DANISH TECHNOLOGICAL INSTITUTE

Energy Savings on Refrigeration Systems by Minimizing Fouling in Water Systems

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Jesper Weinkauff Kristoffersen, Danish Technological Institute

Energy & Climate
Refrigeration and Heat Pump Technology

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1 Project Objectives

The project has been established through the cooperation between Danish Clean Water A/S and Danish Technological Institute.

The purpose of the project is to investigate the effect of the water treatment system, which Danish Clean Water A/S is delivering to an existing system.

The project is based on on-site measurements performed at Arla Foods Holstebro Mejeri. These measurements are used to compare the efficiency of the water treatment system before and after the system is applied.

In general, on-site measurements have a high degree of uncertainty and all results should therefore be evaluated with this in mind.

April 2013

Jesper Weinkauff Kristoffersen

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Refrigeration and Heat Pump Technology

2 Description of the Measurement Site

The refrigeration system from 2009 used in this project consists of four flooded evaporators, three screw compressors and five evaporative condensers. The four evaporators are identical and they are all connected to the same pump separator and cold water system. The three screw compressors are of the type Sabroe SAB 233 L and they are driven by a 450kW motor with VLT. The condensing side of the system consists of four identical Baltimore evaporative condensers from 2009 and one older and larger York/Baltimore evaporative condenser.

This project calculates on one of the evaporators and one of the condensers. Therefore, this report only states capacity, theoretical power consumption and possible savings as if it was a one stage ammonia system with one evaporator and one condenser. All results are scalable to the size and type of system installed at Arla Foods in Holstebro.

3 Cold Water PHE

The test subject is a Vahterus PSHE 7HH-368/2/1, which is a shell and tube plate heat exchanger used to produce cold cooling water. On the primary side, water is pumped through an evaporator and on the secondary side the water is cooled by evaporating ammonia.

3.1 Description of the Measurement Setup

A diagram of the system is shown in appendix A.

To determine the overall heat transfer coefficient of the PHE, the following has been measured:

- Evaporation pressure (temperature)
- Water temperature in and out of the PHE
- Water flow

The evaporator pressure is measured on the liquid separator with a pressure transmitter.

The water temperature in and out of the PHE is measured by the on-site SCADA system.

The water flow through the PHE is calculated by measuring the pump head and pump frequency.

The water flow measured via the pump head is compared with the total flow to the four evaporators measured by the SCADA system TA01FT01.

3.2 Results

All the results are based on average values above ten minutes. Moreover, the results are based on periods of near steady state conditions.

The capacity of a heat exchanger is given by:

$$Q = U \cdot A \cdot LMTD$$

$Where:$

$Q: capacity$
$U: Overall\ heat\ transfer\ value$
$A: Heat\ transfer\ area$
$LMTD: Logarithmic\ mean\ temperature\ difference$

The results for the PHE are presented in Figure 1.

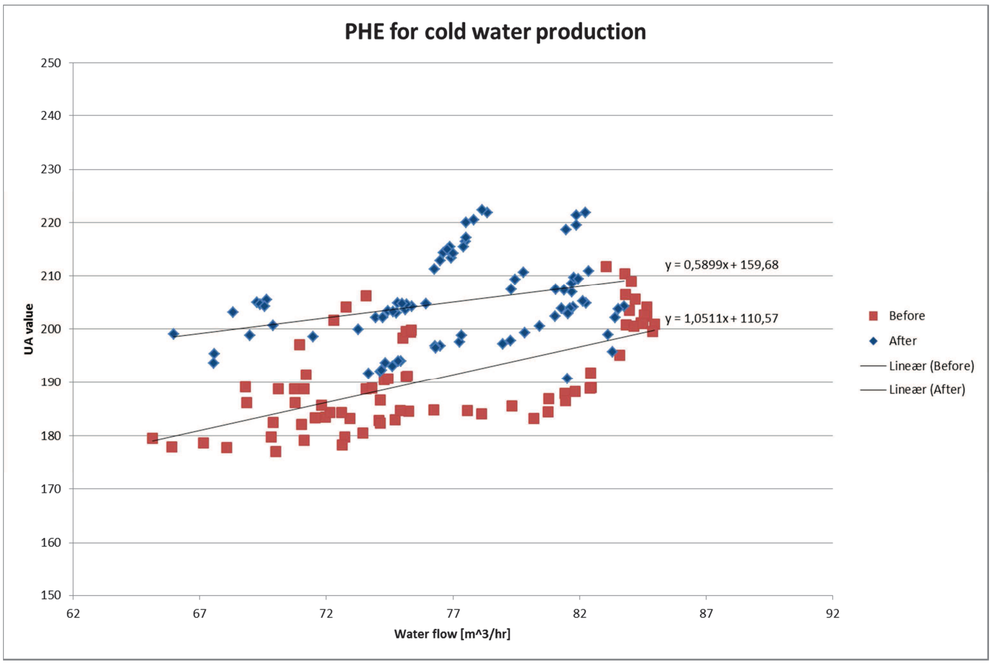


Figure 1 – Results of the plate heat exchanger

The results of the plate heat exchanger indicate a tendency of a slight improvement. However, this improvement is very small and the uncertainty in the measurements might therefore have an influence on the results.

