DTU Library

Comparison of Heat Pump Design and Performance for Modern Modern Refrigerants

Nielsen, Stefan; Christensen, Stefan Wuust; S. Thorsen, Richard; Elmegaard, Brian

Published in:
Proceedings of the13th IIR-Gustav Lorentzen Conference on Natural Refrigerants

Link to article, DOI:
10.18462/iir.gl.2018.1149

Publication date:
2018

Document Version
Publisher's PDF, also known as Version of record

PAPER ID 1149
DOI: 10.18462/iir.gl.2018.1149

COMPARISON OF HEAT PUMP DESIGN AND PERFORMANCE FOR MODERN REFRIGERANTS

Stefan Nielsen[a], Stefan Wuust Christensen[a], Richard S. Thorsen[b], Brian Elmegaard[a]

[a] Department of Mechanical Engineering, Technical University of Denmark

Nils Koppels Allé 403, Kgs. Lyngby, DK-2800, Denmark

[b] Department of Mechanical and Aerospace Engineering, New York University Tandon School of Engineering

6 MetroTech Center, Brooklyn, NY-11201, United State of America

ABSTRACT

Due to increasing awareness of global warming, the types of refrigerants used in heat pumps are changing globally. Regulations for HFC refrigerants are being introduced due to their high global warming potential (GWP). This can create a shift in demand for different refrigerants since HFCs are still commonly used in many countries. As a result, the refrigerant charge will play a significant role when determining the most feasible refrigerant.

This paper presents a numerical study of the performance of natural, HFC, and HFO refrigerants for a one-stage cycle and focuses on the refrigerant charge influence. The study shows that R717 is the most optimal refrigerant, exhibiting a 51% to 87% smaller charge and 12% to 27% lower cost of heat compared to other refrigerants. In addition, the results show that the refrigerant price should be included when conducting economic evaluations.

Keywords: industrial heat pumps, refrigerant charge, modern refrigerants, economic evaluation

1. INTRODUCTION

An aim of today's modern energy solutions is to be less dependent on fossil fuels and to increase energy efficiency. Implementation of heat pumps (HP) has been suggested in numerous studies as a technology that has the potential to help accomplish this aim(Werner, 2017; Münster et al., 2012; Petrović and Karlsson, 2016). HPs have the benefit of being able to produce heat based on electricity from renewable energy sources to provide steady and efficient heat production. However, the efficiency of HPs depends on the properties of the working fluid/refrigerant. In addition, the optimal refrigerant relies on many factors: the thermophysical properties, toxicity, flammability, compatibility with materials, price, impact on the environment and legal restrictions.

Legal restrictions on certain refrigerants were introduced by the Montreal Protocol (Secretariat, 1987) in order to curb for their negative environmental impacts and soon the Kigali amendment (Secretariat, 2016) will be implemented. However, in countries with large markets such as China (Danfoss, 2017), HFC refrigerants are still commonly used for industrial HPs. If strict worldwide regulations are to be implemented, there can be substantial impacts on the way heat pumps are designed and thus the allowed refrigerant charge will then determine the size of the HP.

This paper investigates the efficiency and the cost of heat for different refrigerants within a given capacity with a focus on refrigerant charge. Through a numerical study constructed in the software EES (EES, 2016), the total charge and performance can be calculated for various of refrigerants.

DOI: 10.18462/iir.gl.2018.1149

2. METHODS

Calculation of the HP was carried out using a single stage cycle with a direct expansion evaporator and minimum component requirements. The cycle consists of a compressor, condenser, evaporator, and throttle valve. A principle sketch of the chosen vapor compression system is presented in Fig. 1 with an illustration of the nomenclature used. A pinch point temperature difference was used to find the temperature for the working fluid, the source, and the sink. Fig. 1 illustrates the heat exchange between the working fluid and the secondary system fluid.

