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Holmberg, Henrik; Laukkanen, Timo

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Exergetic performance analysis of heat pumps: two alternative approaches

Abstract

This paper evaluates the performance of a compressor driven and an absorption heat pump on the basis of exergy analysis using two different approaches. Approach 1 is based on the calculation of the effective heat absorbing and emitting temperatures and the entropy generation rate. Approach 2 is based on the commonly used exergy analysis where the real ambient temperature is used. The main goal is to analyze how these two approaches differ from each other. Approach 1 gives the exact improvement potential ($W_{real} - W_{min}$) of the heat pump while Approach 2 underestimates it. The average flow temperature also approximates the correct improvement potential with sufficient accuracy when temperature change of the flow is relatively small. Both approaches can be used when performances of heat pumps are compared. Both approaches also give a more realistic view of the performance than the commonly used COP, especially regarding the absorption heat pump.

Nomenclature

Abbreviations

COP Coefficient of performance [-]

Symbols

$\dot{E}_x$	Exergy [W]
h	Enthalpy [Jkg^{-1}]
$\dot{m}$	Mass flow rate [kgs^{-1}]
$\dot{Q}_{in}$	Heat input [W]
$\dot{Q}_{out}$	Heat output [W]
s	Entropy [$\text{Jkg}^{-1}\text{K}^{-1}$]
T	Temperature [K]
t	Temperature [$^{\circ}\text{C}$]
$\dot{W}$	Work [W]
ε	Rational ratio [-]
η_{ex}	Exergy efficiency [-]
δ	Entropy generation rate [WK^{-1}]

Subscripts

<i>abs</i>	Absorber
<i>con</i>	Condenser
<i>ev</i>	Evaporator
<i>gen</i>	Generator
<i>in</i>	Input
<i>out</i>	Output

1 Introduction

Heat pumps are used to upgrade low temperature heat to high temperature heat which is used in several different heat sinks. Typical heat sinks are industrial processes, heating of spaces and water in buildings and district heat production. The most common heat pump applications are compressor driven heat pumps which use mechanical work for upgrading the low temperature heat. Heat pump applications based on absorption technology do not need any mechanical work. Instead they need an external heat source. The temperature of this heat source must be higher than the temperature of the heat source that is going to be upgraded.

The most common way to evaluate the performance of the heat pump is to define the Coefficient of Performance (COP) value. The COP value depends on the temperature of the heat source (e.g. ground, solar, sea water, air, waste heat), temperature level of the heat sink and a heat pump configuration. Cordin et.al. (2018) made a comprehensive review of high temperature (temperature of the heat sink 90-160°C) compressor driven heat pumps available in the markets. The COP values were reported to vary from 2.2 to 6.5 depending on the temperature lift in the pump. The higher the lift the lower the COP is. Willem et.al. (2017) made a review of compressor driven heat pumps that were used for water heating in households. On the basis of review, most current heat pumps in this field operated in the COP range of 1.8-2.5 but there are potential technologies for the COP range of 2.8 – 5.5. Hepbasli et.al. (2014) reviewed existing compressor driven waste water heat pump systems. For most of these systems the COP value was between 2.5 and 5.7 even though a COP value of 10.63 was also mentioned in the paper. As previous examples show the COP value may vary in the very wide range for compressor driven heat pumps. COP values are typically lower when absorption heat pumps are used. For absorption heat pumps COP values of 1.05-1.8 were reported by Wu et.al. (2014) in a comprehensive review where various absorption based heating technologies were presented. Regarding absorption heat pumps, COP might not be the most relevant efficiency metric, especially when it is used to compare different absorption heat pumps (Jernqvist et al., 1992).

The performance of heat pumps can also be evaluated using the exergy analysis. The traditional exergy analysis has been discussed in detail for example by Szargut (1988) Kotas (1995) and Dincer and Rosen (2007). The great advantage of the exergy analysis is that it makes it possible to calculate the improvement potential of the system without going into details of the system. In the case of heat pumps, only the work input, heat inputs, heat outputs and temperatures of the heat source and sink must be known to define the improvement potential.

For a compressor driven heat pump, the improvement potential describes how much shaft work input could be reduced compared to a reversible heat pump. For an absorption heat pump, the improvement potential means how much more work potential of heat sources is lost compared to some minimum loss of potential. The term describing the improvement potential is the irreversibility rate. The exergy or exergetic efficiency is calculated by dividing the exergy content of output flows by the exergy content of input flows. The exergy content is calculated with respect

to the dead/reference state which is usually the real ambient temperature. The weakness of the exergy analysis is that it does not give any specific information on how the performance of the system could be improved.

