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MODELLING OF SPLIT CONDENSER HEAT PUMP WITH LIMITED SET OF PLATE HEAT EXCHANGER DIMENSIONS

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Abstract:

This paper presents a numerical study of optimal plate dimensions in a split condenser heat pump (SCHP), using ammonia as refrigerant. The SCHP setup differs from a traditional heat pump (THP) setup in the way that two separate water streams on the secondary side of the condenser are heated in parallel to different temperature levels, whereas only one stream is heated in a THP.

The length/width ratio of the plate heat exchangers on the high pressure side of a SCHP was investigated to find the optimal plate dimensions with respect to minimum area of the heat exchangers. The total heat exchanger area was found to decrease with an increasing length/width ratio of the plates. The marginal change in heat exchanger area was shown to be less significant for heat exchangers with high length/width ratios.

In practice only a limited number of plate dimensions are available and feasible in the production. This was investigated to find the practical potential of a SCHP compared to a THP. Using plates optimized for a SCHP in a THP, the total required heat exchanger area increased by approximately 100% for the conditions investigated in this study, indicating that available plate dimensions influence whether a THP or SCHP is beneficial.

Keywords:

Heat Pump, Split Condenser, Plate Heat Exchanger, Ammonia, Vapor Compression, Plate Dimensions

1. Introduction

Heat pumps are widely used in the industry for producing hot water for various industrial processes such as dairies and slaughter houses or for district heating, utility heating in offices, etc. [1, 2, 3, 4, 5]. Due to a long-term goal in the Danish energy policy, where electricity and heat is to be produced without the use of fossil fuels by 2035, heat pumps have also nationally become more and more attractive the recent years. Because of this increasing attractiveness, focus has been put into research on heat pumps to improve COP, pressure drop, production cost, temperature limitations, charge etc.

Some industrial processes require different supply temperatures simultaneously. A split condenser heat pump (SCHP) is able to produce two water streams with different supply temperature simultaneously. Previous studies by Christensen *et al.* [9] compare a SCHP to a traditional heat pump (THP) and show that a SCHP needs less heat exchanger area to produce hot water, keeping the temperatures, COP and total pressure drop on the secondary side constant. The study shows a possible reduction in heat exchange area of 3% for heating water from 40 °C to 85 °C. In the study, an infinite number of plate dimensions with a length/width ratio of 2 was considered. In practice, manufacturers produce a limited set of plate dimensions, so an infinite number of these is not available in the market, and the length/width ratio varies in the available plate dimensions. Due to the practical limitation on the plate dimensions, the SCHP might in practice have more benefits than shown in the study of Christensen *et al.* [9]. Besides a lower manufacturing cost, a reduction in necessary heat exchanger area also has the benefit that it lowers the charge of the system, which is an attractive characteristic when considering future limitations on refrigerants.

Typically, the plate heat exchanger manufacturer buys steel in rolls with a width corresponding to the width of the plate design to minimize material waste. Besides the width and length of the plates, the manufacturer decides on other geometric parameters such as the corrugation angle, the wavelength and the amplitude. This results in different flow patterns and therefore different heat transfer coefficients, influencing the number of plates needed for the heat exchangers in order to transfer the desired heat flow rate. An in-depth understanding of the influence of the different design parameters on the total required heat exchanger area is important in order to optimize plate heat exchangers for different types of heat pump setups. This study presents the influence of varying the length and width of the plate heat exchangers on performance of SCHP and THP.
Ammonia has been chosen as refrigerant for this study. Ammonia is superheated to high temperatures after the compression (due to the slope of entropy curves) which is important to benefit from the SCHP setup. Other refrigerants that does not superheat as much is not expected to benefit from a SCHP setup.
The study is based on a thermodynamic model implemented in EES Engineering Equation Solver [10] which is applied to determine the COP, heat exchanger designs and pressure drop for each system configuration.

2. Methods

2.1. Heat pump layouts

The traditional heat pump (THP), used as a benchmark in this study, was a vapor compression heat pump with two plate heat exchangers on the high-pressure side , one for desuperheating, and one for condensation and subcooling. Ammonia was used as refrigerant. In the THP, one single water stream was heated in the two counter flow plate heat exchangers.

The SCHP was also a vapor compression heat pump with two plate heat exchangers on the high-pressure side and ammonia as refrigerant. In this heat pump setup, two parallel water streams were heated separately in the two counter flow heat exchangers. Principle sketches of both the THP and the SCHP are shown in Figure 1 and Figure 2, respectively.

