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Heat pump COP, part 2: Generalized COP estimation of heat pump processes

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ABSTRACT

Industrial heat pumps (IHP) are a major contributor in the transformation towards a future energy system based on electrical power. The main barrier for IHP integration is the operating cost and thereby the COP. COP is highly dependent on the temperature difference between the source and sink. Today estimation of expected COP is often done by IHP suppliers and involves choices such as working fluid and compressor technology. By extending the method (of part 1) to include a generic and generalized estimation of COP, the feasibility of integration is elaborated to include real process parameters such as working fluid, compressor and heat exchanger characteristics. The method allows analysis of the credibility of assumptions for heat pump performance and estimated COP improvement from changes of the individual characteristics. For systems with predetermined economic constraints (part 1), the extended model may be used to eliminate combinations of working fluids, compressors and heat exchangers that will not lead to a viable IHP integration.

Keywords: Natural working fluids, Process integration, Industrial heat pumps, Compressor, Heat exchanger

1. INTRODUCTION

Since 2014, power generation from renewable sources have supplied more than 50 % of the total Danish electricity consumption (53.9 % in 2016)(Energistyrelsen, 2017). With many ongoing and planned renewable power investments, projections for the years to come suggest a further increase. In the transformation towards a future energy system using electricity as main energy carrier, industrial heat pumps (IHPs) may provide a key option to reduce fossil fuel consumption and carbon emissions.

Currently, the main barrier for IHP integration is the operating cost, where the technical performance of the plant is a major influence (Ommen et al., 2015). The Coefficient Of Performance (COP) is highly dependent on the temperature lift (difference between the source and the sink), the sink and source temperature glide (temperature variation of secondary fluid) and the working fluid.

Today, estimation of COP for a given case is often done by IHP suppliers, and involves choices such as specific cycle design, compressor technology and component sizing. The lengthy estimation process may limit the introduction of IHPs, as seemingly questionable project proposals are not examined thoroughly. By use of a generalized method for screening potentials, it is likely that more projects will receive further attention (Reinholdt et al., 2018).

A number of generalized approaches to the evaluation of thermodynamic performance for heat pump cycles can be found in literature. Angelino and Invernizzi (1988) provide an approach based on the molecular complexity of the applied working fluid, in this case quantified by the partial derivative of entropy with respect to temperature for saturated vapour at a reduced temperature of 0.7. McLinden (1988) and Domanski and McLinden (1992) used reduced properties to estimate and performance and capacities of different refrigerants. Bertinat (1986) similarly investigated refrigerant selection for high temperature heat pumps based on simplified fluid properties such as critical temperatures and normal boiling point temperatures. Generally, these previous works all estimate and compare the performance based on given evaporation and condensation temperature. Although the irreversibility related to desuperheating in the condenser is accounted for in some of the studies,

the complete irreversibility related to the heat transfer between both the heat sink and the condensation and the heat source and the evaporation is neglected. Thus, the ability of the refrigerant to integrate with the sink and source is not accounted for. For industrial heat pumps, which often operate at high temperature differences in the sink and source, this contribution may be significant and should be included to fairly estimate the heat pump performance. Alefeld (1987) made a generalised approach based on the Second Law of Thermodynamics capable of including such contributions, however the results presented in this work also neglected the complete integration with the sink and source.

van de Bor and Ferreira (2013) derives the loss of efficiency directly related to the temperature driving force in the condenser and evaporator to estimate the performance along with the isentropic efficiency of the compressor. This results in an over-prediction ratio accounting for the irreversibility in the expansion and due to the additional temperature difference in the condenser and evaporator. This over-prediction ratio is found to be in agreement with the works of Angelino and Invernizzi (1988) and McLinden (1988). However, a complete generalisation was not derived.

Oluleye et al. (2016) and Oluleye et al. (2017) applies a simple linear regression model to fit the Second Law efficiency directly. This shows a good agreement with simulated data, but requires coefficients to be fitted for each refrigerant at specific temperature ranges and specific component performances such as, isentropic efficiency and minimum temperature differences. Hence, if the values of these coefficients could be correlated to simple refrigerant properties this could provide a simple generalised approach.

Part 1 (Reinholdt et al., 2018) presents a method on how to screen for feasible heat pump integration in industry using a fixed valued of Second Law efficiency. The present work presents a first attempt at deriving a generalised approach to heat pump performance accounting for both the internal irreversibilities related to compression and expansion as well as the external irreversibilities related to both the heat exchanger performance but also the ability of the refrigerant to integrate with the sink and source. Starting from a Second Law of Thermodynamics a general equation for heat pump performance was derived. In this way, the feasibility of integration was elaborated to include real process parameters such as working fluid, compressor efficiency and heat loss, as well as characteristics of both heat exchangers. The method allows analysis of the credibility of assumptions for heat pump performance. Further, it is possible to estimate COP improvement from changes of the individual characteristics. Alternatively, the extended model may be used to eliminate configurations, that will not lead to an economically or technically feasible IHP integration.