Exergy analyses can be made for all kinds of heat pump systems. Arat and Arslan (2017) analyzed a compressor driven geothermal heat pump for different design parameters. On the basis of the analysis the exergetic efficiency typically ranged from 0.45 to 0.65 depending on the parameters. For another geothermal heat pump system the exergetic efficiency was reported to vary from 0.517 to 0.633 depending on the reference/dead state temperature (Hepbasli et.al., 2007). The efficiency decreases as the reference temperature increases. A similar finding was also made by Soltani et.al. (2015) and Saloux et.al. (2016). Stanek et al., 2019 studied a compression-driven heat pump system from both a local (exergy efficiency) and global (thermo-ecological cost) point of view. According to their study, the local (exergy efficiency) approach might lead to increased utilization of non-renewable electricity sources, because exergy efficiencies of power plants using fossil fuels are higher. Thus the global approach (thermo-ecological cost) should be the metric used for evaluating heat pumps. Wu et al. (2018) proposed a hybrid source heat pump water heater system (with R1234ze(Z) and CO₂ as refrigeration fluids) in order to improve the system performance and exergy efficiency. According to their results, the hybrid system improved both the system COP and exergy efficiency by 24.8% and 27.2% at a fixed condensing temperature compared to single cycle configuration. Cakir et.al studied multifunctional heat pump system using just one scroll compressor and which could be run in four different modes, namely air to air, air to water, water to water and water to air, in order to make an experimental energetic and exergetic performance comparison. Water to air pump had the highest exergy efficiency with respect to pump's mean exergy efficiency. Respectively, water to water pump had the lowest mean exergy efficiency. However, rankings of pumps varied depending the process values (e.g. inlet temperature of the heat source)

Li et.al. (2013) used exergy analysis for a more complex system where utilization of wind and solar energy was coupled with a compressor driven heat pump. Dong et.al. (2017) compared thermodynamic performances of the solar integrated air source heat pump (SIASHP) and conventional air source heat pump (ASHP). Results of the study showed that SIASHP has a better thermodynamic cycle effect than ASHP. Cho (2015) analyzed the performance and exergy losses of the R22 and R744 solar-hybrid heat pumps systems experimentally during sunny and cloudy days. On the basis of results, the second law efficiency of the entire R22 heat pump was 6.2% higher than that of the R744. Menberg et al., (2017) studied a compression-driven ground heat source heat pump integrated with a gas-fired boiler system both in heating and in cooling mode. They used non-natural exergy efficiency, natural exergy efficiency and system exergy efficiency in order to understand the effect of gas-fired boiler system to the overall exergy efficiency. Based on their study, the exergy performance of the system in heating mode was twice as high as for cooling mode (30% to 15%), while the natural exergy performance was considerably better in cooling mode (26% to 3%). Kilikis et al., (2018) used exergy-based COP-values to evaluate heat pumps as part of their study on hydrogen economy for nearly net-zero districts.

Zhang et.al. (2016) reported exergetic efficiencies for a LiBr-H₂O absorption heat pump in a paper where they summarized comprehensively the recent research on various industrial heat pump applications in China. The exergetic efficiency was between 0.8-0.85. Liu et.al. (2017) also analyzed a LiBr-H₂O based absorption heat pump using the exergy analysis. In the study, authors introduced a new type of combined heat pump transformer. The exergetic efficiency was reported to be 0.71. The value is c. 33 % higher than the corresponding efficiency of a reference system

Ally et.al. (2017) have published an interesting paper about a ground source heat pump where they have calculated irreversibility rates and the Carnot COP for an existing system. Authors correctly state that the work of an actual compressor must be the sum of the total irreversibility rate and the work input of an ideal Carnot compressor. The reference temperature for irreversibility calculations is the average of the entering and leaving flow temperatures over the system. Authors define the Carnot COP and the reference temperature slightly different from what we are going to do in this paper. Cho (2105) has also calculated irreversibility rates for a compression drive solar hybrid heat pump. In his study, irreversibility rates have been calculated by multiplying the entropy generation by the actual ambient temperature.

In the previous-mentioned studies, the exergy content has been defined with respect to a constant reference state temperature with the exception of Ally at.al. (2017) where the average of flow temperatures has been used. Usually, the reference temperature is the same as the actual ambient temperature which is a normal practice in exergy analysis. A slightly different approach to exergy analysis has been presented by Lampinen and Wiksten (2006). This approach defines the irreversibility rate with respect to the so called heat emitting temperature, which is not always the same as the actual ambient temperature. This approach makes it possible to calculate the minimum shaft work for a compressor driven heat pump when flow temperatures over the heat pump change from a given initial temperature to a desired final temperature. Respectively, the minimum and real loss of work for an absorption heat pump can be calculated using the approach presented by Lampinen and Wiksten (2006).