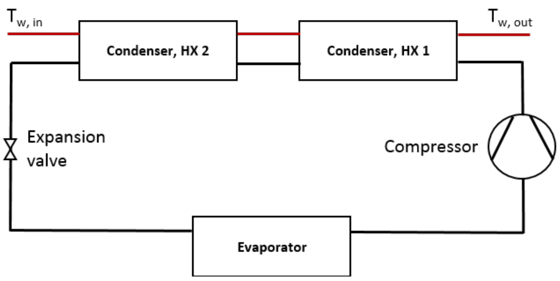

Figure 1: THP setup.

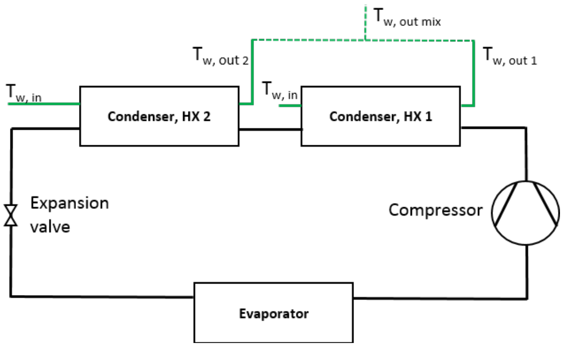

Figure 2: SCHP setup.

Models on the two heat pump setups were built up in EES. The models were based on energy and mass balances for each component. The compressor was modelled with a constant isentropic efficiency. The heat exchangers on the high pressure side were modelled in greater detail as explained below. A detailed description of the model is presented in Christensen et al. [9].

The constant inputs used for the heat pump modelling are shown in Table 1. In addition to these, the configuration was characterized by the distribution of water of the total water mass flow between the two heat exchangers and the refrigerant vapor quality at the split between the heat exchangers.

Concerning the heat exchangers on the high pressure side a symmetric plate pattern was chosen. The design of the plates used in both the THP and the SCHP is illustrated in Figure 3 [7]. The plate dimensions used for modelling are shown in Table 2.

Table 1: Input parameters used for heat pump modelling.

Input parameter	Explanation	Input value	Unit
η_{is}	Isentropic efficiency of vapor compression	0.75	-
T_{evap}	Evaporation temperature	5	°C
$P_{w,in}$	Pressure of water stream	300,000	Pa
ΔT_{sh}	Superheat before compressor	2	°C
$\Delta T_{pinch,sc,\ HX2}$	Temperature difference between water stream inlet and refrigerant outlet in HX2	5	°C
$T_{w,in}$	Temperature of water stream into HX2, THP	40	°C
$T_{w,out}$	Temperature of water stream out of HX1, THP	85	°C
$\dot{m}_{water,\ tot}$	Total mass flow rate of water stream	1	kg/s
$T_{w,in,HX2}$	Temperature of water stream into heat exchanger 2, SCHP	40	°C
$T_{w,in,HX1}$	Temperature of water stream into heat exchanger 1, SCHP	40	°C
$T_{w,out,mix}$	Temperature of the mixed water stream out of heat exchanger 2 and heat exchanger 1, SCHP	85	°C
$\dot{Q}_{heat}$	Heat capacity of the heat pump	188,344	W

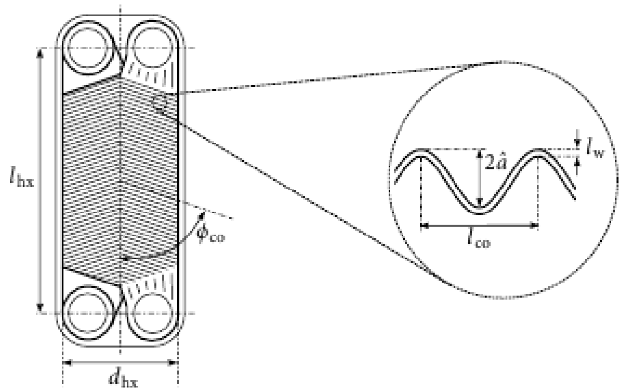

Figure 3: The design of plates in a plate heat exchanger.

Table 2: Input parameters for plates in plate heat exchanger used in both the traditional and split condenser heat pump setup.

Input parameter, HX	Explanation	Input value	Unit
l_w	Thickness of plates in HX 1+2	0.001	m
ϕ_{co}	Corrugation angle	60	°
$2\hat{a}$	Corrugation height	0.0018	m
l_{co}	Corrugation wavelength	0.0080	m

In the model of both the THP and the SCHP, the heat exchangers on the high pressure side were discretized for improved accuracy of the models as presented by Christensen *et al.* [6]. The heat transfer coefficient of single phase flow was calculated using the correlation presented by Martin [7]. The heat transfer coefficient in two phase flow was calculated using the correlation by Y. Yan [8].