In laboratory tests of air to air heat pumps, an uncertainty below 5% may be difficult to achieve.

The total water flow to the four evaporators has been compared with the flow calculated via the pump head. The largest difference observed was 1,5 m^3/hr.

4 Evaporative Condenser

The test subject is a Baltimore Aircoil Model CXV 481, which is an evaporative condenser. This means that the main energy is transferred from the condenser to the outside air by evaporating water. Ammonia gas is condensing inside the tubes. The tubes are covered with a water film from a recirculating spray system. Air is blown past the outside of the tubes and in that way transferring energy from the water to the air, see Figure 2.

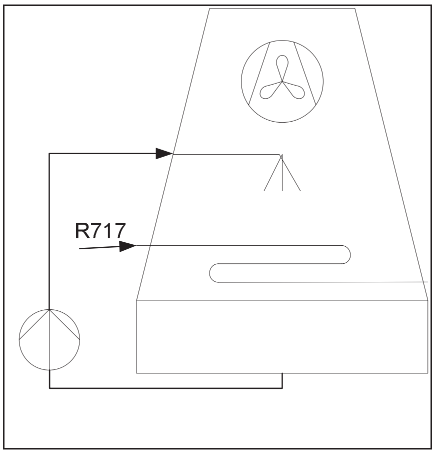


Figure 2 - Evaporative condenser

4.1 Description of the Measurement Setup

A diagram of the system is shown in appendix B.

The water treatment system provided by Danish Clean Water A/S only affects the water side of the evaporative condenser. Therefore, measurements focus on the air/water side of the condenser.

The following has been measured:

- Condensing pressure and temperature in and out of the ammonia
- Ammonia flow in the condenser
- Water temperature of the circulating water
- Air temperature and humidity at the inlet on the air side
- Frequency of ventilators

The ammonia temperature at inlet and outlet of the condenser is measured on the outside of the tube. The thermometers have been placed on the tube wall and subsequently, they have been covered with insulation to prevent the surrounding air from influencing the measurement.

The ammonia flow is measured with a clamp on the ultrasonic flow meter.

The water temperature is measured directly in the flow right before the inlet of the spray section of the condenser.

A sensor for measuring air temperature and humidity was placed at the air inlet behind the sound damping material.

The frequency of the ventilators and, thereby, a relative indication of the air flow were logged by the SCADA system. All three ventilators run in parallel.

4.2 Results

All the results are based on average values above ten minutes. Moreover, the results are based on periods of steady state conditions.

The efficiency of the evaporative condenser on the air side can be expressed in terms of:

$$\varepsilon = \frac{I_2 - I_1}{I_g - I_1}$$

I_1: Enthalpy of air at inlet
I_2: Enthalpy of air at outlet
I_g: Enthalpy of air, if it is in equilibrium with circulating water (maximum achievable)

The results are presented in Figure 3 and Figure 4.

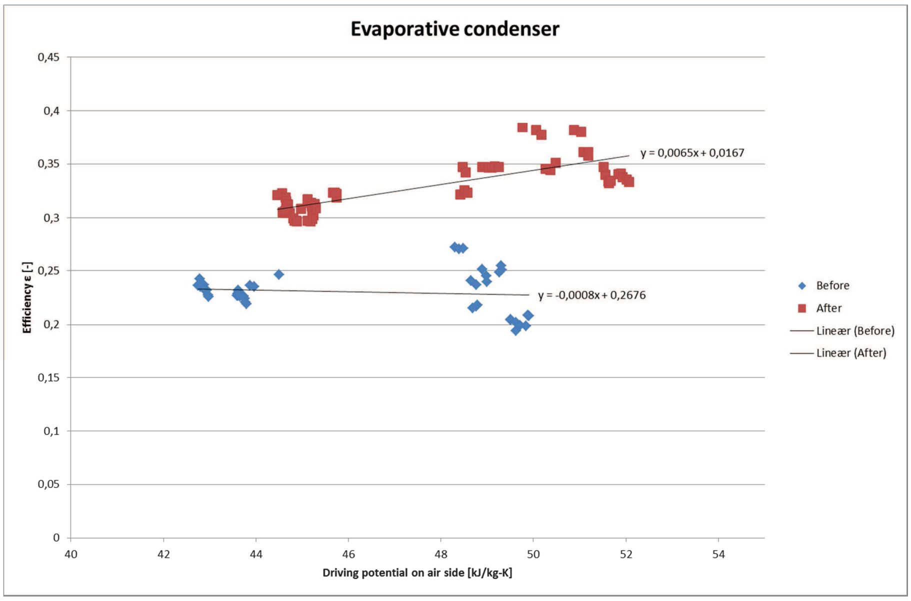


Figure 3 – Results of the evaporative condenser as function of driving potential[1]

The capacity of the system is generally higher during the after measurements. When comparing the before and after measurements, the data are narrowed down to an area of overlap in capacity as shown in Figure 4.