2. METHOD

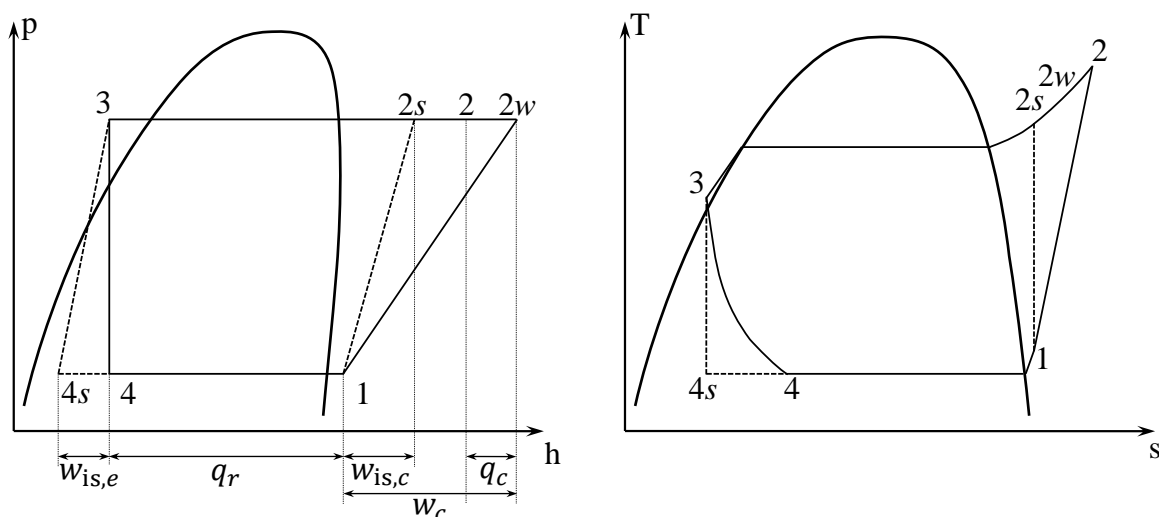


Figure 1. Pressure - enthalpy and temperature - entropy diagram of a simple one-stage heat pump process. The dashed lines represent the internally reversible process 1-2s-3-4s, while solid lines represent a real vapour compression heat pump 1-2-3-4.

Based on the Second Law of Thermodynamics it is known that the Coefficient of Performance (COP) of a reversible heat pump operating between two finite reservoir can be determined as the COP of the Lorenz cycle

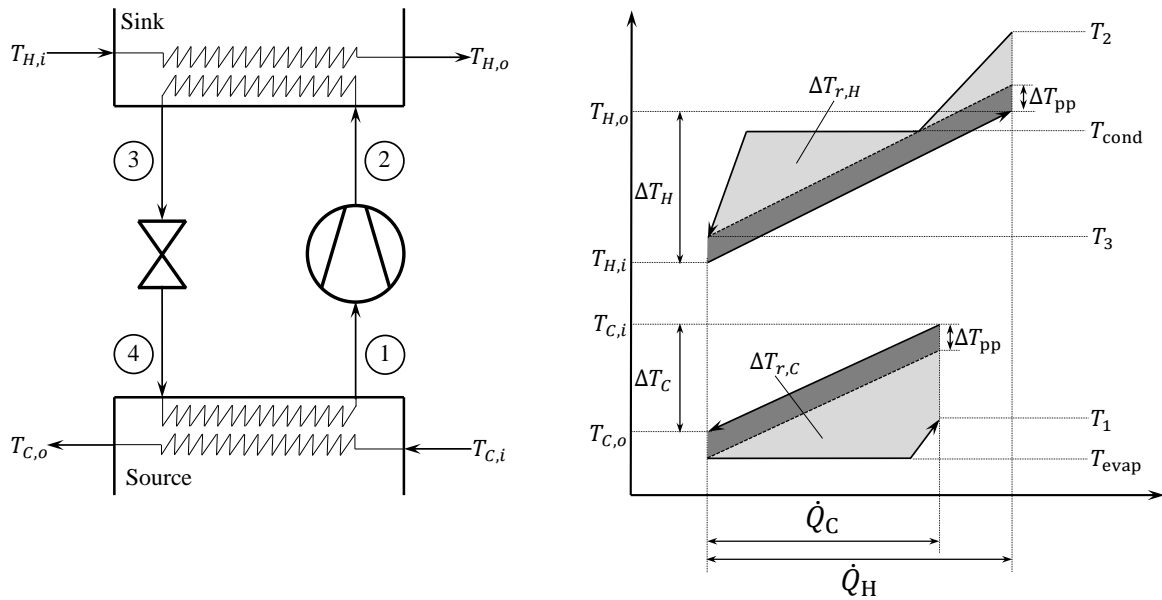


Figure 2. Cycle configuration of a simple one-stage heat pump process and the principle sketch of a temperature heat load diagram

(Lorenz, 1894).

$$\text{COP}_{\text{Lorenz}} = \frac{\bar{T}_H}{\bar{T}_H - \bar{T}_C} = \frac{\bar{T}_H}{\Delta\bar{T}_{\text{lift}}} \quad (1)$$

As seen in Eq. (1), the Lorenz COP is determined by the ratio of the entropic mean temperature of the heat sink, \bar{T}_H , relative to the difference between the entropic mean temperatures of the heat sink and heat source, in the following referred to as the mean temperature lift: $\Delta\bar{T}_{\text{lift}} = \bar{T}_H - \bar{T}_C$. Assuming the heat sink and heat source to have constant heat capacity, the entropic mean temperatures of the sink and source were determined as seen in Eq. 2.