In Christensen *et al.* [9] the model was used in order to calculate COP and necessary heat transfer areas of both heat pump setups using plate heat exchangers without limitations on the dimensions. Input values similar to the values presented in Table 1 were used for that study. The COP of the optimized THP found in Christensen *et al.* [9] is in this study referred to as COP_{THP}* and the total heat exchanger area of the same THP is referred to as A_{THP}*.

2.2. Length/Width ratio

The width of the plates, the amplitude of waves and the number of plates determines the mass flow rate through each channel in a plate heat exchanger. The mass flow rate through each channel has influence on both the heat transfer coefficient and the pressure drop. The mass flux and the length of the plates determines the pressure drop [7, 8].

A different length/width ratio will have an influence on the mass flow through each channel and therefore an influence on the total number of plates needed in order to ensure a limited pressure drop on the secondary side. The pressure drop limitation on the secondary side was chosen to be 50 kPa.

"A split condition" refers to the quality of refrigerant between HX1 and HX2 together with the mass flow rate through each heat exchanger on the secondary side. "The optimal split condition" refers to the split condition where the smallest heat exchanger area is found when keeping temperatures, COP and pressure drop constant. Christensen *et al.* [9] found that the optimal split condition for a SCHP heating water from 40 °C to 85 °C was $\dot{m}_{HX2}/\dot{m}_{total} = 0.6$ and $x_{split} = 0.55$ for a total pressure drop on the secondary side of 50 kPa and a length/width ratio of the plates of 2.

In the current study the influence of a different length/width ratio was investigated for three alternative split conditions, which were compared to the THP reference case. This analysis were based on the four configurations listed in Table 3. For each of these the optimal plate dimensions were determined.

Table 3: Split conditions with respect to length/width ratio.

	$\dot{m}_{\mathrm{HX2}}/\dot{m}_{\mathrm{total}}$	$\dot{m}_{\mathrm{HX1}}/\dot{m}_{\mathrm{total}}$	x_{split}
THP	1	1	1
SCHP1	0.48	0.52	0.4
SCHP2	0.6	0.4	0.55
SCHP3	0.76	0.24	0.75

The length and width of the plates in both HX2 and HX1 were set as free parameters where only the length/width ratio was defined. The length/width ratio was varied to find the total heat exchanger area, which was minimized to find the optimal length and width for possible length/width ratios, such that the COP of the heat pump was equal to COP_{THP}*.

This investigation was done in three steps:

- The length/width ratio was equal in the two heat exchangers at values from 1 to 5.
- The length/width ratio was 3 for HX1 and was varied for HX2 at values from 1 to 5.
- The length/width ratio was 3 for HX2 and was varied for HX1 at values from 1 to 5.

2.3. Limited number of plate dimensions

Manufacturers of plate heat exchangers have a limited number of plate dimensions available when designing heat exchangers. This might have influence on how the SCHP would perform compared to a THP, depending on the plate dimensions available in the production and the design conditions of the heat pumps.

The plate dimensions optimized for SCHP2 (see Table 3) were chosen as possible plate dimensions. The two plate dimensions had a width of d_{HX}=0.275m and d_{HX}=0.362m, both with a length/width ratio of l_{HX}/d_{HX}=3. When having two different plate dimensions, four different combinations are possible. These combinations are listed in Table 4.

Table 4: Options for combination of two different plate dimensions for HX2 and HX1.

	L_{HX2} [m]	d_{HX2} [m]	L_{HX1} [m]	d_{HX1} [m]
HX_opt1	0.825	0.275	1.086	0.362
HX_opt2	0.825	0.275	0.825	0.275
HX_opt3	1.086	0.362	1.086	0.362
HX_opt4	1.086	0.362	0.825	0.275

The performance of the THP was investigated for each different combination of plate dimensions. In addition the total pressure drop on the secondary side was varied. This influences the total heat exchanger area, which influenced the outlet temperature of the water.

2.4. Parameter variation

The parameters describing the heat exchanger dimensions were investigated in order to determine the influence on the total heat exchanger area and the total pressure drop on the secondary side. The benchmark chosen for comparison was the plate dimensions optimized for SCHP2 (Table 3), HX_opt1 from Table 4 and the dimensions described in Table 2. The plate dimensions were varied from a lower limit to an upper limit. The limits and benchmark point are listed in Table 5.

Table 5: Lower limit, benchmark and upper limit for parameters in plate heat exchangers.