[1] The Efficiency ε is a relative value

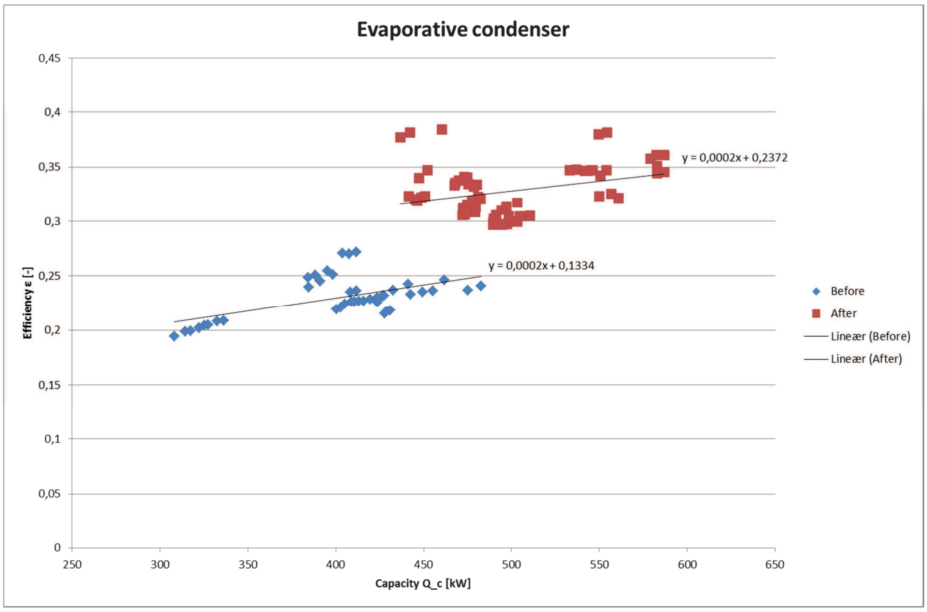


Figure 4 – Results of the evaporative condenser as function of condenser capacity[2]

[2] The Efficiency ε is a relative value

5 Energy Saving Potential

5.1 PHE for Cold Water Production

In general, the before and after measurements do not show a clear tendency of improvement. The relatively high water flow in a plate heat exchanger might flush fouling out of the heat exchanger. Furthermore, the operating temperatures do not provide a good environment for biofouling.

The comparison of the before and after UA values shows an average improvement of 7 %. When converted into energy savings for a typical refrigeration system, this gives a result of 0,6 %.

The calculation is shown in appendix C.

5.2 Evaporative Condenser

The improvement on the evaporative condenser efficiency can be converted into a drop in the condensing temperature of the refrigeration system. When lowering the condensing temperature, the efficiency, COP, improves.

Condensing temperature T_C	**Wet bulb T_w**	**$T_{C.new}$**	**$P_{comp.old}$ [kW]**	**$P_{comp.new}$ [kW]**	**Saving [%]**
20 ºC	3 ºC	17,1 ºC	45,5	38,5	15
25 ºC	6 ºC	21,9 ºC	58,0	50,2	13
30 ºC	20 ºC	28,1 ºC	71,1	66,1	7
35 ºC	18 ºC	32,2 ºC	84,7	77,1	9

5.2.1 Assumptions

1. Cooling capacity Q_e = 400 kW
The measurement period shows a cooling demand of approximately 400 kW for one evaporator.

2. Evaporation temperature T_e = 1ºC
The measurements on the PHE show that the evaporation temperature is quite constant around this temperature.

3. Water film temperature $T_{wf} = T_c$
All the results show a very small temperature difference between the circulating water on the outside of the tubes and the condensing temperature inside the tubes (<0,25 ºC). Therefore, the temperature gradient has been neglected.

4. Efficiency before applying water treatment ε_{old}: 0,25
The efficiency before the application of water treatment is shown in Figure 4 at 450 kW.

5. Efficiency after applying water treatment ε_{new}: 0,32
The efficiency after the application of water treatment is shown in Figure 4 at 450 kW.

6. Determination of a new possible condensation temperature.

The condenser capacity is used to determine the air flow based on the old efficiency. Subsequently, the air flow and the new efficiency are used to calculate the new enthalpy $I_{g.new}$.

$$Q_c = \dot{m}_{air} \cdot \varepsilon_{old} \cdot (I_g - I_1) = \dot{m}_{air} \cdot \varepsilon_{new} \cdot (I_{g.new} - I_1)$$

The new condensing temperature is a function of the new enthalpy and a relative humidity of 100%.

$$T_{C.new}(I_{g.new}; RH=1)$$

The calculation is shown in appendix D.

5.2.2 Comments

The stated energy saving potential is a theoretical calculated value. The system at Arla Foods, Holstebro, operates after floating condensing temperature until it hits a minimum required by the compressor. All measurements have been performed during cold weather and, thus, with a constant condensing temperature. Therefore, the before and after tests have shown a drop in air flow and, thereby, a lower power consumption of the ventilators. An optimum between lowering the condensing temperature and the fan speed should be considered.