$$\bar{T}_H = \frac{\Delta T_H}{\ln\left(\frac{T_{H,o}}{T_{H,i}}\right)} \quad \bar{T}_C = \frac{\Delta T_C}{\ln\left(\frac{T_{C,i}}{T_{C,o}}\right)} \quad (2)$$

Calculating the Lorenz COP at the entropic average temperatures of the sink and source presents the theoretical maximum COP attainable for a heat pump operating between these reservoirs. Hence, a Lorenz efficiency can be defined as the ratio between the actual heat pump COP and Lorenz COP, see Eq. (3).

$$\eta_{\text{Lorenz}} = \frac{\text{COP}}{\text{COP}_{\text{Lorenz}}} \quad (3)$$

Fig. 1 depicts the pressure-enthalpy ($p-h$) and temperature-entropy ($T-s$) diagram of a generic heat pump process for an azeotropic refrigerant under sub-critical operation. The process shown in Fig. 1 represents a simple one-stage heat pump, the schematic of which may be seen in Fig. 2. As seen in Fig. 2, the heat pump process was assumed to be comprised of an evaporator (state 4-1) in which heat from a heat source was added to the process, a compressor (state 1-2) in which the refrigerant was compressed from the evaporation to the condensing pressure, a condenser (state 2-3) where heat was rejected to the heat sink and an expansion process (state 3-4) where the pressure was lowered from the condensing to the evaporation pressure. Further, Fig. 2 presents a sketch of the heat pump process in a heat load - temperature diagram ($\dot{Q}-T$). The $\dot{Q}-T$ diagram shows a graphical representation of the temperature differences between the external sink and source streams and the internal condensation and evaporation streams, respectively. As seen, a minimum temperature difference was applied between both the condensation and the heat sink and between the evaporation and heat source. This minimum temperature difference will in the following be referred to as the pinch point temperature difference, ΔT_{pp} and was applied to ensure a finite heat transfer area. The magnitude of ΔT_{pp} is thus a measure of the heat exchanger performance, consequently a low value of ΔT_{pp} indicates a good heat exchanger performance and vice versa. It was further assumed that the refrigerant was always sub-cooled such that the condenser outlet was $T_{H,o} + \Delta T_{\text{pp}}$.

As seen in the \dot{Q} - T diagram in Fig. 2 the temperature difference in both the condenser and the evaporator can be viewed as the sum of two contributions:

- the pinch point temperature difference ΔT_{pp} induced by the finite size of the heat exchanger and the heat exchanger performance
- the refrigerant temperature difference ΔT_r induced by the ability of the refrigerant to integrate with the sink and source

It may be seen from Fig. 2 that even for an infinite heat transfer area and thus ideal heat exchanger performance ($\Delta T_{pp} = 0$ K), the refrigerant temperature difference will prevail and act as a source of irreversibility in the system. As indicated by Fig. 2 the irreversibilities solely related to the heat exchanger performance (the dark grey area) may only account for a minor part of the total irreversibility related to the heat transfer processes. Although the contribution from the heat exchanger performance cannot be neglected, this may indicate that it may be more important to choose an appropriate refrigerant that reduces ΔT_r , rather than improving heat exchanger performance to reduce ΔT_{pp} .

Two processes are shown in both the p - h and the T - s diagram, the internally reversible process (1-2s-3-4s) and the real vapour compression process (1-2-3-4). The internally reversible process was defined as a heat pump process with an isentropic and adiabatic compression and expansion, thus a system where the only source of irreversibility was related to the temperature differences to the external streams. The real vapour compression process was comprised of an irreversible compression with a given isentropic efficiency, $\eta_{is,c}$, and a given heat loss ratio, f_Q , defined as the ratio between the specific compressor heat loss $q_c = h_{2w} - h_2$ and the specific compression work $w_c = h_2 - h_1$. Further, the expansion was assumed to be isenthalpic, corresponding to an expansion with an isentropic efficiency $\eta_{is,e} = 0$.

The COP of the internally reversible cycle, $COP_{Int,Rev}$, was determined as the ratio between the condenser heat $q_H = q_r + w_{is,c}$ and the net work supplied to the system $w = w_{is,c} - w_{is,e}$. In Eq. (4), this is further presented based on the specific enthalpies. From the T - s diagram in Fig. 1, it can be seen that due to the isentropic compression and expansion of the internally reversible cycle the entropy change was the same ($s_1 - s_3$) over both the condensation and evaporation processes. Dividing both the numerator and denominator with the entropy difference resulted in the final term in Eq. (4), stating that the COP of an internally reversible cycle can be determined solely based on the entropic average temperatures of condensation and evaporation. The entropic average temperatures were determined as the ratio of the enthalpy change to the entropy change as seen in Eq. (5).

$$COP_{Int,Rev} = \frac{q_H}{w} = \frac{q_r + w_{is,c}}{w_{is,c} - w_{is,e}} = \frac{h_{2s} - h_3}{(h_{2s} - h_1) - (h_3 - h_{4s})} = \frac{\bar{T}_{cond,is}}{\bar{T}_{cond,is} - \bar{T}_{evap,is}} \quad (4)$$

$$\bar{T}_{cond,is} = \frac{h_{2s} - h_3}{s_1 - s_3}, \quad \bar{T}_{evap,is} = \frac{h_1 - h_{4s}}{s_1 - s_3} \quad (5)$$

Defining the entropic average temperature differences in the condenser and evaporator, as seen in Eq. (6), resulted in the derivation seen in Eq. (7), thus relating the COP of the internally reversible cycle to the Lorenz COP and the average temperature differences in the condenser and evaporator.