	Explanation	Lower limit	Reference	Upper limit
l_w	Plate thickness	0,0002 m	0,001 m	0,0025 m
ϕ_{co}	Corrugation angle	45°	60°	75°
$\hat{a}$	Corrugation height	0,0006 m	0,0009 m	0,0025 m
l_{co}	Corrugation wavelength	0,004 m	0,008 m	0,02 m
α_{HM}	Single-phase heat transfer coefficient	50%	100%	150%
α_{YAN}	Two-phase heat transfer coefficient	50%	100%	150%

The interval from lower limit to upper limit for each parameter corresponds to a 100% change in the individual parameter. For example the benchmark for the corrugation angle (60°) is the set point corresponding to the value 0 and the lower limit corresponds to -50% and the upper limit corresponds to +50% (a total variation interval of 100%). The total heat exchanger area and the total pressure drop on the secondary side were modelled for all values of the individual parameter. The changes in area and pressure drop were calculated as a percent change with respect to the area and pressure drop of the reference.

3. Results

3.1. Length/width ratio

A relative heat exchanger area is defined as the necessary heat exchanger area divided by the heat exchanger area needed for the optimized THP, A_{THP}*. The relative total area of the heat exchangers on the high pressure side of the heat pump, when the length/width ratios in both heat exchangers were equal, is shown in Figure 4 as a function of the length/width ratio. The total heat exchanger area in both the THP and the SCHP was decreasing with the length/width ratio. The marginal reduction in total heat exchanger area became smaller with a higher length/width ratio for all split conditions. The split condition of SCHP2 (see Table 3) needed the smallest total heat exchanger area when the length/width ratio of the two heat exchangers was higher than 1.4. This corresponds to the findings in Christensen *et al.* [9], where this split condition was the optimal split for a length/width ratio of 2.

The marginal change in total heat exchanger area became less significant when the length/width ratio was above 3. Accordingly, the physical conditions may have an influence on the chosen plate dimensions rather than the length/width ratio, if it is more convenient to have a less tall heat exchanger in the heat pump unit.

The change in necessary area of both heat exchangers, when only the length/width ratio of one heat exchanger was varied, is shown in Figure 5 and Figure 6.

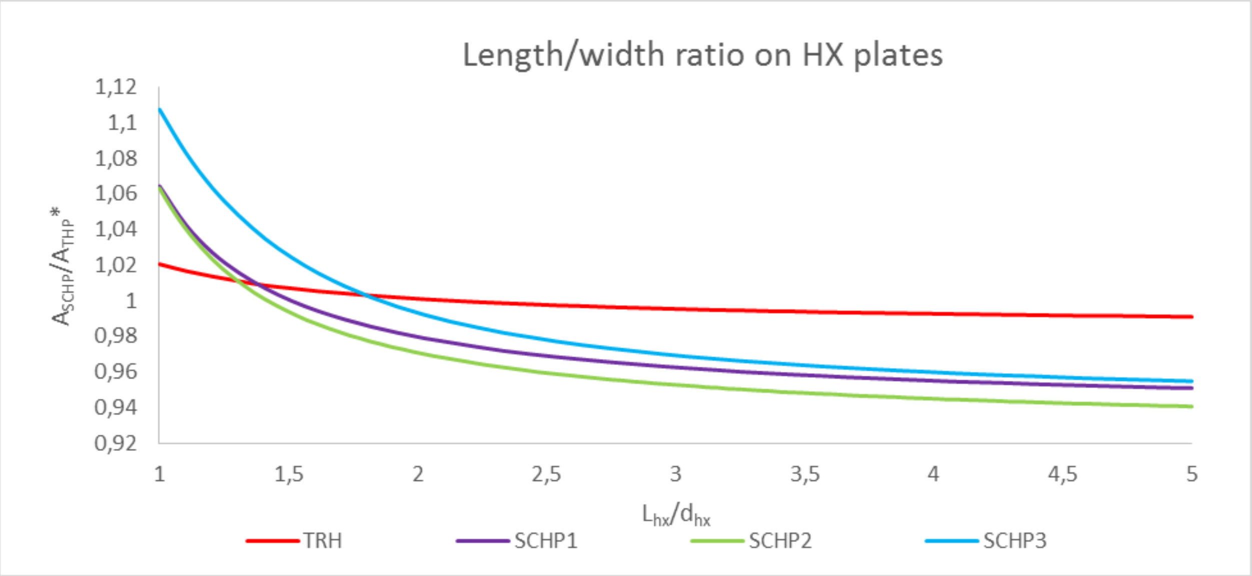

Figure 4: Relative area of plate heat exchangers on high pressure side of the heat pump as a function of the length/width ratio of the plates. Length/width ratio of plates is the same in both heat exchangers. The red curve corresponds to the traditional heat pump. SCHP1-3 corresponds to different plate dimensions for the split condenser heat pump.