The main goal of the paper is to use two approaches of exergy analysis to basic heat pump systems and analyze how results of these two approaches differ from each other. Relevant questions are then how and why results of both analyses differ from each other and how the results should be interpreted. Is it for example possible that these approaches could give controversial information on the performance of heat pumps? Earlier heat pump studies have consistently used only one approach (typically the traditional exergy analysis) and the differences between these two approaches have not been comprehensively analyzed.

Both approaches are applied to a compressor driven heat pump and an absorption heat pump. In this paper, Approach 1 always refers to the theory presented by Lampinen and Wiksten (2006). The performance parameter used in this context is the rational ratio. Approach 2 always means results calculated using the traditional exergy analysis. The performance parameter used in this context is the exergy efficiency.

2 Methods

2.1 Description of heat pumps

Figures 1 and 2 show flow sheets and main components of a compressor driven and an absorption heat pump, respectively. In both cases, low temperature heat from a heat source A is transferred into heat sink B. For a compressor driven heat pump, the COP is defined as follows:

$$COP = \frac{\dot{Q}_{con}}{\dot{W}} \quad (1)$$

where $\dot{Q}_{con}$ represents heat output of the process and $\dot{W}$ is shaft work of the compressor (see Fig. 1). If losses between the electric motor and compressor are neglected, the shaft work is the same as the shaft power of the compressor.

For an absorption heat pump, the COP is defined as follows:

$$COP = \frac{\dot{Q}_{con} + \dot{Q}_{abs}}{\dot{Q}_{gen}} \quad (2)$$

where $\dot{Q}_{con}$ and $\dot{Q}_{abs}$ represent heat released into the environment in the condenser and absorber, respectively. $\dot{Q}_{gen}$ denotes the heat input into the generator (see Fig. 2).

An absorption heat pump does not need any mechanical work input. In reality, most practical applications are equipped with a circulation pump that circulates absorption solution between the absorber and generator. It is, however, possible to build an absorption heat pump or a cooling system that does not consume mechanical work at all, for example a gas refrigerator.

2.2 Approach 1

When Approach 1 is used the entropy generation rate of the heat pump and the effective heat emitting and absorbing temperatures are defined. The basic theory of Approach 1 has been explained comprehensively by Lampinen and Wiksten (2006). This chapter just summarizes necessary equations needed to define the rational ratio for both heat pump types. In this paper, the rational ratio means the ratio between the real COP and the best possible COP. All notations used in this chapter refer to Figures 1 and 2.

2.2.1 Compressor driven heat pump

A compressor driven heat pump is a cycle process. The work input into the process may be written as follows (Lampinen and Wiksten, 2006):

$$\dot{W}_{in} = \left(\frac{T_{out}}{T_{in}} - 1 \right) \dot{Q}_{ev} + T_{out} \delta \quad (3)$$

where $\dot{W}_{in}$ is the real work input into the system, T_{out} is the effective heat emitting temperature, T_{in} is the effective heat absorbing temperature, $\dot{Q}_{ev}$ is heat input into the process in a evaporator and δ is the entropy generation rate over the heat pump process. Derivation of Eq. (3) is based on the assumption that work input into the system is a negative one.

Heat input $\dot{Q}_{ev}$ is not known but heat output $\dot{Q}_{con}$ in a condenser is known. Heat output represents $\dot{Q}_{con}$ the heat demand of the heat sink that is produced by the heat pump. Work input into the compressor may also be expressed as follows:

$$\dot{W}_{in} = \dot{Q}_{con} - \dot{Q}_{ev} \quad (4)$$

Substituting $\dot{Q}_{ev} = \dot{Q}_{con} - \dot{W}_{in}$ in (3) $\dot{W}_{in}$ becomes

$$\dot{W}_{in} = \left(1 - \frac{T_{in}}{T_{out}} \right) \dot{Q}_{con} + T_{in} \delta \quad (5)$$

where the term $T_{in} \delta$ represents the irreversibility rate of the heat pump. If T_{in} is replaced with the actual ambient temperature the term represents the Goyu-Stodola law.