$$\Delta \bar{T}_{cond,is} = \bar{T}_{cond,is} - \bar{T}_H = \Delta \bar{T}_{r,H} + \Delta \bar{T}_{pp}, \quad \Delta \bar{T}_{evap,is} = \bar{T}_C - \bar{T}_{evap,is} = \Delta \bar{T}_{r,C} + \Delta \bar{T}_{pp} \quad (6)$$

$$COP_{Int,Rev} = \frac{\bar{T}_H + \Delta \bar{T}_{cond,is}}{(\bar{T}_H + \Delta \bar{T}_{cond,is}) - (\bar{T}_C - \Delta \bar{T}_{evap,is})} = COP_{Lorenz} \frac{1 + \frac{\Delta \bar{T}_{r,H} + \Delta \bar{T}_{pp}}{\bar{T}_H}}{1 + \frac{\Delta \bar{T}_{r,H} + \Delta \bar{T}_{r,C} + 2\Delta \bar{T}_{pp}}{\Delta \bar{T}_{lift}}} \quad (7)$$

A general form of heat pump COP can be seen in the first statement of Eq. (8). However, recognising that real vapour compression heat pumps operate with isenthalpic expansion and thus with $\eta_{is,e} = 0$, this was reduced to

the second statement.

$$\text{COP} = \frac{q_r + (1 - f_Q) \frac{w_{is,c}}{\eta_{is,c}}}{\frac{w_{is,c}}{\eta_{is,c}} - \eta_{is,e} w_{is,e}}, \quad \text{COP}|_{\eta_{is,e}=0} = \eta_{is,c} \frac{q_r}{w_{is,c}} + (1 - f_Q) \quad (8)$$

As seen from Eq. (4), q_r can be expressed based on the internally reversible COP and the work of isentropic compression and expansion. Substituting Eq. (4) into Eq. (8) resulted in the following relation for heat pump COP, see Eq. 9

$$\text{COP}|_{\eta_{is,e}=0} = \left(\text{COP}_{\text{Lorenz}} \frac{1 + \frac{\Delta\bar{T}_{r,H} + \Delta\bar{T}_{pp}}{\bar{T}_H}}{1 + \frac{\Delta\bar{T}_{r,H} + \Delta\bar{T}_{r,C} + 2\Delta\bar{T}_{pp}}{\Delta\bar{T}_{\text{lift}}}} \eta_{is,c} \left(1 - \frac{w_{is,e}}{w_{is,c}} \right) + 1 - \eta_{is,c} - f_Q \right) \quad (9)$$

Applying the definition in Eq. (3), a general relation for the Lorenz efficiency of a heat pump with isenthalpic expansion, Eq. (10), was derived.

$$\eta_{\text{Lorenz}}|_{\eta_{is,e}=0} = \frac{1 + \frac{\Delta\bar{T}_{r,H} + \Delta\bar{T}_{pp}}{\bar{T}_H}}{1 + \frac{\Delta\bar{T}_{r,H} + \Delta\bar{T}_{r,C} + 2\Delta\bar{T}_{pp}}{\Delta\bar{T}_{\text{lift}}}} \eta_{is,c} \left(1 - \frac{w_{is,e}}{w_{is,c}} \right) + (1 - \eta_{is,c} - f_Q) \frac{\bar{T}_H}{\Delta\bar{T}_{\text{lift}}} \quad (10)$$

As seen, the derived relations presented in Eqs. (9) and (10) present a generalised approach to COP or η_{Lorenz} calculation based on known: operating conditions in terms of sink and source temperatures, compressor characteristics in terms of $\eta_{is,c}$ and f_Q , heat exchanger characteristics in terms of the $\Delta\bar{T}_{pp}$, and refrigerant characteristics in terms of the ratio $\frac{w_{is,e}}{w_{is,c}}$ and the refrigerant induced temperature differences $\Delta\bar{T}_{r,C}$ and $\Delta\bar{T}_{r,H}$. As seen the ratio $\frac{w_{is,e}}{w_{is,c}}$ plays an important role in determining the impact of the refrigerant on heat pump performance. As seen from the derived relations, the lower the ratio $\frac{w_{is,e}}{w_{is,c}}$ was: the higher the heat pumps performance would be. Hence, knowledge of this ratio was found to be a key parameter in the generalisation of heat pumps performance.

Further approximations were applied to the heat exchanger pinch and the refrigerant induced temperature difference in the evaporator, see Eq. (11). For the assumed constant capacity sink and source streams the entropic average temperature induced by the heat exchanger pinch was approximated with reasonable accuracy as the actual value of the pinch point temperature difference. The refrigerant induced temperature difference in the evaporator, $\Delta\bar{T}_{r,C}$, was approximated as half the value of the source inlet-outlet temperature difference, ΔT_C . This results in a good approximation for an azeotropic refrigerant with moderate superheating and a constant capacity source.

$$\Delta\bar{T}_{pp} \approx \Delta T_{pp}, \quad \Delta\bar{T}_{r,C} \approx \frac{1}{2} \Delta T_C \quad (11)$$