The influence of changing the length/width ratio of HX1 (Figure 5) is approximately the same as changing the length/width ratio of HX2 (Figure 6), when the length/width ratio is above 2.

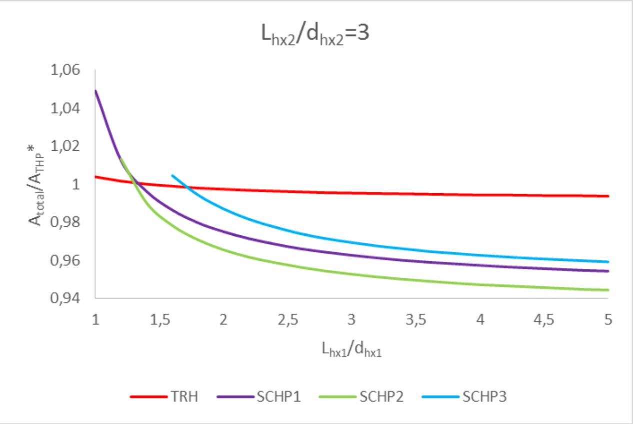

Figure 6: Relative area of heat exchangers as a function of length/width ratio of HX1, when length/width ratio of HX2 is 3..

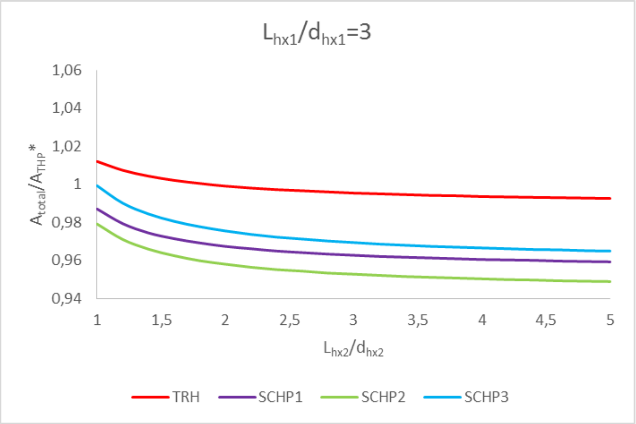

Figure 5: Relative area of heat exchangers as a function of length/width ratio of HX2, when length/width ratio of HX1 is 3..

3.2. Limited number of plate dimensions

When choosing a length/width ratio of 3 for both heat exchangers, the optimal dimensions for heating water from 40°C to 85°C is shown in Table 6. The optimal plate dimensions are shown for the THP and for the three different split conditions mentioned in Table 3.

Table 6: Optimal dimensions for different splits, when length/width ratio for both heat exchangers is 3.

	L_{HX2} [m]	d_{HX2} [m]	L_{HX1} [m]	d_{HX1} [m]
THP	0.720	0.240	0.423	0.141
SCHP1	0.762	0.254	1.101	0.367
SCHP2	0.825	0.275	1.086	0.362
SCHP3	0.903	0.301	1.167	0.389

The performance of the THP is illustrated in Figure 7 and Figure 8 for the four different combinations of plate dimensions mentioned in Table 4. The two different plate dimensions (length and width for HX1 and HX2) used in Table 4 are the ones optimized for SCHP2 in Table 6.

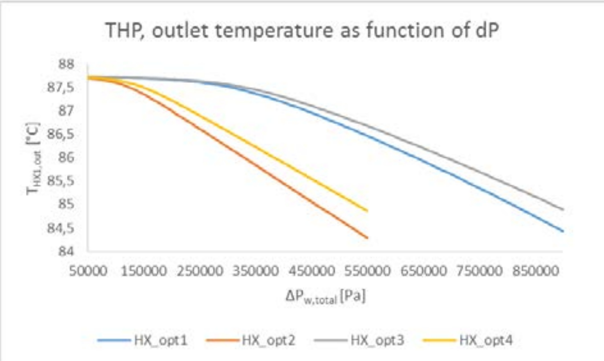

Figure 7: Outlet temperature of water as a function of the total pressure drop on the secondary side of the heat exchangers.

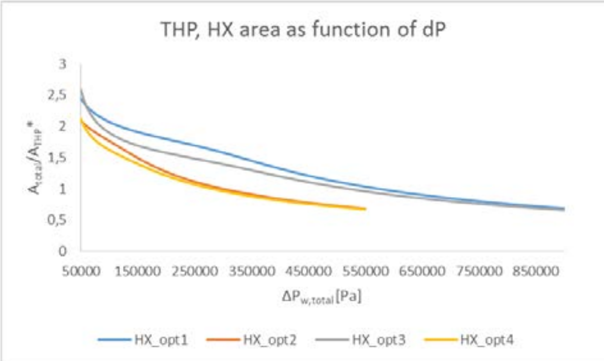

Figure 8: Relative area of heat exchangers as a function of the total pressure drop on the secondary side of the heat exchangers.