The power consumption of the ventilators has not been measured and a value for the actual energy saving at Arla Foods is therefore not provided.

6 Conclusion

This project shows that it is possible to save 7-15% on the energy consumption of the compressors when using the water treatment system from Danish Clean Water A/S.

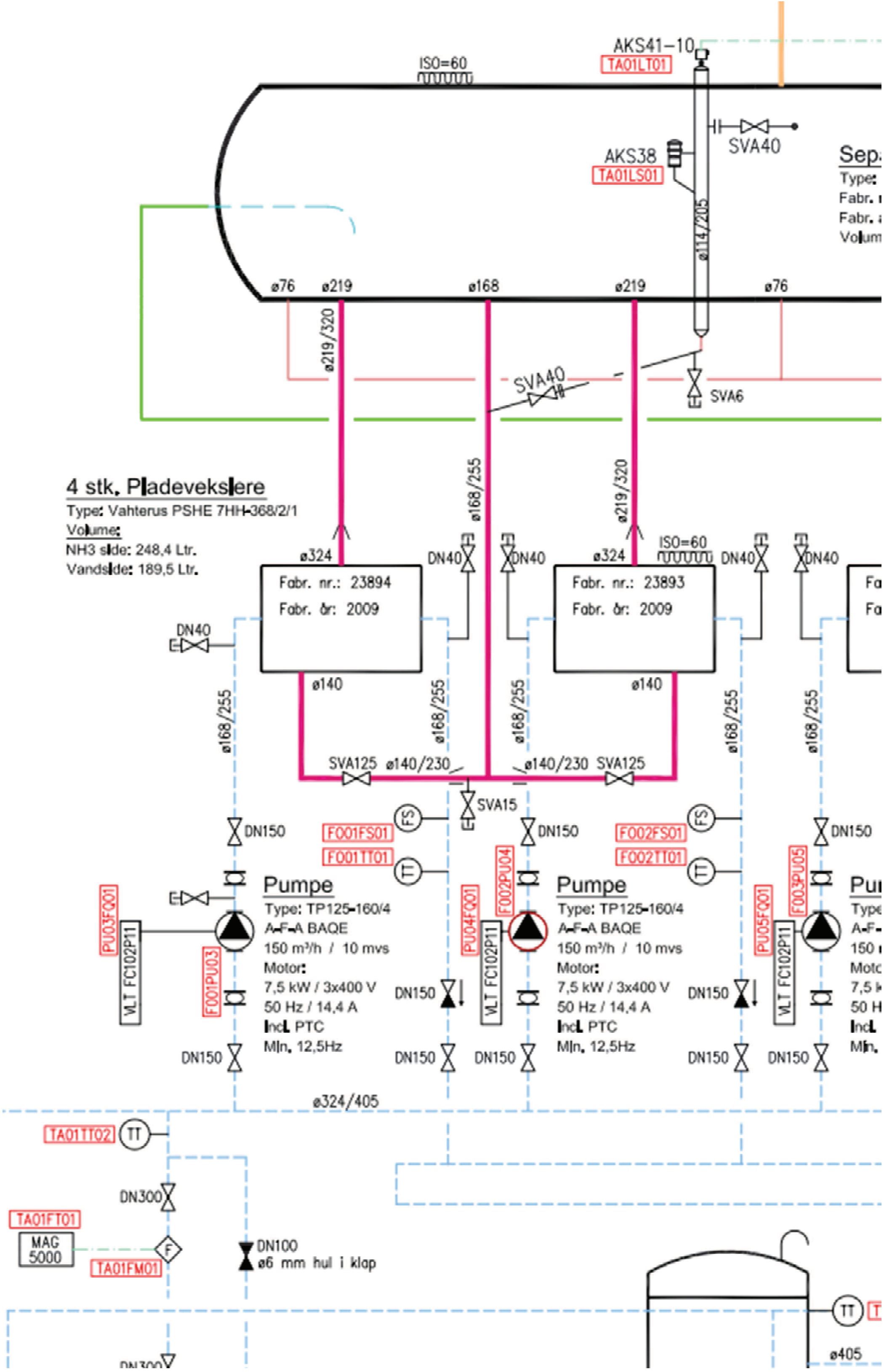
In terms of the cold water PSHE, no clear improvement was identified. The improvement is, however, very clear for the evaporative condensers.

The saving potential for this exact system is used as a test subject in this project. The saving potential depends on the level of biofilm in a system which differs greatly from system to system.

The calculated savings in this report is for a single stage system with one evaporator and one condenser. In connection with the system at Arla Foods, Holstebro, the improvement on the evaporative condenser used for test should be similar to the three other evaporative condensers in that they all share the same supply water. The saving potential is linear scalable with the capacity. The savings are calculated at fixed conditions and a dynamic yearly calculation has not been performed.