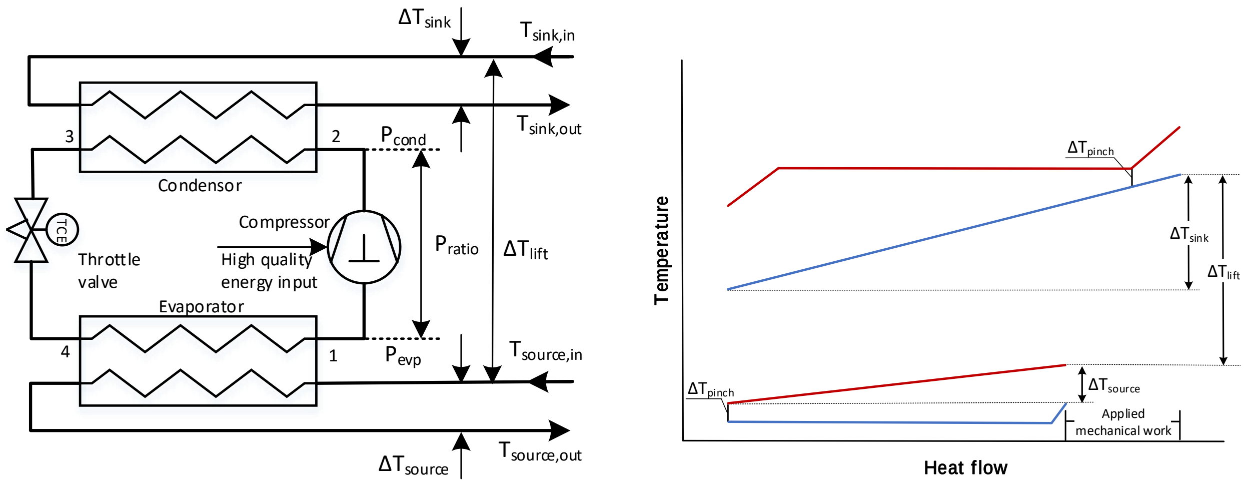


Figure 1. (left) Principle sketch of the vapour compression cycle
(right) Q,T - Diagram of a subcritical cycle

The performance of the HP was based on estimated values for the component parameters, see Table 1. The design criteria were estimated for a low-temperature solution for the future district heating system that is further described in (Lund et al., 2014). The values were estimated using a reciprocating piston compressor and a chevron plate heat exchanger constructed by SWEP that can handle a high variation of refrigerants.

Table 1. Used parameters to evaluate the performance of the heat pump

Type of data	Value	Unit	Description
Capacity	150	kW	Sink heat transfer rate
Fluid	H_2O	-	Secondary fluid
Efficiency	0.8	-	Compressor isentropic efficiency
-	0.8	-	Compressor volumetric efficiency
-	0.95	-	Electrical motor efficiency
Temperature	60	$^{\circ}C$	Sink outlet temperature
-	15	$^{\circ}C$	Source inlet temperature
Temperature lift	35	K	Sink lift
-	10	K	Source lift
Temperature difference	5	K	Pinch point in Evaporator and Condenser
-	5	K	Superheat and Subcooling

2.1. Calculation of refrigerant charge

The calculation of charge for each component was based on an estimation of its minimum volume and the fluid density. It was assumed that a two-phase flow occurs in the condenser and evaporator and single-phase flow occurs in the other components. It was also assumed that the throttle valve was connected directly to the evaporator. The total refrigerant charge, M_{total}, was defined by Eq. (1).

$$M_{\text{total}} = M_{\text{cond}} + M_{\text{evp}} + M_{\text{comp}} + M_{\text{pipe,dis}} + M_{\text{pipe,liq}} + M_{\text{pipe,suc}} \quad (1)$$

DOI: 10.18462/iir.gl.2018.1149

The pipe volume was dependent on the location of each main component and an estimated minimum velocity to ensure oil return to the compressor (Guntoft and Lauritsen, 2003). 2 meter pipes between each component were included and thus the pipe charge was given by Eq. (2).