Applying these approximations, infer that the performance of a heat pump with given operating conditions and component characteristics can be determined if the refrigerant parameters $\frac{w_{is,e}}{w_{is,c}}$ and $\Delta\bar{T}_{r,H}$ are known. Methods to estimate these parameters may thus be helpful to perform a simple performance estimation of a potential heat pump installation. In the present work a simple linear approximation, in the form seen in Eq. (12) was applied to fit these values for the refrigerants Ammonia and Isobutane. This fit was based on a simulation model applied in the range of operating conditions and component characteristics seen in Table 1.

$$\Delta\bar{T}_{r,H} = a (T_{H,o} - T_{C,o} + 2\Delta T_{pp}) + b\Delta T_{\text{sink}} + c, \quad \frac{w_{is,e}}{w_{is,c}} = a (T_{H,o} - T_{C,o} + 2\Delta T_{pp}) + b\Delta T_{\text{sink}} + c \quad (12)$$

Further, the model was applied to several additional refrigerants in the full range of operating conditions and component characteristics seen in Table 1. This was done to give an estimation of the variability of these two parameters both between different refrigerants but also between different operating conditions

Table 1. Range of simulated operating conditions and component characteristics

Operating conditions:	Range	Component characteristics:	Range
$T_{H,o}$	300 K to $0.9T_{crit}$	$\eta_{is,c}$	0.4 - to 0.9 -
$\Delta\bar{T}_{lift}$	20 K to 80 K	f_Q	0.0 - to 0.6 -
ΔT_{sink}	10 K to 50 K	ΔT_{pp}	0.0 K to 10 K
ΔT_{source}	10 K to 30 K	ΔT_{SH}	0.0 K to 10 K

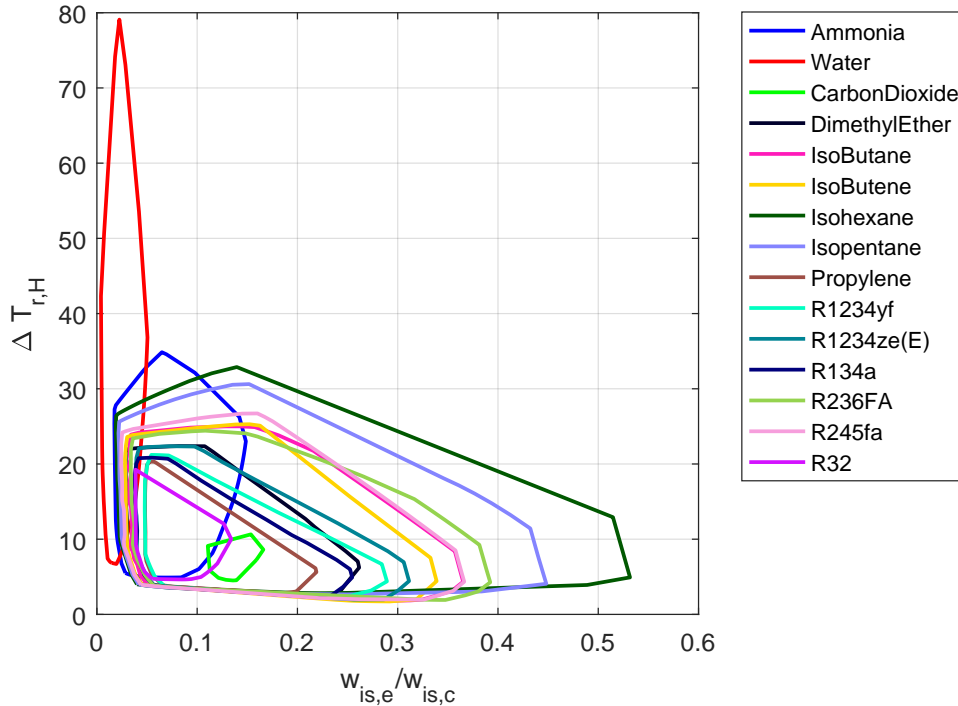


Figure 3. Variation of $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for a range of refrigerants under the operating conditions and component characteristics presented in Table 1

3. RESULTS

Fig. 3 shows the variation of the parameters $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for a number of selected refrigerants under the range of operating conditions and component characteristics presented in Table 1. As seen, both parameters show a large variability with $\frac{w_{is,e}}{w_{is,c}}$ ranging 0.01 to 0.55 and $\Delta\bar{T}_{r,H}$ from 5 K to 80 K. The variation within the individual refrigerants also differ. While refrigerants such as ammonia and water generally have low values of $\frac{w_{is,e}}{w_{is,c}}$ and high values of $\Delta\bar{T}_{r,H}$ the opposite was found for refrigerants such as R1234yz and isobutane. It should be noted that a small variation could also be caused by a low amount of feasible operating conditions in the given range. This was e.g. the case for carbondioxide as supercritical conditions were not considered.

All in all Fig. 3 shows that the variation of $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ was large both for individual refrigerants and between refrigerants. Thus, a further analysis of the sources of variation was needed. Fig. 4 shows the variation among all simulated refrigerants divided into smaller intervals of the temperature lifts and sink temperature differences. As seen this results in significantly smaller variations under operating conditions with small temperature lifts. However, the variation increases as the temperature lift increases. Further, it may be seen that the value of both $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ should be expected to increase as the temperature lift increases. Fig. 4 also shows that as the sink temperature difference increases, the variation of $\frac{w_{is,e}}{w_{is,c}}$ tends to decrease and further that a lower value of $\frac{w_{is,e}}{w_{is,c}}$ should be expected. This was caused by the increased level of subcooling attained with the increase of ΔT_{sink} . Over all, Fig. 4 shows that the selection of refrigerants was most critical when the temperature lift was high and the sink temperature difference was low. Under these conditions the variation of $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ indicate that the choice of refrigerant may have a large impact on the heat pump performance.

