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EXERGETIC METHOD FOR AIR-SOURCE HEAT PUMPS EFFICIENCY ANALYSIS

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Abstract:According to the Second Law of Thermodynamics, the heat transfer from a low temperature region to a high temperature one requires a work done by the system.In this study exergetic method is used to quantitative determine the exergy losses from all sub-systems of an air-source heat pump (compressor, condenser, expansion valve and evaporator). Also, the system exergetic efficiency or cycle reversibility degree is estimated and energy-exergy flow diagram of the analyzed system is represented.This study is focused on air/water systems as it has many advantages due to are widely used in many applications.
Keywords: air-sourceheat pump, exergy analysis, efficiency, energy-exergy flow diagram

1. INTRODUCTION

Heat pump systems, as well as refrigeration systems, are devices designed to take heat from environment with lower temperature Q_E (named cold source, which is in contact with the system's evaporator) and give it to an environment with higher temperature Q_{Cd-SC} (named hot source, which is in contact with system's condenser), as shown in the scheme from Figure 1 [1]. According to the Second Law of Thermodynamics, this heat transfer from a low temperature region to a high temperature one requires a work done by the system, i.e. compressor mechanical work - W in Figure 1, in which performs a working fluid (refrigerant).

For the vapor compression systems, depending on the cold/hot sources types, heat pumps may be: air/air, air/water or water/water. This study is focused on air/water systems as it has many advantages due to are widely used in many applications [2-6]. As represented in Figure 1, for this type of heat pump system the cold source is outdoor air, while the hot source is heated water which may be used for domestic hot water supply or for radiator heating units [4, 5].

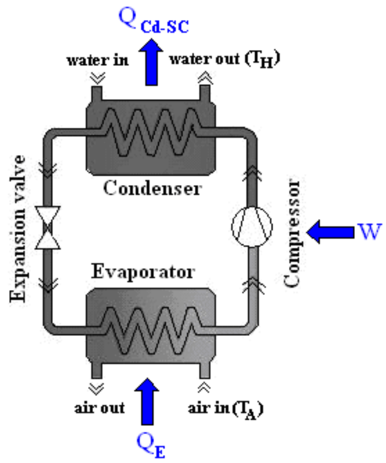


Figure 1. Components and heat flow in an air/water heat pump system [1]

In order to appreciate the performances of heat pump systems energy-exergy analysis is often employed [2-6]. The term exergy was introduced in 1952 by Rant as "energy's ability to perform work", while the other part of the energy is called anergy. In the case of all irreversible (real) processes, exergy is lost. This decrease in exergy is referred to as exergy loss [3].

In this paper, the exergetic method is used to identify the exergy losses for each sub-system of an air/water heat pump system (compressor, condenser, expansion valve and evaporator). So, both internal and external irreversibility of working processes are identified and exergetic efficiency determined. Further, all these quantities are used to represent the energy-exergy flow diagram of the studied system.

2. THERMODYNAMIC ASPECTS REGARDING AIR/WATER HEAT PUMPS

Figure 2 shows heat pump thermodynamic processes represented in both T-s and lg(p)-h diagrams (a and b) [reproduction from ref. 2]. So, the cycle consists of the following processes [2, 6]:

≡ 1-2 adiabatic irreversible compression of the working fluid in gaseous state from evaporation pressure (p_E) and temperature (T_E) to the condensation pressure (p_C) and temperature $T_2>T_C$. During this process the specific entropy increases with $\Delta s_C=s_2-s_1$ [kJ/kg]. Compression 1-2s is an isentropic one (adiabatic reversible process during which the entropy remains constant);

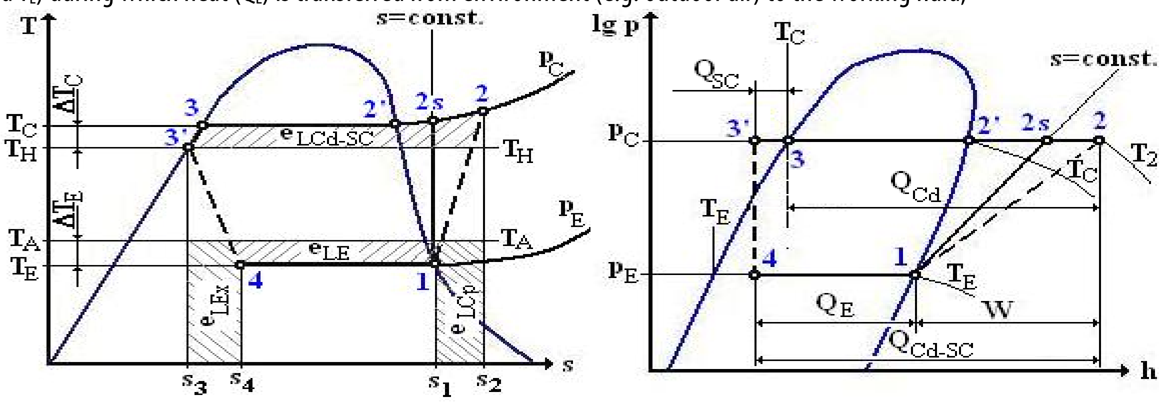
≡ 2-2' isobaric cooling of the working fluid in vapor state from T_2 to condensation temperature T_C;

≡ 2'-3 isobaric-isothermal condensation of the working fluid, process developed at condensation pressure and temperature (p_C and T_C) during which heat (Q_{Cd-SC}) is transferred to environment (e.g. heated water) from the working fluid;

≡ 3-3′ isobaric subcooling of liquid working fluid from condensation temperature T_C to $T_{SC}=T_C-\Delta T_C=T_H$;

≡ 3′-4 adiabatic irreversible expansion of liquid working fluid, followed by pressure and temperature decrease from p_C to p_E and from T_{SC} to T_E. During this process the specific entropy increases with $\Delta s_{Ex}=s_4-s_{3'}$ [kJ/kg];

≡ 4-1 isobaric-isothermal evaporation of the liquid working fluid, process developed at evaporation pressure and temperature (p_E and T_E) during which heat (Q_E) is transferred from environment (e.g. outdoor air) to the working fluid;



a) b)

Figure 2. Vapor compression heat pump cycle with specific exergy losses (T-s Diagram, figure a) and performance parameters (lg p-h Diagram, figure b) [reproduction from ref. 2].