$$M_{\text{pipe}} = \rho_{\text{ref}} \cdot V_{\text{pipe}} = \rho_{\text{ref}} \cdot \left(\frac{\dot{m} \cdot \nu}{U} L \right) \tag{2}$$

Where ρ was the density, $\dot{m}$ the mass flow rate, ν the specific volume, U minimum velocity, and L the pipe length. The compressor volume was determined by Eq.(3) based on the frequency, the suction volume flow,$\dot{V}_{comp,in}$, and the volumetric efficiency, η_v. The frequency was assumed to be 1400 rpm. The refrigerant in the oil chamber was neglected.

$$M_{\text{comp}} = \rho_{\text{comp}} \cdot V_{\text{comp}} = \frac{\rho_{\text{in}} + \rho_{\text{out}}}{2} \cdot \left[\left(\frac{1400}{60} \right)^{-1} \cdot \dot{V}_{\text{comp,in}} \cdot \eta_v \right] \tag{3}$$

To determine the charge for the condenser and evaporator, the ratio of liquid and vapor must be known for each location inside the heat exchanger. This was established by discretizing the plate length equidistantly based on the enthalpy difference. For each control volume, the required volume and average density were identified. The total charge for the condenser and evaporator were then defined by Eq. (4).

$$M = \sum_{\text{CVs}} M_i = \sum_{\text{CVs}} \overline{\rho_i} \cdot V_i = \sum_{\text{CVs}} (\alpha_i \rho_g + (1 - \alpha_i) \rho_f) \cdot \left(\frac{A_{s,i}}{2} \cdot b \right) \tag{4}$$

In which α was the void fraction, A_s the heat exchangers surface area, and b was the plate width. The required heat exchanger area was calculated based on correlations for plate heat exchangers. These were chosen based on the literature review of available correlations for two-phase flow conducted in Eldeeb et al. (2016). The correlations used are shown in table 2. Poggi et al. (2008) provided a literature review that suggested Hughmark (1962) to be an accurate correlation for a broad variety of flow types, which was further confirmed in Woldesemayat and Ghajar (2007).

Table 2. Plate heat transfer correlations applied

Phase	Media	Correlation
Liquid, Vapour	H_2O, Rxxx	Martin (1996)
Two-phase condensation	Rxxx	Yan et al. (1999)
Two-phase evaporation	Rxxx	Amalfi et al. (2016)

2.2. Economic evaluation

An economic assessment of the design was conducted based on both investment and operating costs. This assessment was based on the following assumptions

- The heat source was pre-installed and freely available. Hence, it was assumed to have no cost.
- The interest rate was 7 % and the inflation rate 2 %. The annual operation time in hours was set to 3500 hours. The plant lifetime was 15 years
- The Purchased Equipment Cost (PEC) for a reciprocal compressor was dependent on the swept volume. In addition, the PEC of the electrical motor was dependent on the shaft power and a fixed electrical motor efficiency
- The PEC for the heat exchanger was dependent on the heat transfer area
- The price for the refrigerant and taxes was based on the Danish market (AGA, 2018; SKAT, 2018)

DOI: 10.18462/iir.gl.2018.1149

- The total investment cost was 4.16 times the total PEC of the main components. This accounts for the additional costs of building a new system at an existing facility (Bejan et al., 1996)
- The operation cost and maintenance costs was assumed to be a fix factor of 20 % of the capital investment cost (Ommen et al., 2014)

The electricity price was based on the Danish market (Energistyrelsen, 2015) and was correlated with the cost for industrial consumers. The PEC for the components was based on prices from Ommen et al. (2014).

An Exergoeconomic cost balance was used to validate the cost of heat production, where the quality of the heat consumed and produced were taken into account. An economic and exergy balance can be derived by combining exergy streams and direct cost (investment, operation, and maintenance), which was further discussed in Bejan et al. (1996). The ambient temperature and pressure were assumed 20 $^{\circ}C$ and 1 atm, respectively.

2.3. Working fluid

Five different working fluids (HFC, Natural, and HFO) were chosen for the analysis due to their availability, fluid properties, and safety regulations as presented in Table 3.