Figure 7 indicates that it is not possible for a THP to deliver the same water temperatures and still keep the same pressure limit if the plates are not exactly dimensioned for this. If the pressure limit of 50,000 Pa is to be kept, the total area of the two heat exchangers has to be significantly higher than for SCHP2. The high area results in a slightly higher outlet temperature of the water. The outlet water temperature will become approximately 87.5°C with a total heat exchanger area aproximately 100% higher than SCHP2 if the mass flow rate of the water is kept constant. To control the outlet temperature a bypass of water would need to be installed.

Figure 7 and Figure 8 indicate, that HX_opt2 and HX_opt4 have an advantage with a lower pressure drop on the secondary side than HX_opt1 and HX_opt3. HX_opt2 and HX_opt4 both have the smallest possible dimensions in HX1. Table 6 also indicates that the optimal length and width for HX1 in the THP is significantly smaller than the omtimal length and width for SCHP2.

3.3. Parameter variation

The variation of parameters in Table 5 is illustrated in Figure 9 and Figure 10, showing the percent change in total heat exchanger area and total pressure drop on the secondary side respectively. For all parameters an increase in total heat exchanger area results in a decrease in total pressure drop and vice versa. The parameters having the highest influence on the total heat exchanger area are the amplitude of the waves and the uncertainty of the heat transfer coefficient from the Holger Martin correlation in [7], whereas other parameters have minor impact. The marginal increase in total heat exchanger area from an increased amplitude is almost constant, whereas the marginal decrease in pressure drop decreases with an increase in amplitude. An increase of the amplitude of 25% (of the interval from lower limit to upper limit) will increase the total heat exchanger area with 35% for change in HX1 and 30% for change in HX2. The same change in amplitude results in a decrease in total pressure drop of -43% for a change in amplitude in HX1 and -40% for a change in amplitude in HX2. With respect to the total pressure drop the most significant parameters are the amplitude, the corrugation angle and the uncertainty of the Holger Martin correlation. The wavelength is significant if it is decreased.

Figure 9 and Figure 10 indicate that the dimensions in the plates of the heat exchangers and the uncertainty of the correlations used for calculation the heat transfer coefficients have a significant influence on both the total heat exchanger area and the total pressure drop. Therefore, the dimensions chosen for the heat exchangers will have a significant influence on the performance of the heat pump, and a more thorough investigation of the dimensions is key to optimize both the SCHP and the THP.

