	X	X	X	X

5.1 *Cooling*

For the cooling mode experiments the cooling water flow rates were kept high and the gas cooler pressure just above transcritical, as shown in the optimization chapter. The evaporation temperature was kept close to constant, and the choke inlet temperature was kept within ±1K of the values found in the optimization. At medium cooling water flow, tables with a wider cooling water flow rates were created and linear interpolation was performed as on the fly as it was difficult to precisely adjust the cooling water flow to a specific value.

It was difficult to compare medium water flow results, as cooling water flow and temperatures were not exactly as set in the previous experiments. As a workaround the experiments were run using the found optimum settings, and then the simulations were run after hand using the actual data (evaporation temperature, cooling water temperature and cooling water flow) from the experiments as simulation input. In this way the model accuracy could be verified. This was done for all the cooling experiments. Simulated performance of COP and cooling capacity are within a 5% error margin when compared to the actual experimental values (Figure 5-1).

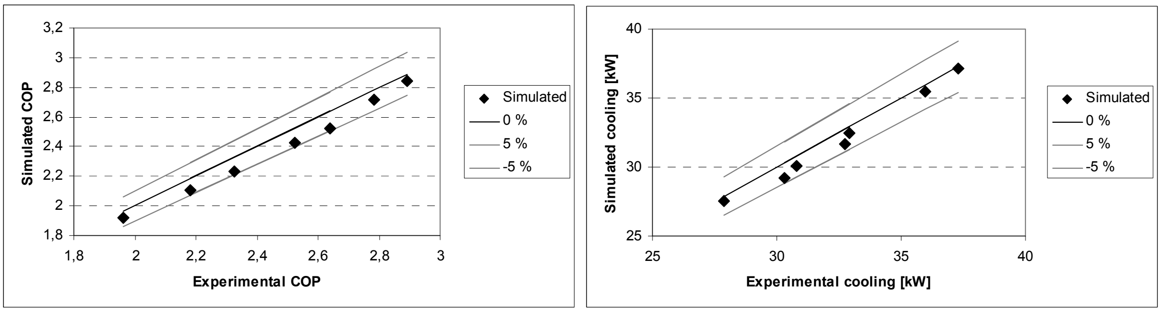


Figure 5-1: Model accuracy, cooling. COP (left), cooling capacity (right)

5.2 Heating

There were no earlier experiments using a suction gas heat exchanger, so the experiments were run with settings found through the simulations, and when completed, simulations were run again using the actual data from the experiments and compared (as for the cooling).

COP calculations were quite accurate, but the 20/70 and the 25/80 experiments deviated more than the rest. This could be linked to problems with the cooling water flow measurements, which behaved rather odd at low cooling water flows, even after recalibrating the flow measurement device.

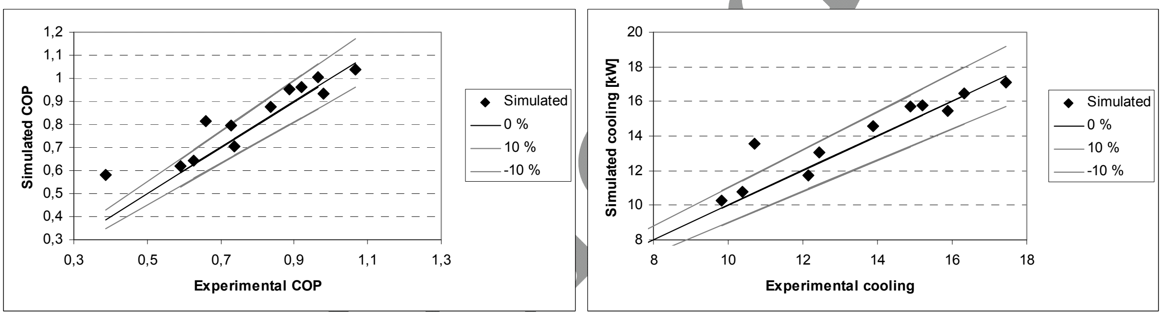


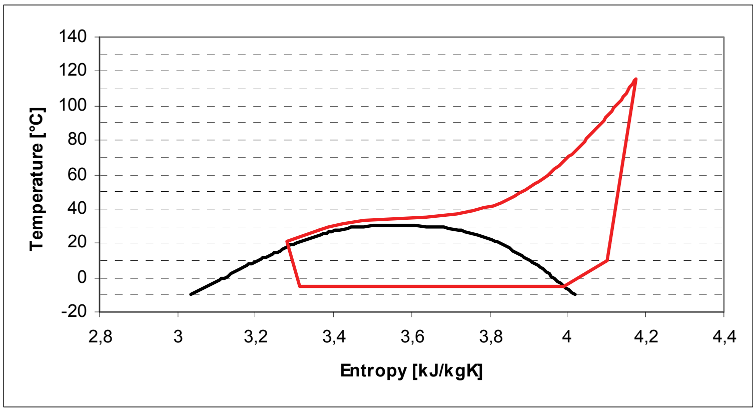
Figure 5-2: Model accuracy, heating. COP (left), cooling capacity (right)

6 Discussions

6.1 System performance

A lower gas cooler pressure limit of 74 bar was set in order to ensure transcritical operation, as the model was created for that purpose, but higher efficiencies can be achieved using a lower pressure for high cooling water flow and/or low temperature (as shown in previous experiments, Table 2-1), but this was not tested in this thesis.