The best possible heat pump does not generate any entropy. In this case, Eq. (5) gives for the minimum work

$$\dot{W}_{in,min} = \left(1 - \frac{T_{in}}{T_{out}} \right) \dot{Q}_{con} \quad (6)$$

where $\dot{W}_{in,min}$ is the minimum work input needed by the compressor. On the basis of Eqs. (1) and (6), the best possible COP_{max} becomes

$$COP_{max} = \frac{\dot{Q}_{con}}{\dot{W}_{in,min}} = \frac{\dot{Q}_{con}}{\left(1 - \frac{T_{in}}{T_{out}} \right) \dot{Q}_{con}} = \frac{T_{out}}{T_{out} - T_{in}} \quad (7)$$

Using notations in Fig. 1, the effective heat emitting temperature (T_{out}) and the effective heat absorbing temperature (T_{in}) are defined as follows (Holmberg and Ahtila, 2014):

$$T_{out} = \frac{T_{A2} - T_{A1}}{\ln \frac{T_{A2}}{T_{A1}}} \quad (8a)$$

$$T_{in} = \frac{T_{B2} - T_{B1}}{\ln \frac{T_{B2}}{T_{B1}}} \quad (8b)$$

If T_l is the same as T_2 the effective temperature in (8a) and (8b) is the same as T_l (Lampinen and Heikkiä, 1995). For example, if heat into the evaporation comes from the ground or sea, the effective temperature T_{in} is the same as the ground or sea temperature. When effective temperatures are used, temperatures of the heat source and heat sink may change in an evaporator and a condenser and Eq. (7) still gives exactly the correct value for the COP_{max} . This also means that the temperature of the heat source does not have to be the real ambient temperature. For example when waste heat streams are used as heat sources, the temperature of the heat source typically changes in an evaporator and its temperature is higher than the real ambient temperature.

For a compressor driven heat pump the performance parameter that describes the relative improvement potential of the pump is expressed as follows:

$$\varepsilon = \frac{COP}{COP_{max}} = \frac{\dot{W}_{in,min}}{\dot{W}_{in}} \quad (9)$$

This ratio ε is called the rational ratio in this paper when Approach 1 is used. Some references may also call Eq. (9) the exergy or the exergetic efficiency. However, this paper consistently calls Eq. (9) the rational ratio to avoid misunderstandings between the first and the second approach.

2.2.2 Absorption heat pump

For an absorption heat pump the energy balance is

$$\dot{Q}_{ev} + \dot{Q}_{gen} = \dot{Q}_{abs} + \dot{Q}_{con} \quad (10)$$

where $\dot{Q}_{ev}$ is the heat input into the evaporator and $\dot{Q}_{gen}$ is the heat input into the generator. $\dot{Q}_{abs}$ and $\dot{Q}_{con}$ represent heat outputs from the absorber and condenser, respectively. In Eq. (10) $\dot{Q}_{abs} + \dot{Q}_{con}$ is the same as the heat demand of flow A.

An absorption heat pump does not need any mechanical work which means that $\dot{W}_{in}$ in Eq. (3) is zero. An ideal absorption heat pump does not either generate any entropy. As a result of this, Eq. (3) gives for an ideal absorption heat pump: $T_{in} = T_{out}$. This result means that the effective heat absorbing and emitting temperatures are the same. In practice, this result means that there is no longer any heat pump but all heat from the heat source (flow C in Fig. 2) to the heat sink (flow A in Fig. 2) is transferred in a heat exchanger where the heat transfer area is infinite and no temperature differences exist ($T_{C1} = T_{A2}$ and $T_{C2} = T_{A1}$). Because the temperature of the flow C is higher than the real environmental temperature T_o , flow C has ability to make work with respect to the real ambient temperature T_o . This is the minimum loss of work that is lost in water heating. Using the symbols in Fig. 2 the minimum loss of work becomes

$$\dot{W}_{loss,min} = \left(1 - \frac{T_o}{T_{A,out}}\right) \cdot (\dot{Q}_{con} + \dot{Q}_{abs}) = \left(1 - \frac{T_o}{T_{A,out}}\right) \cdot (\dot{Q}_{ev} + \dot{Q}_{gen}) \quad (11)$$

where $\dot{W}_{loss,min}$ is the minimum loss of work, T_o is the real ambient temperature and $T_{A,out}$ is the effective heat emitting temperature which is calculated using Eq. (8a).

All real systems generate entropy and therefore the real loss of work is always greater than $\dot{W}_{loss,min}$. Using Eq. (3) the real loss of work becomes

$$\dot{W}_{loss,real} = \dot{W}_{loss,min} + T_{A,out} \delta = \left(1 - \frac{T_o}{T_{A,out}}\right) \cdot (\dot{Q}_{ev} + \dot{Q}_{gen}) + T_{A,out} \delta \quad (12)$$

where $\dot{W}_{loss,real}$ is the real loss of work and δ is the entropy generation rate over the absorption heat pump. For the absorption heat pump in Fig. 2, the entropy generation rate [W/K] is calculated as follows:

$$\delta = \dot{m}_A (s_{A2} - s_{A1}) + \dot{m}_B (s_{B2} - s_{B1}) + \dot{m}_C (s_{C2} - s_{C1}) \quad (13)$$

where $\dot{m}$ is the mass flow rate of the flow and s is the specific entropy. Subscripts A1 – C2 are the same as in Fig. 2. If heat into the evaporator is taken from the ground or sea, temperature T_{B1} may be the same as temperature T_{B2} . In this case the entropy generation rate becomes

$$\delta = \dot{m}_A (s_{A2} - s_{A1}) + \dot{m}_C (s_{C2} - s_{C1}) - \frac{\dot{Q}_{ev}}{T_{B,in}} \quad (14)$$

For a compressor driven heat pump the entropy generation rate can also be calculated using Eq. (13) or Eq. (14). The entropy generation rate must always be a positive one.