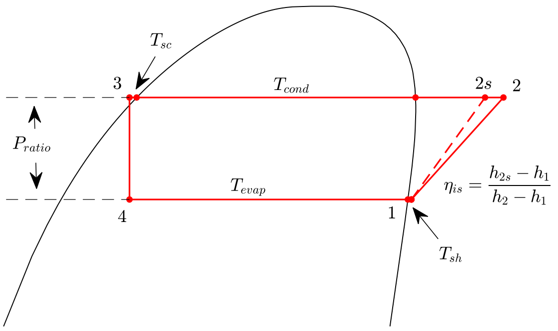
Even with HFCs being phased out due to their high GWP, they are still being widely used because of their low safety requirements. R134a and R410A were included in this analysis since they are commonly used HFC refrigerants for heat pumps. Natural refrigerants have gained parity with HFC refrigerants and are frequently used as working fluids for industrial heat pumps. Commonly used natural refrigerants are ammonia (R717), propane (R290) and carbon dioxide (R744). R744 will not be simulated in this context due to the significantly different design criteria and related economic assumptions. In addition, the HFO refrigerant R1234yf was included as it is expected to be a potential replacement for R134a (Ansari et al., 2013). The benefit of natural refrigerants and HFOs are their low GWP, as presented in Table 3.

Table 3. GWP and ASHRAE classification, (Linde, 2017)

Refrigerant	*Group*	*GWP 100 [years]*	*ASHRAE Safety Group*	*ASHRAE Flammability*	*ASHRAE Toxicity*
R717 (Ammonia)	*Natural*	*0*	*B2L*	*Yes (Low)*	*Yes*
R290 (Porpane)	*Natural*	*3*	*A3*	*Yes (Highly flammable)*	*No*
R1234yf	*HFO*	*4*	*A2L*	*Yes (Low)*	*No*
R134a	*HFC*	*1430*	*A1*	*No*	*No*
R410A	*HFC*	*2088*	*A1*	*No*	*No*

3. RESULTS

The HP system was designed with the estimated parameters given in Table 1, where Fig. 2 illustrates the HP systems' state points and performance data for the refrigerant R717.



State point	*T [°C]*	*P [kPa]*	*h [kJ · kg⁻¹]*	*ρ [kg · m⁻³]*
1	*5*	*430*	*1475*	*3.4*
2	*163*	*2415*	*1810*	*12.1*
3	*52*	*2415*	*449*	*560.2*
4	*0*	*430*	*449*	*17.1*

T_{cond} = 57°C T_{evp}=0°C COP=3.1 P_{ratio}=5.6

Figure 2 & Table 4. R717 state point and performance data

DOI: 10.18462/iir.gl.2018.1149

A comparison of the refrigerant performance was based on exergy efficiency of the system, the total charge, Coefficient of Performance (COP), and cost of heat is presented in Table 5. The data shows that R717 has the best performance. This is reflected in its low exergy destruction, high exergy efficiency for the system, high COP, low cost of heat, which all relate mostly to the performance of the system. In addition, the ammonia cycle is shown to have the lowest charge. The economic benefits of R717 are further demonstrated in Ommen et al. (2014), which shows R717 as the ideal refrigerant and that the components set the limitations.

Table 5. Comparison of the total exergy destruction and efficiency, total charge, COP, and cost of heat

Parameter	*Unit*	*R717*	*R290*	*R134a*	*R410A*	*R1234yf*
Exergy destruction	$[kW_{ex}]$	*22.3*	*27.6*	*26.9*	*25.2*	*31.7*
Exergy efficiency	*[-]*	*0.67*	*0.64*	*0.64*	*0.65*	*0.62*
COP	*[-]*	*3.1*	*2.6*	*2.6*	*2.8*	*2.3*
Total Charge	*[kg]*	*3.5*	*7.1*	*15.9*	*27.1*	*20.3*
Specific Heat Charge	$[g \cdot kW^{-1}]$	*23*	*47*	*106*	*181*	*135*
Cost of heat	$[USD \cdot MWh^{-1}]$	*44*	*51*	*55*	*50*	*60*