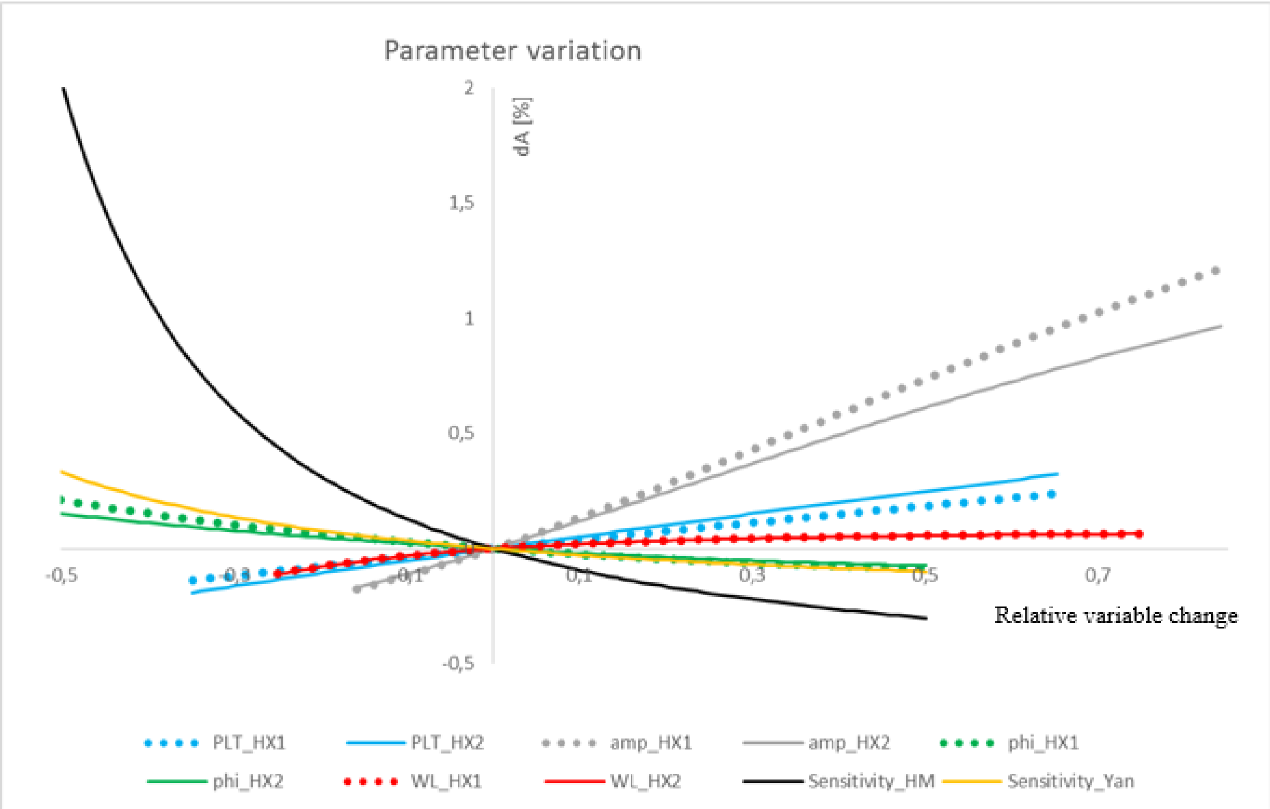

Figure 9: Parameter variation of plate dimensions showing the percent change in total heat exchanger area of the SCHP with respect to the relative change from in reference point.

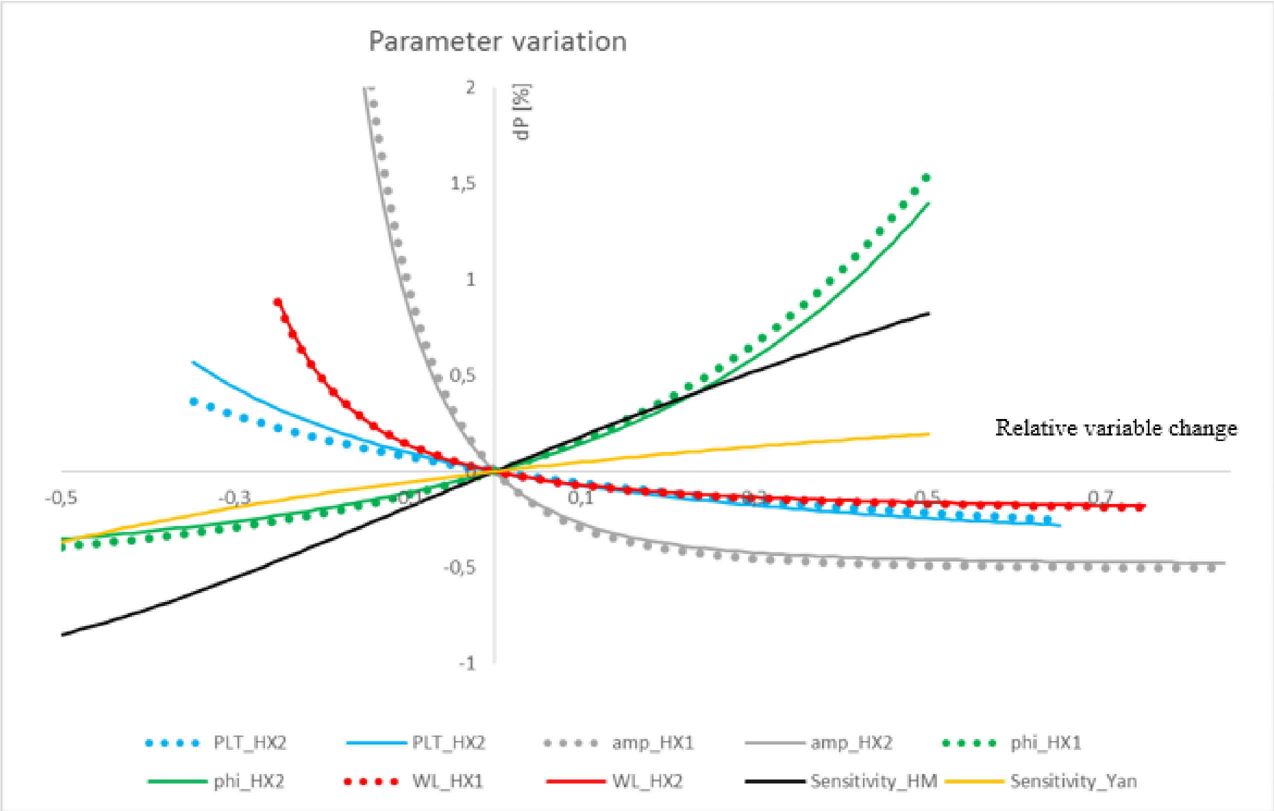

Figure 10: Parameter variation of plate dimensions showing the percent change in total pressure drop on the secondary side of the SCHP with respect to the relative change from the reference point.