For an absorption heat pump the performance parameter that describes the relative improvement potential of the pump can be expressed as a rational efficiency as:

$$\varepsilon = \frac{\dot{W}_{loss,min}}{\dot{W}_{loss,real}} \quad (15)$$

When the rational ratio is defined, Eq. (11) only takes into account the exergy content of flow C, which is the same as the exergy content of flow A but flow B is ignored. The exergy content of flow B is not taken into account, because the system is no longer a heat pump but a heat exchanger where flows A and C only exist.

2.3 Approach 2

Approach 2 is based on the traditional exergy analysis. The performance parameter to be defined is the exergy efficiency. The basic theory of the exergy analysis has been extensively explained for

example by Szargut et al. (1988) or Kotas (1995). This chapter just summarizes necessary equations needed to define the exergy efficiency for both heat pump systems.

2.3.1 Compressor driven heat pump

The exergy efficiency for a compressor driven heat pump is defined as follows (Szargut et al., 1998):

$$\eta_{ex} = \frac{\Delta \dot{E}x_A}{-\Delta \dot{E}x_B - \dot{W}_{in}} \quad (16)$$

where η_{ex} is the exergy efficiency, $\Delta \dot{E}x_A$ is the exergy increase of the heat sink (flow A in Fig. 1), $\Delta \dot{E}x_B$ is the exergy decrease of the heat source (flow B in Fig. 2) and $\dot{W}_{in}$ represents feeding shaft work into the compressor. $\dot{W}_{in}$ has a negative sign which means that $-\dot{W}_{in} > 0$.

The exergy increase $\Delta \dot{E}x_A$ and decrease $\Delta \dot{E}x_B$ are calculated as:

$$\Delta \dot{E}x_A = \dot{m}_A [(h_{A2} - h_{A1}) - T_o (s_{A2} - s_{A1})] \quad (17a)$$

$$\Delta \dot{E}x_B = \dot{m}_B [(h_{B2} - h_{B1}) - T_o (s_{B2} - s_{B1})] - \frac{(T - T_o)}{T} \dot{Q}_{in} \quad (17b)$$

where $\dot{m}_{in,i}$ is the mass flow rate, h is the specific enthalpy, s is the specific entropy and T_o represents temperature at the reference environment. $\dot{Q}_{in}$ denotes the heat flow into the system taken into on the system boundary and T is the temperature of the heat source. In practice, the last term is needed if the heat is taken from a heat source which is at constant temperature (e.g. ground or sea water) and the temperature of which is higher than the real environmental temperature.

2.3.2 Absorption heat pump

For an absorption heat pump, the exergy efficiency is defined as follows:

$$\eta_{ex} = \frac{\Delta \dot{E}x_A}{-\Delta \dot{E}x_B - \Delta \dot{E}x_C} \quad (18)$$

where $\Delta \dot{E}x_A$ and $\Delta \dot{E}x_B$ are calculated using Equations (17a) and (17b), respectively. Term $\Delta \dot{E}x_C$ represents the exergy decrease in the generator and it is calculated in the same way as $\Delta \dot{E}x_B$.

2.4 Case descriptions

Both approaches are applied to base cases shown in Figures 3 and 4. All necessary process values of the base cases are also shown in Figures 3 and 4. Both heat pumps represent a realistic case where heat pumps are used to produce warm water from a low temperature water stream. Flue gas for an absorption heat pump comes from a gas engine that combusts natural gas. Specific heat capacities of water and flue gases are assumed to be constant in all calculations. Pressure losses of flows A, B and C are ignored.

The temperature in the reference environment is 5°C. Pressure and concentration of various species (N_2 , O_2 , CO_2 , ...) in the reference environment are not defined because it is not necessary to define the chemical exergy when heat pumps are analyzed. Concentration of the flue gas is neither needed to know because heat capacities are constant and chemical exergy is not calculated. All heat into the evaporator comes from a low temperature water stream, which means that the term $\frac{(T - T_o)}{T} \dot{Q}_{in}$ is zero for both heat pumps. The evaporation temperature of the refrigerant in Figs. 3 and 4 is so low that the process can always absorb heat in the evaporator at 5°C (the real environmental temperature).