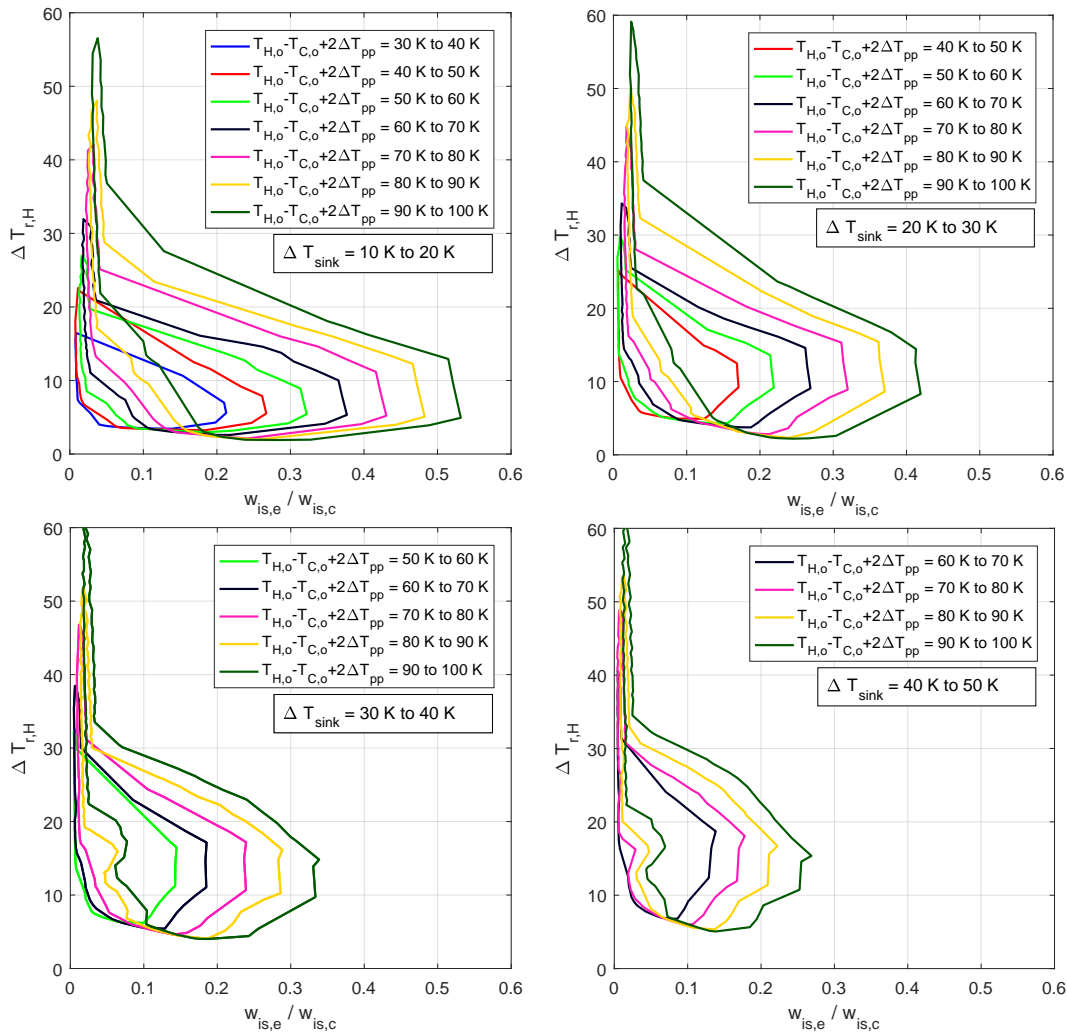


Figure 4. Variation of $\Delta T_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for all refrigerants divided into ranges of temperature lift and sink temperature difference

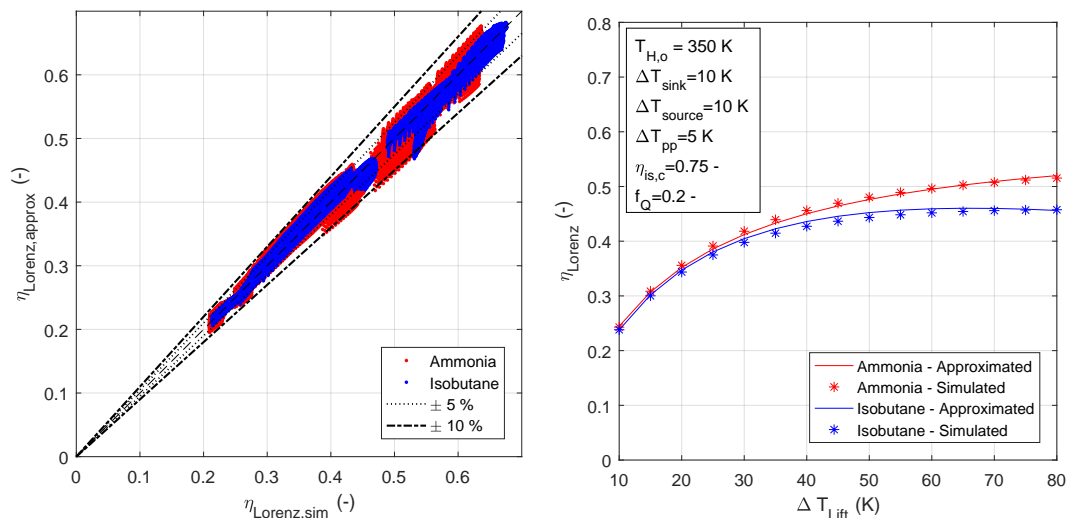


Figure 5. Simulated COP vs. the approximated COP using the linear models for $\Delta T_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ with the coefficients seen in Table 2.