Table 5 further indicates that R290 requires a considerably lower charge than HFCs. However, R290 is flammable which increases the safety requirements and furthermore the cost of heat is not competitive with the non-natural alternatives. The possible replacement refrigerant for HFCs, R1234yf, has the lowest exergy efficiency and the second highest cost of heat, which indicates that further improvements are required for it to be selected for future HPs. The HFC refrigerant R410A shows a lower cost of heat than R134a. However, R134a has a lower charge and a lower GWP which results in a smaller impact on the environment compared to R410A.

By changing the key parameters in a sensitivity analysis, it is possible to identify the parameters that have a significant influence on the cost of heat. The parameter variation is shown in Fig. 3 for R134a since it is a common refrigerant for HPs. As expected from Ommen et al. (2014) and Fig. 4, the cost of heat is sensitive to the economic parameters, isentropic efficiency, and volumetric efficiency of the compressor. The figure also illustrates that increasing the subcooling has a positive effect on the cost of heat, indicating that the price of the extra refrigerant is inconsequential when accounting for the HPs increase in efficiency.

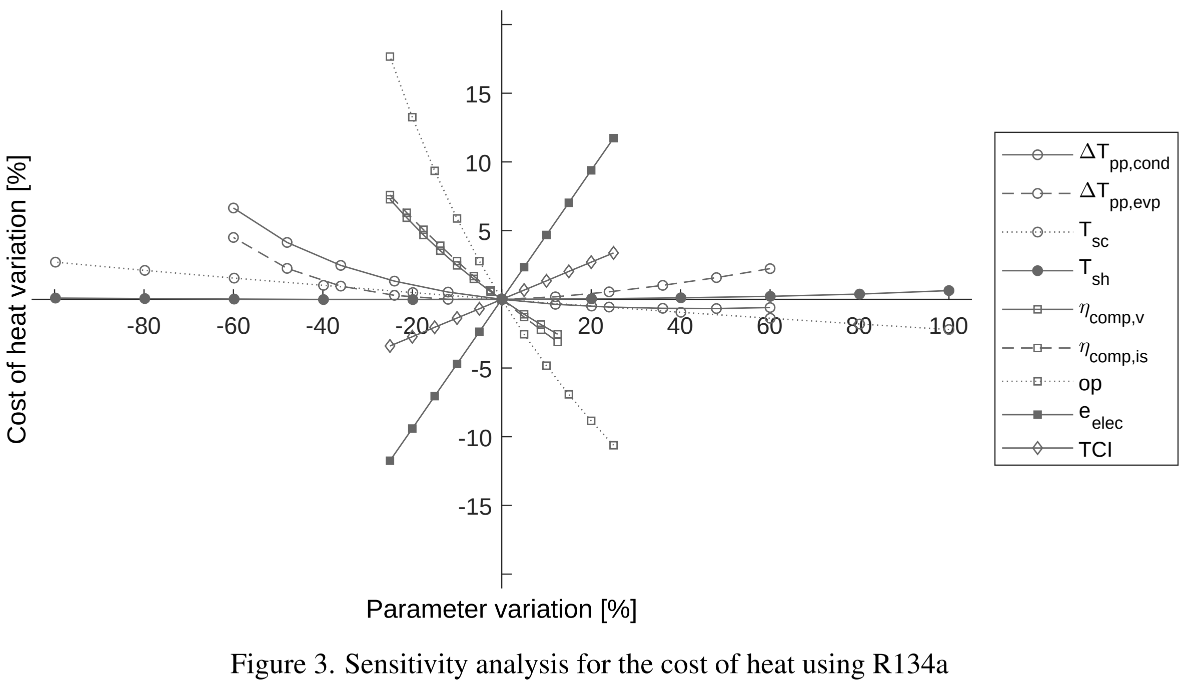


Figure 3. Sensitivity analysis for the cost of heat using R134a

DOI: 10.18462/iir.gl.2018.1149

As Fig.3 illustrates, the cost of heat is highly dependent on the economic evaluation of the total capital investment (TCI). The TCI and the equipment cost of the HP system is analyzed and showed in Fig. 4 for the different refrigerants.