The high cooling water flow experiments showed no significant change compared to the 2008 experiments. This was expected due to low sensitivity to the SGXH efficiency). The temperature approach was low due to large temperature difference in the gas cooler, which resulted in very good heat transfer (Figure 6-1).



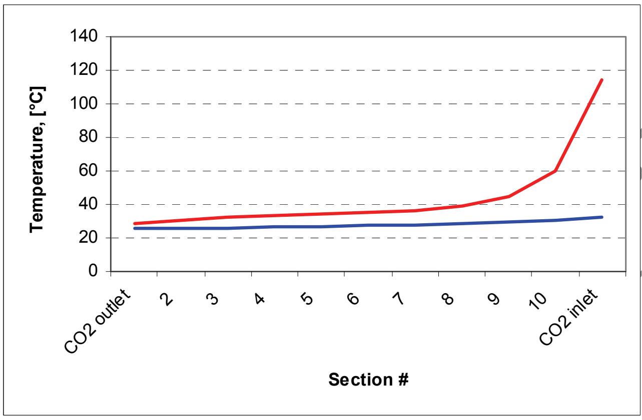
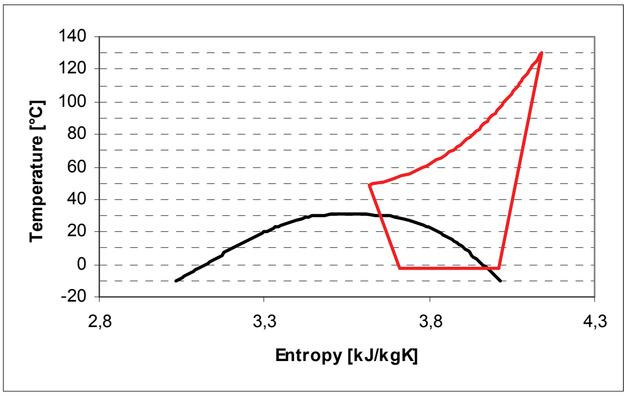


Figure 6-1: Cycle (upper) and gas cooler temperature profile (lower) at full water flow using 25 °C cooling water temperature. The blue line is the cooling water temperature.

The 25 °C cooling water temperature experiment actually showed reduced efficiency, but the 2008 experiment at 80 bar was very close to the optimum choke temperature setting, with slightly higher cooling water flow and evaporation temperature, which is probably the cause of the slight COP reduction.

Combined cooling and water heating on the current system showed very poor performance. The gas cooler temperature approach was very high (Figure 6-2), which resulted in low efficiency and low cooling capacity. A gas cooler with higher performance should be able to improve the performance by reducing the temperature approach.

By running simulations using the conditions from the experiments, it was possible to check the validity of the model, and it shows good accuracy, within 5% error for the cooling capacity and system efficiency (COP), and within 10% error for most of the water heating experiments. The model shows some improvement potentials, but it was accurate enough for the intended use.



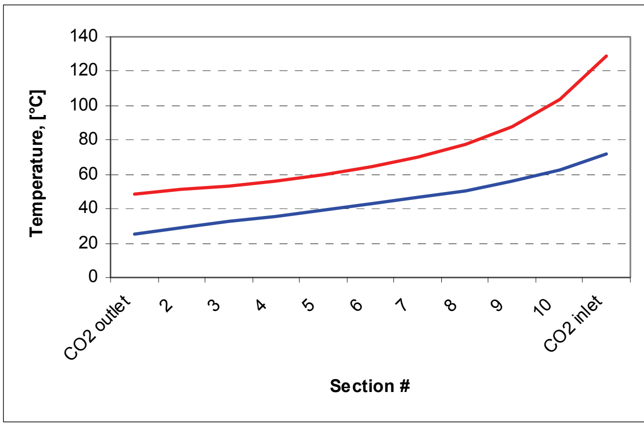


Figure 6-2: Cycle (upper) and gas cooler temperature profile (lower) at heating mode with 70°C outlet temperature using 25 °C cooling water. The temperature difference between the CO_2 and the water was much lower than for the full flow experiment, so less heat was transferred.

6.2 System control

In a traditional transcritical system the gas cooler temperature approach is used as the control parameter for finding the optimum high side pressure. This works fine with a constant suction gas heat exchanger. When introducing a variable suction gas heat exchanger, the gas cooler outlet temperature can no longer be used for this purpose, as it is affected both by the gas cooler pressure and the efficiency of the suction gas heat exchanger. The high side pressure therefore has to be controlled by other means, for instance cooling medium temperature and flow. The evaporation temperature also has to be taken into account (Sarkar et al, 2004).