4. Discussion

In this study the length/width ratio of the plates in plate heat exchangers was analyzed to find the total heat exchanger area and the total pressure drop on the secondary side. These parameters have high importance when designing heat pumps. A more compact heat exchanger might not be cheaper in production cost, as the production method might vary from manufacturer to manufacturer.

The influence from change in parameters were modelled using correlations for heat transfer coefficients and friction factors from the literature [7, 8]. The correlations come with high uncertainties as most correlations are made from experimental setups and converted into more general models. The correlations might not be precise if the dimensions modelled vary from the experiments performed to make the correlation. Furthermore, the correlation of Yan *et al.* [8] only describes a corrugation angle of 60°, which adds uncertainty to the parameter variation of the corrugation angle. However, the sensitivity analysis showed that the influence from uncertainties from the correlation of Yan *et al.* [8] do not show big influences compared to other parameters. Still, the uncertainty of the correlation seems to be significant.

The model is not limited toan integer number of plates. Practically, the number of plates needs to be rounded up to nearest whole number of plates, which will have influence on both the total heat exchanger area and the total pressure drop. Depending on the length and width of the plates a round up in number of plates might be a significant change.

5. Conclusion

From the numerical investigation on the split condenser heat pump (SCHP) with limited plate dimensions presented in this paper, the following conclusions can be drawn: The total heat exchanger area in both the traditional heat pump (THP) and the SCHP is decreasing with the length/width ratio.

The marginal reduction in total heat exchanger area becomes smaller with a higher length/width ratio for all split conditions in the SCHP. The marginal change in total heat exchanger area becomes less significant when the length/width ratio is above 3.

It was not possible for the THP to deliver the same water temperatures as the SCHP without exceeding the the pressure drop limit, when plate dimensions were designed for a SCHP. To keep the same pressure drop on the secondary side the total heat exchanger area had to be increased significantly, aproximately 100% for the case investigated in this study.

A change in the design parameters of the plates in a plate heat exchanger had significant influence on both the total heat exchanger area needed and the total pressure drop on the secondary side. The amplitude of the waves and the accuracy of the correlation of Holger Martin [7] showed the biggest influence on the total heat exchanger area. Furthermore, the corrugation angle, the amplitude of the waves and the friction coefficient calculated by the correlation of Holger Martin [7] showed the biggest influence on the total pressure drop on the secondary side of the heat exchangers. The wave length and the plate thickness showed the least influence on both the total heat exchanger area and the total pressure drop. However, the influence was still significant even though these parameters showed the least influence of the parameters chosen to investigate.

This study shows the importance of optimizing plate dimensions for the specific conditions that the heat pump is to operate at.

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Nomenclature

Parameter	Explanation	Unit
A	Heat transfer area	m^2
A_{THP}*	Total heat transfer area of traditional heat pump from Christensen *et al.* [9]	m^2
$\hat{a}$	Amplitude of waves in plate heat exchanger	M
α_{HM}	Local heat transfer coefficient from Holger Martin correlation [7]	W/(m^2 K)
α_{HM}	Local heat transfer coefficient from Yan *et al.* correlation [7]	W/(m^2 K)
amp	Amplitude of waves in plate heat exchanger	m
COP_{SCHP}	Coefficient Of Performance for SCHP system	-
COP_{THP}	Coefficient Of Performance for THP system	-
COP_{THP}*	Coefficient of performance of traditional heat pump from Christensen *et al.* [9]	-
d_{hx}	Width of plate in plate heat exchanger	m
ΔP	Pressure difference/pressure drop	Pa
ΔT	Arithmetic temperature difference	°C
η_{is}	Isentropic efficiency of vapor compression	-
l_{co}	Corrugation wavelength in plate heat exchanger	m
l_{hx}	Length of plate in plate heat exchanger	m
l_w	Plate thickness in heat exchangers	m
$\dot{m}$	Mass flow rate	kg/s
P	Pressure	Pa

PLT	Plate Thickness	m
ϕ_{co}	Corrugation angle of waves in plate heat exchanger	°
$\dot{Q}$	Heat transfer rate	W
SCHP	Split Condenser Heat Pump	-
T	Temperature	°C
THP	Traditional Heat Pump	-
X	Vapor quality	-
WL	Corrugation wavelength in plate heat exchanger	m

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