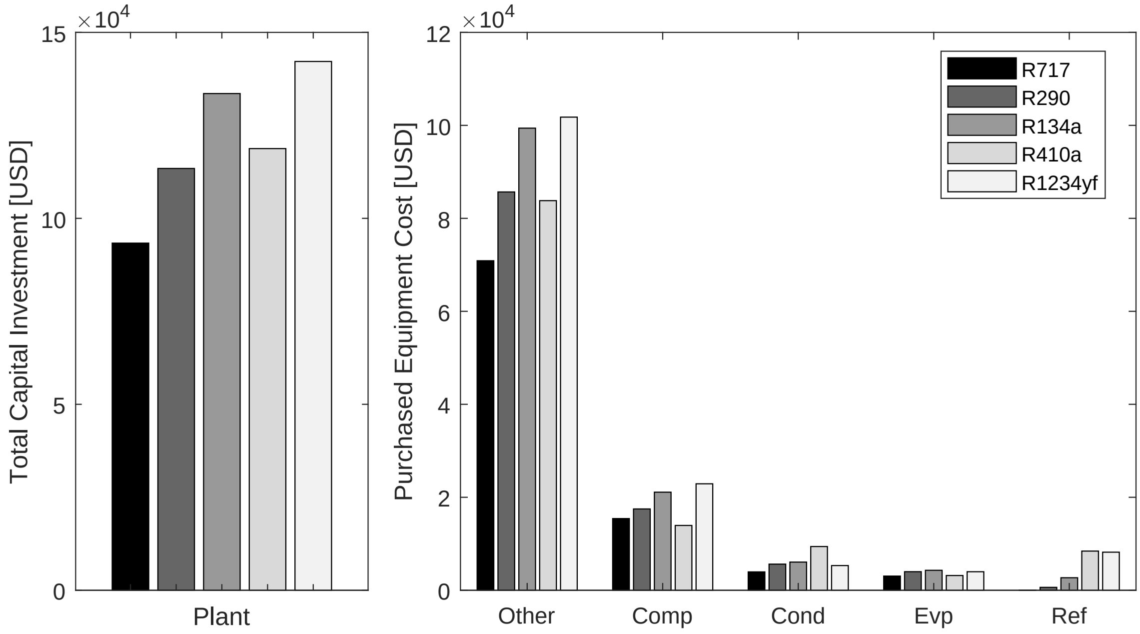


Figure 4. (left) Total capital investment of the HP system. (right) Purchased Equipment Cost

From Fig. 4 it is clear that R717 has a significantly lower TCI than the other refrigerants. This is due to smaller heat exchangers are required as well as a significantly lower mass flow rate that compensates for the higher specific volume in the compressor size. The figure also indicates that the refrigerant cost can influence the investment cost for non-natural refrigerants due to the high taxation and price. This may increase substantial when adding a liquid receiver, increasing the liquid pipe length, increasing the plate width of the condenser, or increasing the subcooling (IIR, 2014). Furthermore, Poggi et al. (2008) found that current systems of this type can have a specific cooling charge of 500-1000 g/kW suggesting that constructed systems may have a significant higher refrigerant charge. Though, R717 and R290 will not be affected as much since the price for 1 kg is relatively low. From this figure it is clear that the refrigerant price cannot be neglected when conducting economic evaluations.

4. DISCUSSION

This investigation focused on one-stage systems for simplicity and in order to ensure equal conditions for each refrigerant. As a result, the economic evaluation may have some uncertainties such as the type of components that is required for certain refrigerants. Though, the economic advantages of R717 shown in this paper follow Ommen et al. (2014) where R717 was proved to be an excellent refrigerant for industrial HPs.