The uncertainty of the measurements is not dealt with in this project, but when carrying out before and after measurements, small offsets are not going to have a significant effect on the results.

Appendix A



AKS41-10
TA01LT01
ISO=60
SVA40
AKS38
TA01LS01
ø114/205
ø76
ø219
ø168
ø219
ø76
ø219/320
SVA40
SVA6
ø168/255
ø219/320
4 stk. Pladevekslere
Type: Vahterus PSHE 7HH-368/2/1
Volume:
NH3 side: 248,4 Ltr.
Vandside: 189,5 Ltr.
ø324
DN40
DN40
ø324
ISO=60
DN40
DN40
Fabr. nr.: 23894
Fabr. år: 2009
Fabr. nr.: 23893
Fabr. år: 2009
DN40
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ø140
ø168/255
ø168/255
ø168/255
ø168/255
ø168/255
SVA125
ø140/230
ø140/230
SVA125
SVA15
DN150
F001FS01
F001TT01
DN150
F002FS01
F002TT01
DN150
Pumpe
Type: TP125-160/4
A-F-A BAQE
150 m³/h / 10 mvs
Motor:
7,5 kW / 3x400 V
50 Hz / 14,4 A
Incl. PTC
Min. 12,5Hz
PU03FQ01
VLT FC102P11
F001PU03
PU04FQ01
F002PU04
PU05FQ01
F003PU05
DN150
DN150
DN150
DN150
DN150
DN150
DN150
ø324/405
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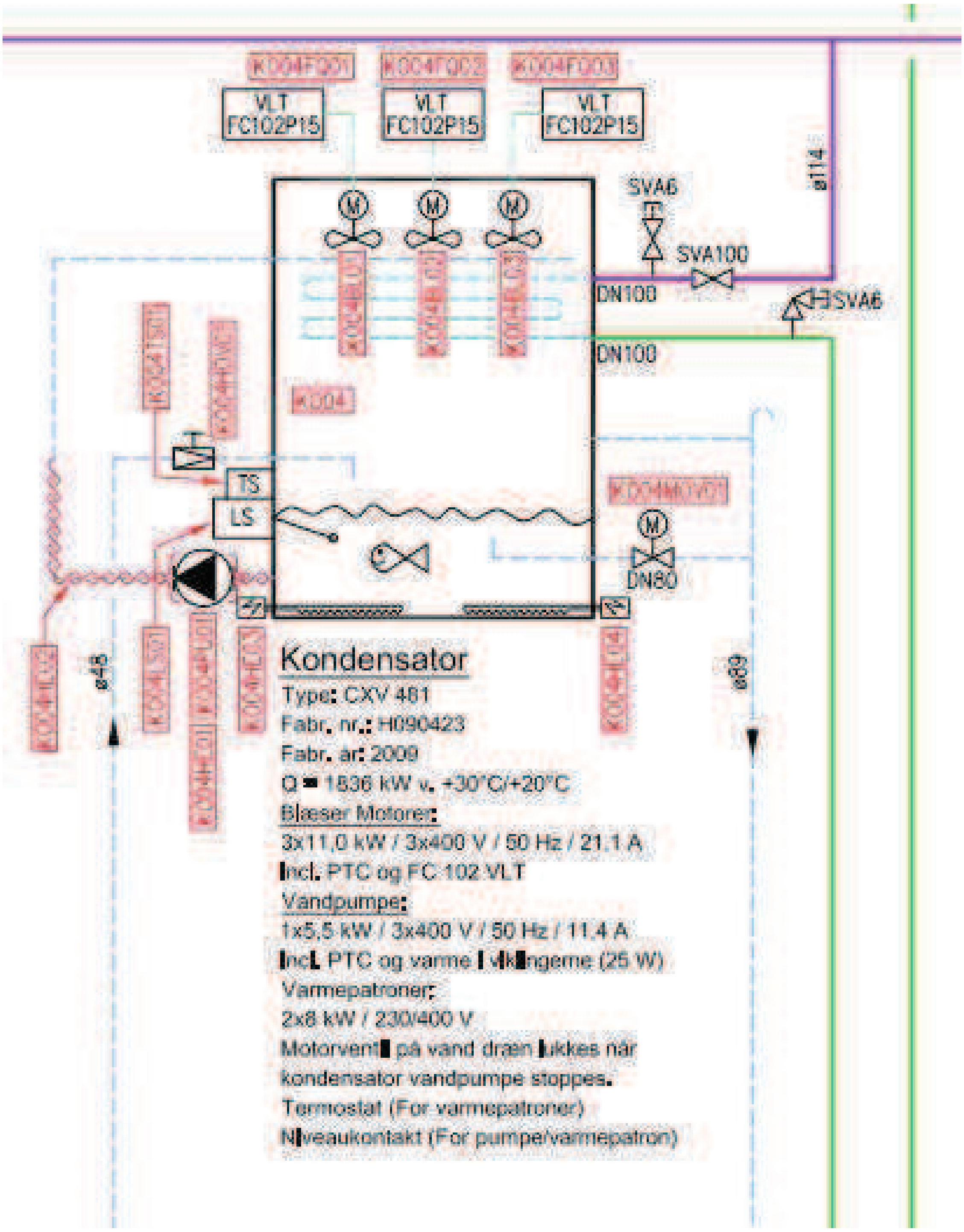
TT
DN300
TA01FT01
MAG 5000
TA01FM01
DN100
ø6 mm hul i klap
TT
ø405