Using the simulated data for ammonia and isobutane the coefficients for the linear models seen in Eq. (12) were found. These are presented in Table 2. Fig. 5 shows the simulated Lorenz efficiency versus the Lorenz efficiency approximated by the use of Eq. (9) in combination with linear approximation models for the range of operating conditions and component characteristics seen in Table 1. As seen, the generalised approach was

capable of estimating the Lorenz efficiency within a $\pm 10\%$ deviation. Most isobutane solutions were within a $\pm 5\%$ deviation. Further, Fig. 5 shows the evolution of the simulated and approximated Lorenz efficiency for an ammonia and an isobutane heat pump delivering heat at 350 K for fixed component characteristics and sink and source temperature glides. As seen, the approximated values show a good agreement with the simulated results over the entire range of temperature lifts. Further, this shows that the Lorenz efficiency increases with temperature lift for a heat pump with fixed component characteristics. This behaviour was caused by the increasing importance of the condenser and evaporator temperature differences at low temperature lifts. As seen in Eq. (10), the temperature differences in the denominator are divided by the temperature lift and thus this contribution diminished as the temperature lift increased.

Table 2. Coefficient for the linear models for $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for both ammonia and isobutane

	Ammonia			Isobutane		
	a	b	c	a	b	c
$\frac{w_{is,e}}{w_{is,c}}$	0.0014	-0.0015	0.039	0.0035	-0.0033	0.053
$\Delta\bar{T}_{r,H}$	0.20	0.20	0.016	-0.0011	0.30	2.4

Fig. 6 shows $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for ammonia as a function of the temperature difference between the sink and source outlet and the sink temperature difference with the color scale indicating the reduced temperature of the heat sink. Similar trends were observed for isobutane. As seen, both $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ were a function of the reduced temperature. High reduced temperature results in high $\frac{w_{is,e}}{w_{is,c}}$ and higher $\Delta\bar{T}_{r,H}$. Hence, if the reduced temperature was included in the estimation of $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ higher accuracy should be expected.

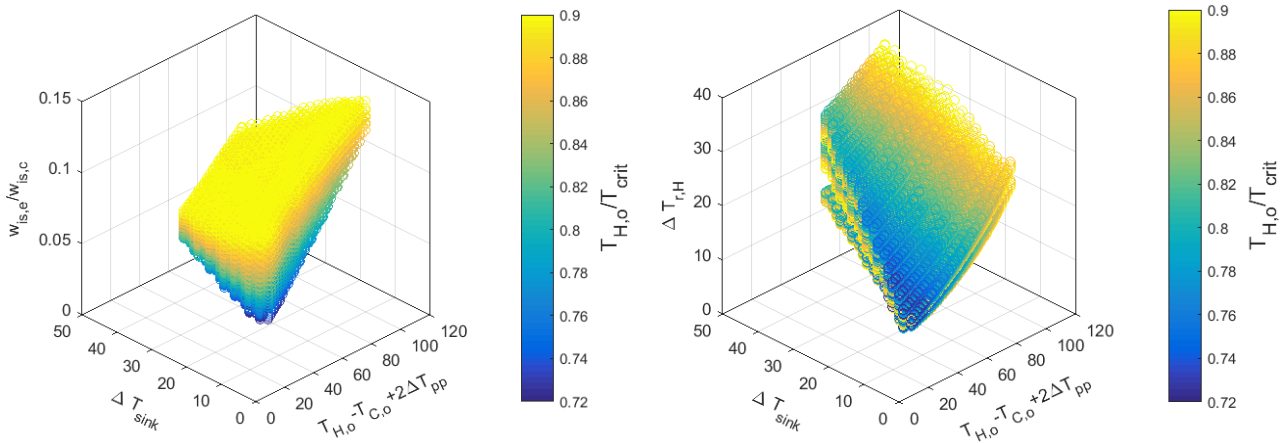


Figure 6. $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for ammonia as a function of the temperature lift and sink temperature difference. The color scale indicates the reduced temperature of the sink outlet.

Fig. 7 shows $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for all simulated refrigerants with the color indicating the heat capacity ratio, $\kappa = \frac{c_p}{c_v}$ for the given refrigerant. As seen, the values of $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ were highly dependent on refrigerants heat capacity ratio. High κ values results in high $\frac{w_{is,e}}{w_{is,c}}$ but conversely low $\Delta\bar{T}_{r,H}$. Hence, an even more general model could be attained if the relation between κ and $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ was determined.