For controlling of the suction gas heat exchanger, the choke inlet temperature can be used as a control parameter. As can be seen from the results, the simulations are accurate enough to use the choke inlet temperature as the control parameter. An initial idea is to generate lookup-tables from which a controller can interpolate to find set points.

6.3 Model accuracy

The model predicts the system behavior better than expected. Even though the model was very simple, cooling efficiencies (COP) were calculated within 5 % error margin. In the combined cooling and heating experiments (water heating) the simulation error was within 10% (with a few exceptions), and is considered good enough for optimization purposes. The reason for the low errors is likely to be caused by the modeling process, which used previous measurement data. The outliers are probably caused by varying cooling water flow at low flow rates.

The evaporator pressure drop model should have been revised after installing the suction gas heat exchanger. However, the pressure drops are at a very small scale, and the calculation of overall system efficiency seems to be almost unaffected by this. The gas cooler pressure drop is a result of using the erroneous mass flow calculations. The error should have been caught earlier, but the pressure drops are very small compared to the total pressure, so this should have almost no impact on the system performance calculations.

6.4 The concept of variable SGHX bypass

As shown in section 4.2, the benefit of a variable SHGX becomes increasingly more evident when cooling water temperature exceeds 25 °C (or achievable gas cooler outlet temperature above 25°C, since the temperature approach is very small for high cooling water flow rates). For the system in question, because of the low gas cooler outlet temperatures, it is therefore difficult to determine if the concept of a variable SGHX is beneficial.

The simulations show that a RSW system with an efficient gas cooler and low cooling water temperature (up until around 25°C) available does not need a variable SGHX; a high efficiency SGHX will be satisfactory. In this kind of setup, the gas cooler temperature approach can be used for simple gas cooler pressure control.

It's when the cooling water temperature/gas cooler outlet temperature exceeds 25°C that the variable suction gas heat exchanger starts to show real benefit. As can be seen in Figure 4-3 the ideal SGHX efficiency declines with increasing cooling water temperature until zero efficiency (full bypass). Such high cooling temperatures are unlikely for RSW refrigeration using sea water as the cooling medium, but the concept should be useful for refrigeration systems which utilize air as the cooling medium (where the air temperature is high, and the gas cooler temperature approach will be higher). Using air as a cooling medium generally means a higher temperature approach, and will have the same impact on the system as increasing the cooling water temperature. Also, ambient air temperature tends to experience greater variation than sea water.

A word of advice for designing the suction gas heat exchanger is to calculate the UA value for the SGHX at all simulation points, and then select the highest one for manufacturing, as the highest efficiency may not have the highest UA-value (Watts/ΔK). An observation is that too high SGHX efficiency is far worse than not using an SGHX at all (ignoring liquid carryover), as can be seen from Figure 4-2. The best example of this is the 2008 experiments, where hot water production was not even possible due to non-ideal SGHX efficiency. So if it is decided to not use a variable SGHX even though high gas cooler outlet temperature may occur, the efficiency of the SGHX should be kept low enough to allow for sufficiently high gas cooler pressure at the highest cooling medium temperatures.

7 Conclusions

The simulation results from the developed simulation tool indicate that the RSW system performance can be increased compared to running a system with a traditional non-variable suction gas heat exchanger, but only when the gas cooler outlet temperatures are high (above ~25°C), which for the system in question would mean high cooling water temperatures or reduced cooling water flow rate. Such operating conditions are not to be expected for a RSW system with low cooling water temperatures available.

The cooling experiments with transcritical gas cooler pressure show little or no improvement in system performance at temperature levels of 25 °C and below compared to previous experiments, as the optimum setting turned out to be maximum suction gas heat exchange (no bypass). Previous experiments also show that for low cooling water temperatures (20°C and below), the optimum gas cooler pressure is subcritical. The simulation model showed good accuracy in predicting COP and cooling capacity.

The introduction of a variable suction gas heat exchanger once again made it possible to use the system for water heating. The optimum operation was to bypass the SGHX altogether, and any use for a SGHX should be to evaporate liquid carryover from the evaporator. In other words, if the current system is to be used for water heating, the possibility to fully bypass the SGHX is essential.

The concept of a variable SGHX appears not to beneficial in a water cooled RSW system, but the simulations indicate that it has potential in areas where air is used as the cooling medium, for instance commercial or mobile refrigeration. For cooling purposes, experiments and simulations show that a non-variable suction gas heat exchanger should be satisfactory for a RSW system when low cooling water temperatures are available. It is strongly recommended that a system to be used for simultaneous cooling and heating should have an improved design compared to the current setup, as this mode of operation shows low cooling capacity and poor energy efficiency.

8 Bibliography

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