Appendix B


K004FQ01
K004FQ02
K004FQ03
VLT FC102P15
VLT FC102P15
VLT FC102P15
SVA6
SVA100
DN100
SVA6
DN100
ø114
K004
TS
LS
K004MOV01
DN80
ø48
ø89
Kondensator
Type: CXV 481
Fabr. nr.: H090423
Fabr. år: 2009
Q = 1836 kW v. +30°C/+20°C
Blæser Motorer:
3x11,0 kW / 3x400 V / 50 Hz / 21,1 A
Incl. PTC og FC 102 VLT
Vandpumpe:
1x5,5 kW / 3x400 V / 50 Hz / 11,4 A
Incl. PTC og varme i viklingerne (25 W)
Varmepatroner:
2x8 kW / 230/400 V
Motorventil på vand dræn lukkes når
kondensator vandpumpe stoppes.
Termostat (For varmepatroner)
Niveaukontakt (For pumpe/varmepatron)

Appendix C

PROCESS SPECIFICATION FOR PRESENT SITUATION

EVAPORATOR

Evaporation temperature (T_E) [°C] 1,0 ΔT_{SH} [K] : 0

Temperature of air or water entering evaporator [°C] : 3

T_E : 1 [°C] p_E : 446,5 [kPa] UA-value : 200,000 [kW/K]

NOTE: Evaporating pressure = suction pressure

CONDENSER

Condensing temperature (T_C) [°C] 30 ΔT_{SC} [K] : 2

Temperature of air or water entering condenser [°C] : 25,0

T_C : 30 [°C] p_C : 1170 [kPa] UA-value : 92,668 [kW/K]

NOTE: Condensing pressure = discharge pressure

SUCTION GAS HEAT EXCHANGER (SGHX)

No SGHX: 0,0 T_4 : 28,0 [°C] η_T : 0,00 [-]

REFRIGERANT

R717

CYCLE CAPACITY

Cooling capacity ($\dot{Q}_E$) [kW] 400 $\dot{Q}_E$: 400,00 [kW] $\dot{V}_S$: 358,32 [m^3/h] $\dot{m}$: 0,354 [kg/s] η_{VOL} : 7,166 [-]

COMPRESSOR PERFORMANCE

Isentropic efficiency (η_{IS}) [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}$: 68,37 [kW] Displacement rate ($\dot{V}_D$) [m^3/h] : 50

COMPRESSOR HEAT LOSS

Heat loss factor (f_Q) [%] 10 f_Q : 10,0 [%] T_2 : 86,2 [°C] $\dot{Q}_{LOSS}$: 6,84 [kW]

COMPRESSOR INLET TEMPERATURE

T_1 [°C] : 3,0 $\Delta T_{SH,SL}$: 2,0 [K]

Calculate Print Help State Points Changes COP : 5,851 COP* : 5,877

CHANGE OF PARAMETERS: CONSEQUENCES FOR ENERGY AND ECONOMY

CHANGE OF PARAMETERS

CHANGES OF COMPONENTS [+/-]

Reduction of $\dot{Q}_E$ [%] : 0

Isentropic efficiency [%] : 0

Evaporator UA-value [%] : 7

Condenser UA-value [%] : 0

CHANGES ON SECONDARY SIDE OF EVAPORATOR

Temperature of fluid on secondary side is increased by [K] : 0,0

CHANGES ON SECONDARY SIDE OF CONDENSER

Ambient temperature for condenser is decreased by [K] : 0,0

ENERGY

OVERALL VIEW OF OPERATION (PRESENT & NEW SITUATION)

	$\dot{Q}_E$	$\dot{W}$	COP	T_E	T_C
	[kW]	[kW]	[-]	[°C]	[°C]
Present	400,00	68,37	5,851	1,0	30,0
New	400,00	67,98	5,884	1,1	30,0
% Changes	0,0	-0,6	0,57	-	-

COMPRESSOR

Change of compressor capacity : -0,47 [%]

ECONOMY

COST OF ENERGY AND HOURS OF OPERATION

Cost of one kWh : 0,7 Dkr

Hours of operation [h] : 5000

ANNUAL SAVINGS WITH NEW PARAMETERS

Savings : 1928 [kWh]

Savings : 1350 [Dkr]

Calculate | Print | Help | State Points | Process | COP : 5,851 | COP* : 5,877

Appendix D

CYCLE SPECIFICATION

TEMPERATURE LEVELS	PRESSURE LOSSES	QUALITY OUT OF EVAPORATOR	REFRIGERANT
T_E [°C] : 1,0 T_C [°C] : 20,0 ΔT_{SC} [K] : 1,0	Δp_{SL} [K] : 0,5 Δp_{DL} [K] : 0,5	n_{CIRC} [-] 2,00	R717