4. DISCUSSION

As seen in the presented results, it seems possible to produce similar generalisation methods, as those derived by Angelino and Invernizzi (1988), McLinden (1988) and Alefeld (1987), which additionally include the irreversibility related to the integration with the heat sink and heat source. Using the derived relation it was shown that the refrigerant induced temperature difference in the evaporator could be estimated solely based on the source temperature difference and the evaporator pinch point temperature difference, under the assumption that the refrigerant was azeotropic. Hence, for an azeotropic refrigerant this contribution was found to be refrigerant independent. This was not the case for the refrigerant induced temperature difference in the condenser. How-

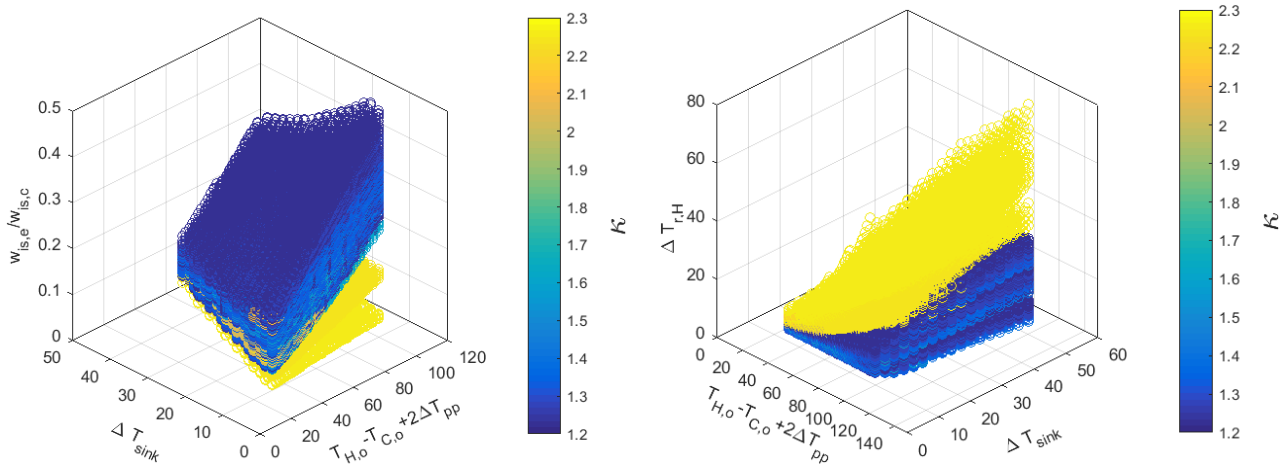


Figure 7. $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ for all simulated refrigerants as a function of the temperature lift and sink temperature difference. The color scale indicates the heat capacity ratio of the refrigerant.

ever, this value was shown to correlate well with the reduced temperature of the heat sink and the refrigerant molecular complexity, quantified by the heat capacity ratio. Finally, the internal irreversibilities were accounted for by the ratio of isentropic expansion work to isentropic compression work. This ratio hence accounted for similar effects as those introduced by Angelino and Invernizzi (1988), McLinden (1988) and Alefeld (1987) and was found to vary in agreement with their findings, hence reducing heat pump performance for increased molecular complexity and increasing reduced temperature. Conversely, it was found that the refrigerant induced temperature difference in the condenser was reduced for more complex molecular structures thus to some extent counteracting the increased internal irreversibility. Finally, it was found that both refrigerant specific parameters were highly dependent not only the temperature lift, as suggested by Angelino and Invernizzi (1988), McLinden (1988) and Alefeld (1987) but also on the sink temperature difference.

5. CONCLUSIONS

A first attempt towards a generalised performance estimation method for heat pump performance was presented. A general equation for the COP and Lorenz efficiency of a heat pump was derived linking the performance of a heat pump to the operating conditions, component characteristics as well refrigerant properties. This revealed that for an azeotropic refrigerant, the refrigerant impact on the performance could be accounted for by two parameters: $\Delta\bar{T}_{r,H}$ the refrigerant induced temperature difference in the condenser and $\frac{w_{is,e}}{w_{is,c}}$ the ratio between the work of isentropic expansion and compression. Fitting a linear model of these two parameters for ammonia and isobutane allowed an approximation of the Lorenz efficiency within $\pm 10\%$ for a large variation of operating conditions and component characteristics. Higher accuracy should be expected if the estimation methods accounted for the reduced temperature of the heat sink. Additionally, it was found that the values of $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ correlate well with the heat capacity ratio, κ , of the refrigerant. Hence, estimation methods of $\Delta\bar{T}_{r,H}$ and $\frac{w_{is,e}}{w_{is,c}}$ based solely on heat capacity ratio and the reduced temperature of the heat sink may be possible thus resulting in a generalised model for performance estimation.

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NOMENCLATURE

<i>symbols</i>		<i>Subscripts</i>	
a	coefficient for the linear model	approx	approximation
b	coefficient for the linear model	c	compression
c	coefficient for the linear model	C	cold stream (heat source)
COP	coefficient of performance (-)	cond	condenser
f_Q	compressor heat loss ratio (-)	e	expansion
h	enthalpy (kJ kg^{-1})	evap	evaporator
q	heat (kJ kg^{-1})	H	hot stream (heat sink)
s	entropy (kJ (kg K)^{-1})	i	inlet
T	temperature (K)	is	isentropic
\bar{T}	entropic average temperature (K)	Int	internally
w	work (kJ kg^{-1})	Lift	lift
		Lorenz	lorenz cycle
<i>Greek</i>		o	outlet
Δ	difference	pp	pinch point
η	efficiency (-)	r	refrigerant
κ	heat capacity ratio (-)	Rev	reversible

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