For this paper, the oil container for the compressor was not included in order to simplify the model. However, Corberán et al. (2008) found that the compressor contained 28% of the refrigerant charge for a 100kW water/water propane heat pump, indicating that the oil container can have significant influence on small HPs and should be included for future work. In addition, a liquid receiver was neglected and can considerably increase the refrigerant charge of the system. If a liquid receiver should be added, a thorough estimation of its dimensions is required to maintain the minimum possible refrigerant charge.

Table 5 indicates a smaller specific heat charge compared to the collected data from Poggi et al. (2008), which found that a similar cooling system should have a specific cooling charge of 500-1000 g/kW. However, Cavallini et al. (2010), Corberán et al. (2008) and Hrnjak and Litch (2008) achieved a specific cooling charge of around 30 g/kW, 25.5 g/kW, and 20 g/kW for propane, propane, and ammonia, respectively. This suggests there is a

DOI: 10.18462/iir.gl.2018.1149

large potential for charge minimization in future systems. However, the refrigerant prices on HFCs and HFOs would still influence the economic parameters.

5. CONCLUSION

This report investigated the performance and charge of one-stage heat pump systems for commonly used refrigerants. The results showed that the refrigerant prices are important to implement when conducting an economic evaluation on a heat pump. These results are valid for HPs with small refrigerant charge and may have a significant higher influence if the refrigerant charge would increase.

The investigation showed that R717 is the most optimal refrigerant, with a 51% to 87% smaller charge and 12% to 27% lower cost of heat. However, replacing HFCs involves challenges due to the safety requirement restrictions on R717 and R290. The potential replacement refrigerant for R134a, R1234yf, has the highest cost of heat, indicating that further improvements are required in order for it to be the preferred refrigerant.

NOMENCLATURE

Symbols

A	Area (m^2)
b	Plate width (m)
e	electricity cost ($USD_{ex} \cdot kWh^{-1}$)
h	enthalpy ($kJ_{ex} \cdot kg^{-1}$)
L	length (m)
M	Charge (kg)
$\dot{m}$	Mass flow rate ($kg \cdot s^{-1}$)
op	Operation hours (h)
P	Pressure (kPa)
T	Temperature (K)
U	velocity ($m \cdot s^{-1}$)
V	Volume (m^3)
$\dot{V}$	Volume flow rate ($m^3 \cdot s^{-1}$)

Abbreviations

atm	Atmospheric pressure
COP	Coefficient Of Performance
GWP	Global Warming Potential
HFC	Hydro Fluoro Carbons
HFO	Hydro Fluoro Olefin
HP	Heat Pump
PEC	Purchased Equipment Cost
rpm	Revolutions per minute
TCI	Total Capital investment

Subscripts & Superscripts

cond	Condenser
comp	Compressor
dis	Discharge
evp	Evaporator
f	Fluid
g	Gas
is	Isentropic
liq	Liquid
pp	Pinch Point
ref	Refrigerant
sc	Subcooling
suc	Suction
sh	Superheat
s	Surface
v	Volumetric

Greek symbols

α	Void fraction
Δ	Difference
η	Efficiency
ν	Specific volume ($m^3 \cdot kg^{-1}$)
ρ	Density ($kg \cdot m^{-3}$)

REFERENCES

AGA, 2018. www.AGA.dk. Accessed: 2018-03-01.

Amalfi, R.L., Vakili-Farahani, F., Thome, J.R., 2016. Flow boiling and frictional pressure gradients in plate heat exchangers. part 2: Comparison of literature methods to database and new prediction methods. International Journal of Refrigeration 61, 185 – 203.

Ansari, N., Yadav, B., Kumar, J., 2013. Theoretical exergy analysis of hfo-1234yf and hfo-1234ze as an alternative replacement of hfc-134a in simple vapour compression refrigeration system 4, 137–144.

Bejan, A., Tsatsaronis, G., Moran, M., 1996. Thermal Design and Optimization. John Wiley ans sons.