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 $\dot{Q}_E$: 400,0 [kW] $\dot{Q}_C$: 442,3 [kW] $\dot{m}$: 0,340 [kg/s] $\dot{V}_S$: 350,1 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}_{CP}$: 45,5 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 f_Q : 10,0 [%] T_2 : 59,9 [°C] $\dot{Q}_{LOSS}$: 4,55 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 $\dot{Q}_{SL}$: 1123 [W] T_8 : 2,0 [°C] $\Delta T_{SH,SL}$: 1,0 [K]

Calculate | Print | Help | State Points | Auxiliary | COP : 8,790 COP* : 8,815

CYCLE SPECIFICATION

TEMPERATURE LEVELS	PRESSURE LOSSES	QUALITY OUT OF EVAPORATOR	REFRIGERANT
T_E [°C] : 1,0 T_C [°C] : 25,0 ΔT_{SC} [K] : 1,0	Δp_{SL} [K] : 0,5 Δp_{DL} [K] : 0,5	n_{CIRC} [-] 2,00	R717

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 $\dot{Q}_E$: 400,0 [kW] $\dot{Q}_C$: 453,6 [kW] $\dot{m}$: 0,348 [kg/s] $\dot{V}_S$: 357,6 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}_{CP}$: 58,0 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 f_Q : 10,0 [%] T_2 : 74,1 [°C] $\dot{Q}_{LOSS}$: 5,80 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 $\dot{Q}_{SL}$: 1147 [W] T_8 : 2,0 [°C] $\Delta T_{SH,SL}$: 1,0 [K]

Calculate Print Help State Points Auxiliary COP : 6,892 COP* : 6,911

CYCLE SPECIFICATION

TEMPERATURE LEVELS	PRESSURE LOSSES	QUALITY OUT OF EVAPORATOR	REFRIGERANT
T_E [°C] : 1,0 T_C [°C] : 30,0 ΔT_{SC} [K] : 1,0	Δp_{SL} [K] : 0,5 Δp_{DL} [K] : 0,5	n_{CIRC} [-] 2,00	R717

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 $\dot{Q}_E$: 400,0 [kW] $\dot{Q}_C$: 465,4 [kW] $\dot{m}$: 0,355 [kg/s] $\dot{V}_S$: 365,4 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}_{CP}$: 71,1 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 f_Q : 10,0 [%] T_2 : 88,1 [°C] $\dot{Q}_{LOSS}$: 7,11 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 $\dot{Q}_{SL}$: 1172 [W] T_8 : 2,0 [°C] $\Delta T_{SH,SL}$: 1,0 [K]

Calculate Print Help State Points Auxiliary COP : 5,627 COP* : 5,643

CYCLE SPECIFICATION

TEMPERATURE LEVELS	PRESSURE LOSSES	QUALITY OUT OF EVAPORATOR	REFRIGERANT
T_E [°C] : 1,0	Δp_{SL} [K] : 0,5	n_{CIRC} [-] 2,00	R717
T_C [°C] : 35,0 ΔT_{SC} [K] : 1,0	Δp_{DL} [K] : 0,5		

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 $\dot{Q}_E$: 400,0 [kW] $\dot{Q}_C$: 477,6 [kW] $\dot{m}$: 0,363 [kg/s] $\dot{V}_S$: 373,7 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}_{CP}$: 84,7 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 f_Q : 10,0 [%] T_2 : 102,0 [°C] $\dot{Q}_{LOSS}$: 8,47 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 $\dot{Q}_{SL}$: 1199 [W] T_8 : 2,0 [°C] $\Delta T_{SH,SL}$: 1,0 [K]

Calculate Print Help State Points Auxiliary COP : 4,723 COP* : 4,738

CYCLE SPECIFICATION

TEMPERATURE LEVELS	PRESSURE LOSSES	QUALITY OUT OF EVAPORATOR	REFRIGERANT
T_E [°C] : 1,0	Δp_{SL} [K] : 0,5	n_{CIRC} [-] 2,00	R717
T_C [°C] : 17,1 ΔT_{SC} [K] : 1,0	Δp_{DL} [K] : 0,5		

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 $\dot{Q}_E$: 400,0 [kW] $\dot{Q}_C$: 435,9 [kW] $\dot{m}$: 0,336 [kg/s] $\dot{V}_S$: 346,0 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}_{CP}$: 38,5 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 f_Q : 10,0 [%] T_2 : 51,6 [°C] $\dot{Q}_{LOSS}$: 3,85 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 $\dot{Q}_{SL}$: 1110 [W] T_8 : 2,0 [°C] $\Delta T_{SH,SL}$: 1,0 [K]