3 Results and discussion

3.1 Compressor driven heat pump

Table 1 shows results of the base case for a compressor driven heat pump (Fig. 3). The rational ratio shows that the theoretical COP could be almost 4 times higher than the existing one. The difference between the $W_{in,real} - W_{in,theoretical}$ represents the exact theoretical improvement potential of the pump when it transfers heat from flow B to flow A. If the number of heat pumps was infinite, the theoretical improvement potential could be achieved (Lampinen and Heikkilä, 1995). In practice, the shaft work could be reduced by heating the flow A stepwise at several stages. Figure 5 illustrates schematically how heating could be carried out stepwise at two stages. The shaft work reduces because less amount of the refrigerant needs to be compressed to the pressure level which still condenses at the temperature of T_{A2} . At single stage heating, all refrigerant needs to be compressed to this pressure level.

Figures 6a and 6b show how the rational ratio and the exergy efficiency change when T_{B1} and T_{B2} change. All other operational values such as cycle process values of the heat pump and T_{A1} and T_{A2} are exactly the same as in the base case (Fig. 3).

Figure 6a shows that the rational ratio and the exergy efficiency get exactly the same value when $T_{B1} = T_{B2} = 5^\circ\text{C}$. In this case, the effective heat absorbing temperature T_{in} is 5°C which is also the real environmental temperature. Because the effective temperature is the same as the environmental temperature, the minimum feeding shaft work is the same for both approaches. In the calculation case, the minimum shaft work would be 9.9 kW, which is obtained by multiplying the real shaft work either by the rational ratio or the exergy efficiency. Szargut et.al. (1998) define the exergy efficiency of a compressor driven heat pump this way. Szargut et.al. (1998) have not

discussed how to define the rational ratio or exergy efficiency if the temperature of the heat source differs from the real environmental temperature.

If the average temperature of the heat source is higher than the real environmental temperature both the rational ratio and the exergy efficiency decline as Fig. 6a and 6b show. Even though both performance parameters become worse, the rational ratio decreases more drastically. This results from the different behavior of irreversibility rates of Approach 1 and 2. Irreversibility rates for Approaches 1 ($T_{in}\delta$) and 2 ($T_o\delta$) as a function of the outlet temperature of the heat source are shown in Fig. 7. For Approach 1, the irreversibility rate is calculated by multiplying the entropy generation rate by the effective heat absorbing temperature. For Approach 2, the irreversibility rate is calculated by multiplying the entropy generation rate by the real environmental temperature. The entropy generation rate is independent on the approach but the effective heat absorbing temperature is higher than the real environmental temperature (see Table 2) which explains the different behavior of the rational ratio and the exergy efficiency.

Table 2 shows the sum of the minimum shaft work and the irreversibility rate for Approach 1. The sum is the same as the real shaft work as Eq. (6) suggests. The result means that the irreversibility rate of approach 1 reveals the exact improvement potential of the system. Instead, Approach 2 which is typically used in exergy analyses underestimates the improvement potential because the temperature T_o in the irreversibility rate is independent on heat source temperature. The heat source temperature only affects the entropy generation rate. This is also the main difference between Approaches 1 and 2. Approach 2 gives the exact improvement potential of the system only in the case where the heat source temperature is the same as the real environmental temperature. It is, however, necessary to state that the average flow temperature used by Ally et.al. (2017) also estimates the correct improvement potential with sufficient accuracy in these calculation cases. For example, for the inlet temperature of 293.15 (T_{B1}) and the outlet temperature of 278.15 K (T_{B2}) the average flow temperature is 285.65 K while the effective heat absorption temperature T_{in} is 285.58 K in our calculations. This is a negligible difference in practical calculations. For bigger temperature changes, the difference between the average and the effective heat absorption temperature increases. Even though the term $T_o\delta$ does not give the exact improvement potential, it also describes the behavior of losses in a correct way. When the entropy generation rate increase, losses also increase and the exergy efficiency becomes worse.

Figure 7 and Table 2 also show that the minimum shaft work reduces as the heat source temperature increases. Higher heat source temperature makes it possible to increase the pressure of the refrigerant in the evaporator, which reduces the work needed in the compressor. As mentioned earlier, process values of the refrigerant are the same for all temperatures of t_{B2} in Fig. 7 and therefore the real shaft work is constant although the effective heat absorbing temperature increases.