DOI: 10.18462/iir.gl.2018.1149

Cavallini, A., Riva, E.D., Col, D.D., 2010. Performance of a large capacity propane heat pump with low charge heat exchangers. International Journal of Refrigeration 33, 242 – 250.

Corberán, J.M., Martínez, I.O., Gonzálvez, J., 2008. Charge optimisation study of a reversible water-to-water propane heat pump. International Journal of Refrigeration 31, 716 – 726. Refrigeration with Ammonia and Hydrocarbons.

Danfoss, 2017. Overview of refrigerant trends. URL: www.refrigerants.danfoss.com.

EES, 2016. Energy equation solver software. http://www.fchart.com/ees/. Accessed: 2017-11-29.

Eldeeb, R., Aute, V., Radermacher, R., 2016. A survey of correlations for heat transfer and pressure drop for evaporation and condensation in plate heat exchangers. International Journal of Refrigeration 65, 12 – 26.

Energistyrelsen, 2015. Årlig energistatistik. www.ens.dk. [online; accessed 2017-08-10].

Guntoft, S., Lauritsen, A.B., 2003. Køleteknik (Danish title). volume 2. Nyt Teknisk Forlag.

Hrnjak, P., Litch, A.D., 2008. Microchannel heat exchangers for charge minimization in air-cooled ammonia condensers and chillers. International Journal of Refrigeration 31, 658 – 668. Refrigeration with Ammonia and Hydrocarbons.

Hughmark, G.A., 1962. Holdup in gas-liqued flow. Chemical Engineering Progress 58, 62–65.

IIR, 2014. Refrigerant charge reduction in refrigerant systems. International institute of refrigerant.

Linde, 2017. Industrial gases. www.linde-gas.com. Accessed: 2017-11-29.

Lund, H., Werner, S., Wiltshire, R., Svendsen, S., Thorsen, J.E., Hvelplund, F., Mathiesen, B.V., 2014. 4th generation district heating (4gdh): Integrating smart thermal grids into future sustainable energy systems. Energy 68, 1 – 11.

Martin, H., 1996. A theoretical approach to predict the performance of chevron-type plate heat exchangers. Chemical Engineering and Processing: Process Intensification 35, 301 – 310.

Münster, M., Morthorst, P.E., Larsen, H.V., Bregnbæk, L., Werling, J., Lindboe, H.H., Ravn, H., 2012. The role of district heating in the future danish energy system. Energy 48, 47 – 55.

Ommen, T., Jensen, J.K., Markussen, W.B., Elmegaard, B., 2014. Technical and economic working domains of industrial heat pumps: part 1 - vapour compression heat pumps .

Petrović, S.N., Karlsson, K.B., 2016. Residential heat pumps in the future danish energy system. Energy 114, 787 – 797.

Poggi, F., Macchi-Tejeda, H., Leducq, D., Bontemps, A., 2008. Refrigerant charge in refrigerating systems and strategies of charge reduction. International Journal of Refrigeration 31, 353 – 370.

Secretariat, U.O., 1987. The 1987 montreal protocol on substances that deplete the ozone layer.

Secretariat, U.O., 2016. The kigali amendment (2016): The amendment to the montreal protocol agreed by the twenty-eighth meeting of the parties (kigali, 10-15 october 2016).

SKAT, 2018. E.a.7.12.5 afgiftens størrelse og beregning. www.skat.dk. Accessed: 2018-03-01.

Werner, S., 2017. International review of district heating and cooling. Energy 137, 617 – 631.

Woldesemayat, M.A., Ghajar, A.J., 2007. Comparison of void fraction correlations for different flow patterns in horizontal and upward inclined pipes. International Journal of Multiphase Flow 33, 347 – 370.

Yan, Y.Y., Lio, H.C., Lin, T.F., 1999. Condensation heat transfer and pressure drop of refrigerant r-134a in a plate heat exchanger. International Journal of Heat and Mass Transfer 42, 993 – 1006.