Calculate Print Help State Points Auxiliary COP : 10,401 COP* : 10,430

CYCLE SPECIFICATION

TEMPERATURE LEVELS

T_E [°C] : 1,0

T_C [°C] : 21,9 ΔT_{SC} [K] : 1,0

PRESSURE LOSSES

Δp_{SL} [K] : 0,5

Δp_{DL} [K] : 0,5

QUALITY OUT OF EVAPORATOR

n_{CIRC} [-] 2,00

REFRIGERANT

R717

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 $\dot{Q}_E$: 400,0 [kW] $\dot{Q}_C$: 446,5 [kW] $\dot{m}$: 0,343 [kg/s] $\dot{V}_S$: 352,9 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}_{CP}$: 50,2 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 f_Q : 10,0 [%] T_2 : 65,3 [°C] $\dot{Q}_{LOSS}$: 5,02 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 $\dot{Q}_{SL}$: 1132 [W] T_8 : 2,0 [°C] $\Delta T_{SH,SL}$: 1,0 [K]

Calculate | Print | Help | State Points | Auxiliary

COP : 7,966   COP* : 7,989

CYCLE SPECIFICATION

TEMPERATURE LEVELS	PRESSURE LOSSES	QUALITY OUT OF EVAPORATOR	REFRIGERANT
T_E [°C] : 1,0 T_C [°C] : 28,1 ΔT_{SC} [K] : 1,0	Δp_{SL} [K] : 0,5 Δp_{DL} [K] : 0,5	n_{CIRC} [-] 2,00	R717

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 $\dot{Q}_E$: 400,0 [kW] $\dot{Q}_C$: 460,9 [kW] $\dot{m}$: 0,352 [kg/s] $\dot{V}_S$: 362,4 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 η_{IS} : 0,700 [-] $\dot{W}_{CP}$: 66,1 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 f_Q : 10,0 [%] T_2 : 82,8 [°C] $\dot{Q}_{LOSS}$: 6,61 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 $\dot{Q}_{SL}$: 1162 [W] T_8 : 2,0 [°C] $\Delta T_{SH,SL}$: 1,0 [K]

Calculate Print Help State Points Auxiliary COP : 6,054 COP* : 6,072

CYCLE SPECIFICATION

TEMPERATURE LEVELS	PRESSURE LOSSES	QUALITY OUT OF EVAPORATOR	REFRIGERANT
T_E [°C] : 1,0	Δp_{SL} [K] : 0,5	n_{CIRC} [-] 2,00	R717
T_C [°C] : 32,2 ΔT_{SC} [K] : 1,0	Δp_{DL} [K] : 0,5		

CYCLE CAPACITY

Cooling capacity $\dot{Q}_E$ [kW] 400 — $\dot{Q}_E$: 400,0 [kW] — $\dot{Q}_C$: 470,8 [kW] — $\dot{m}$: 0,359 [kg/s] — $\dot{V}_S$: 369,0 [m^3/h]

COMPRESSOR PERFORMANCE

Isentropic efficiency η_{IS} [-] 0,7 — η_{IS} : 0,700 [-] — $\dot{W}_{CP}$: 77,1 [kW]

COMPRESSOR HEAT LOSS

Heat loss factor f_Q [%] 10 — f_Q : 10,0 [%] — T_2 : 94,4 [°C] — $\dot{Q}_{LOSS}$: 7,71 [kW]

SUCTION LINE

Unuseful superheat $\Delta T_{SH,SL}$ [K] 1,0 — $\dot{Q}_{SL}$: 1184 [W] — T_8 : 2,0 [°C] — $\Delta T_{SH,SL}$: 1,0 [K]

Calculate | Print | Help | State Points | Auxiliary | COP : 5,189 | COP* : 5,204

$$\varepsilon = 0,25$$

$$\varepsilon = \frac{I_2 - I_1}{I_g - I_1}$$

$$rh = 1$$

$$P1 = 1,013$$

$$I_1 = \mathbf{h}\left[AirH2O\ ; T = T_{wet}\ ; R = rh\ ; P = P1\right]$$

$$t_g = t_k$$

$$I_g = \mathbf{h}\left[AirH2O\ ; T = t_g\ ; R = 1\ ; P = P1\right]$$

$$Q = \dot{m}_{air} \cdot \varepsilon \cdot \left[I_g - I_1\right]$$

$$Q = \dot{m}_{air} \cdot \varepsilon_{new} \cdot \left[I_{g;new} - I_1\right]$$

$$\varepsilon_{new} = 0,32$$

$$T_{k;new} = \mathbf{T}\left[AirH2O\ ; h = I_{g;new}\ ; R = 1\ ; P = P1\right]$$

Parametric Table: Table 1

	T_{wet}	t_k	Q	$T_{k;new}$
Run 1	*3*	*20*	*436*	*17,09*
Run 2	*6*	*25*	*446,5*	*21,9*
Run 3	*20*	*30*	*461*	*28,15*
Run 4	*18*	*35*	*471*	*32,23*

Appendix E

The drawing below presents a simplified version of the refrigeration system at Arla Foods Holstebro. The drawing is provided by Danish Clean Water A/S.