Compared to the conventional COP both the rational ratio and the exergy efficiency give a more objective view of the performance of the heat pump. For example, the COP completely ignores the influence of temperatures T_{B1} and T_{B2} on the performance even though both exergy approaches show that these temperatures have an influence on the performance. In particular, this is important

to consider when performance of various heat pumps are compared and temperatures of heat sources and sinks between heat pumps are clearly different.

3.2 Absorption heat pump

Table 3 shows results of the base case for an absorption heat pump (Fig. 4). As mentioned in Methods, the absorption heat pump does not need any mechanical work and therefore it is impossible to calculate any minimum shaft work for the pump. Instead, the exergy analysis assesses how much work potential is lost in the pump when the flow A is heated. In a base case, 9.9 kW of mechanical work is at least lost in water heating. In this case, heating of water occurs in an ideal heat exchanger where the inlet temperature of flow C is 55°C and the outlet temperature 30°C.

All real systems generate entropy and therefore the real loss of work is always greater than $W_{loss,min}$ (see Table 3). Both the rational ratio and the exergy efficiency also assess the improvement potential of the heat pump in more objective way than the COP value. The COP only takes into account if the heat input into the generator changes but ignores the quality of the heat source. The quality of the heat source describes its theoretical working potential with respect to the real ambient temperature. The working potential is the same as the exergy content of the flow. The rational ratio and the exergy efficiency consider both changes in heat inputs and the quality of the heat source. Table 3 does not show any COP value for the theoretical heat pump. This is not meaningful because the theoretical heat pump is not any heat pump but a heat exchanger, even though an unrealistic one.

Table 3 also shows that the exact values for the rational ratio and the exergy efficiency are different. The reason for this is that Approach 1 evaluates the loss of mechanical work with respect to the effective temperature of the flow A (a flow to be heated) while Approach 2 evaluates the loss of work with respect to the real environmental temperature. Because Approach 2 evaluates the loss of work with respect to the environmental temperature, it also considers that flow B (heat source of the evaporator) has potential to make mechanical work. However, this potential is usually small compared to the potential of flow C (heat source of the generator). Approach 1 does not consider this potential at all.

Figures 8a and 8b show how the rational ratio and the exergetic efficiency change when T_{B1} and T_{B2} change. All other operational values are the same as in the base case. Both performance parameters behave in the same way as they do in the case of a compressor driven heat pump. As in the case of the compressor driven heat pump, the entropy generation rate is independent on the approach but the effective heat absorbing temperature is higher than the real environmental temperature explaining the different behavior of the rational ratio and the exergy efficiency. The only difference compared to the compressor driven pump is that the loss of power in Approach 1 is calculated with respect to the effective temperature $T_{A,out}$ instead of $T_{B,in}$.

Figure 9 shows how the rational ratio and the exergy efficiency behave as a function of T_{C1} which is the inlet temperature of the heat source into the generator. In Fig. 9, the outlet temperature of the heat source is always assumed to be 30°C lower than the inlet temperature. Even though the exact values of the rational ratio and the exergy efficiency are not the same they behave in a similar

way. When T_{C1} increases performance parameters become worse, because the entropy generation rate over the heat pump increases. For example, the entropy generation is 0.057 kW/K for the base case. When T_{C1} is 400°C and T_{C2} 370°C the entropy generation rate is already 0.099 kW/K. Figure 9 also shows that the temperatures of T_{C1} and T_{C2} have an influence on the performance of the heat pump. Therefore it is even more important to evaluate the performance of an absorption heat pump using the exergy method than in the case of a compressor driven heat pump. On the basis of this study, both approaches can be used for this purpose.

4 Conclusions

The performance of compressor driven and absorption heat pumps has been evaluated using the rational ratio (Approach 1) and the exergy efficiency (Approach 2) as well as the conventional COP. The rational ratio has been calculated in this paper using effective heat absorbing and emitting temperature while the exergy efficiency has been defined using the real ambient temperature. For both pumps, rational ratios are lower than exergy efficiencies. Even though the exact values are different, both parameters evaluate the change of the performance in the same way. As the entropy generation increases both parameters become worse as they should do. Therefore both approaches can be used when the performance of different heat pumps are compared. For a compressor driven heat pump, the rational ratio is a slightly better performance parameter than the commonly used exergy efficiency which is normally used in exergy studies because it reveals the exact improvement potential of the heat pump. The exergy efficiency reveals the real potential only in cases where the temperature of the heat source is the same as the real environmental temperature. In other cases, it underestimates the potential. It is, however, necessary to state that the average flow temperature of the heat source (for example, used by Ally et.al. (2017)) also approximates the correct improvement potential with sufficient accuracy, at least when the temperature change is relatively small.

Both the rational ratio and the exergy efficiency give a much more realistic view of the performance of the heat pump than the commonly used COP. Especially, for an absorption heat pump the rational ratio or the exergy efficiency should be used instead of the COP because the COP completely ignores the exergy content of the heat source that is used in the generator.

This paper also shows that it is not complex to calculate performance parameters which are based on the exergy analyses. In authors' opinion, the rational ratio and/or the exergy efficiency should be commonly used parameters when the performance of heat pumps is evaluated or compared.

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6 Tables

Table 1. Results of the base case for a compressor driven heat pump

	Theoretical heat pump	Real heat pump
Q_{ev} , kW	76.7	57.1
Q_{con} , kW	84	84
W_{in} , kW	7.3	27.9
Rational ratio	1	0.26
Exergy efficiency	1	0.33
COP	11.5	3.0

Table 2. The sum of minimum shaft work and the irreversibility rate for approach 1, $T_{B1} = 293.15\text{K}$.

T_{B2} °C	T_{in} K	W_{min} , kW	δ kW/K	$W_{min} + T_{in}\delta$ kW	W_{real} kW
278.15	285.58	7.961	0.0698	27.9	27.9
283.15	288.12	7.286	0.0715	27.9	27.9
288.15	290.64	6.614	0.0732	27.9	27.9

$$T_{in} = (T_{B2} - T_{B1})/\ln(T_{B2}/T_{B1})$$

Table 3. Results of the base case for an absorption heat pump

	Theoretical heat pump	Real heat pump
Q_{ev} , kW	-	20.3
Q_{gen} , kW	84	63.7
$Q_{heating}$, kW	84	84
$W_{loss, min}$, kW	9.9	9.9
$W_{loss, real}$, kW	9.9	27.8
Rational ratio	1	0.36
Exergy efficiency	1	0.39
COP	-	1.31

7 Figure captions and figures

Fig. 1. Main components of a compressor driven heat pump.

Fig. 2. Main components of an absorption heat pump, $T_{C,in} > T_{A,out} > T_{B,in}$.

Fig. 3. The base case of a compressor driven heat pump.

Fig. 4. The base case of an absorption heat pump.

Fig. 5. Heating of flow A at two stages.

Fig. 6 a). Rational ratio (ε) and exergetic efficiency (η_{ex}) when the inlet temperature of the heat source (t_{B1}) changes, **6b)** Rational ratio (ε) and exergetic efficiency (η_{ex}) when the outlet temperature of the heat source (t_{B2}) changes.

Fig. 7. Irreversibility rates ($T_{in}\delta$, $T_o\delta$), minimum shaft work (W_{min}) and real shaft work (W_{real}) when the outlet temperature of the heat source changes (t_{B2}).

Fig. 8 a) Rational ratio (ε) and exergy efficiency (η_{ex}) when the inlet temperature of the heat source of the evaporator (t_{B1}) changes, **8b)** Rational ratio (ε) and exergetic efficiency (η_{ex}) when the outlet temperature of the heat source of the evaporator (t_{B2}) changes.

Fig. 9. Rational ratio (ε) and exergy efficiency (η_{ex}) when the inlet temperature of the heat source of the generator (t_{C1}) changes, $t_{C2} = t_{C1} - 30^\circ\text{C}$.

Fig. 1. Main components of a compressor driven heat pump.

Fig. 2. Main components of an absorption heat pump, $T_{C,in} > T_{A,out} > T_{B,in}$.

Fig. 3. The base case of a compressor driven heat pump.

Fig. 4. The base case of an absorption heat pump.

Fig. 5. Heating of flow A at two stages.

Fig. 6 a). Rational ratio (ε) and exergetic efficiency (η_{ex}) when the inlet temperature of the heat source (t_{B1}) changes, **6b)** Rational ratio (ε) and exergetic efficiency (η_{ex}) when the outlet temperature of the heat source (t_{B2}) changes.

Fig. 7. Irreversibility rates ($T_{in}\delta$, $T_o\delta$), minimum shaft work (W_{min}) and real shaft work (W_{real}) when the outlet temperature of the heat source changes (t_{B2}).

Fig. 8 a) Rational ratio (ε) and exergy efficiency (η_{ex}) when the inlet temperature of the heat source of the evaporator (t_{B1}) changes, **8b)** Rational ratio (ε) and exergetic efficiency (η_{ex}) when the outlet temperature of the heat source of the evaporator (t_{B2}) changes.

Fig. 9. Rational ratio (ε) and exergy efficiency (η_{ex}) when the inlet temperature of the heat source of the generator (t_{c1}) changes, $t_{c2} = t_{c1} - 30^\circ\text{C}$.

8 Keywords

Rational ratio, exergy efficiency, irreversibility rate, COP, absorption heat pump, compressor driven heat pump