Performance parameters of heat pump systems are [2]:

≡ heat generated during condensation-subcooling process (heating capacity). In T-s diagram this heat corresponds to $(s_2 22'33's_3)$ surface area (see Figure 2a):

$$\left|Q_{Cd-SC}\right|=\left|Q_{Cd}\right|+\left|Q_{SC}\right|=\left(h_2-h_3\right)+\left(h_3-h_{3'}\right)=h_2-h_{3'}\ \text{[kJ/kg]} \tag{1}$$

≡ heat required for evaporation process. In T-s diagram this heat corresponds to $(s_4 41 s_1)$ surface area (see Figure 2a):

$$Q_E=h_1-h_4\ \text{[kJ/kg]} \tag{2}$$

≡ compressor work. In T-s diagram this heat corresponds to $(s_4 41 s_1 s_2 22'33' s_3 s_4)$ surface area (see Figure 2a):

$$\left|W\right|=\left|Q_{Cd-SC}\right|-Q_E=h_2-h_{3'}-\left(h_1-h_4\right)=h_2-h_1\ \text{[kJ/kg]} \tag{3}$$

where $h_4=h_{3'}$ is due to expansion process;

≡ heat pump efficiency:

$$\eta=\frac{\left|Q_{Cd-SC}\right|}{\left|W\right|}=\frac{\left|Q_{Cd-SC}\right|}{\left|Q_{Cd-SC}\right|-Q_E}=\frac{1}{1-Q_E/\left|Q_{Cd-SC}\right|}=\frac{h_2-h_{3'}}{h_2-h_1} \tag{4}$$

≡ heat pump exergetic efficiency:

$$\eta_{ex}=\frac{\left|W_{min}\right|}{\left|W\right|}=1-\frac{\Sigma e_{Lj}}{\left|W\right|}=1-T_A\frac{\Sigma\Delta s_j}{\left|W\right|}=1-\Sigma\overline{e_{Lj}} \tag{5}$$

where: W_{min} [kJ/kg] is minimum compressor work needed to realize the inverse Carnot cycle between cold source temperature, T_A and hot source temperature, T_H (see Figure 2a); $\Sigma\Delta s_j$ [kJ/kg] is the sum of specific entropies changes during j=4 heat pump processes;

$\Sigma e_{Lj}=T_A\Sigma\Delta s_j=e_{LCp}+e_{LEx}+e_{LE}+e_{LC}$ [kJ/kg] is the sum of specific exergy losses due to internal irreversibility of processes taken place in compressor (e_{LCp}) and expansion valve (e_{LEx}) and external irreversibility of heat transfer processes in evaporator (e_{LE}) and condenser (e_{LC}). According to Gouy-Stodola Theorem these specific exergy losses are [6]:

$$e_{LC}=T_A\Delta s_c=T_A\left(s_2-s_1\right)\ \text{[kJ/kg]} \tag{6}$$

$$e_{LEx}=T_A\Delta s_{Ex}=T_A\left(s_4-s_{3'}\right)\ \text{[kJ/kg]} \tag{7}$$

$$e_{LE}=T_A\Delta s_E=\left(\frac{T_A}{T_E}-1\right)Q_E\ \text{[kJ/kg]} \tag{8}$$

$$e_{LCd-SC}=T_A\Delta s_{Cd-SC}=T_A\left(\frac{\left|Q_{Cd-SC}\right|}{T_H}-\left|\Delta s_{Cd-SC}\right|\right)=T_A\left(\frac{\left|Q_{Cd-SC}\right|}{T_H}-\left|s_2-s_{3'}\right|\right)\ \text{[kJ/kg]} \tag{9}$$

3. ENERGY-EXERGY BALANCE OF AIR/WATER HEAT PUMPS: RESULTS AND DISCUSSION

As mentioned, actual operatingconditionsof a heat pumpsystemlead toexergy losses which cause a decrease of system exergeticefficiency. Torevealthe magnitude of these exergy losses the following case study is analyzed: an air/waterheat pump functioning by mechanicalvaporcompression ofR134a working fluid, operatingunder the following conditions:

≡ outdoor air temperature (the entry into evaporator): $T_A = 283K\ (10°C)$;

≡ temperature gradient in the evaporator (temperature difference between air temperature at the entry into evaporator and evaporation temperature): $\Delta T_E = 12 \div 20$ deg. We consider: $\Delta T_E = 15$deg. ;

≡ heated water temperature (at the outlet ofthe condenser): $T_H = 318K\,(45°C)$;

≡ temperature gradient in the condenser (temperature difference between hot water at the outlet of the condenser and condensation temperature): $\Delta T_C = 3 \div 5$deg. We consider: $\Delta T_C = 5\,\text{deg.}$;

≡ temperature gradient of condensate subcooling (temperature difference between condensation temperature and subcooled working fluid temperature in the condenser): $\Delta T_{SC} = 4 \div 7$deg. We consider: $\Delta T_{SC} = 5$ deg. ;

≡ compression process efficiency: $\eta_{Cp} = 0.83$;

≡ mechanical efficiency of the compressor: $\eta_{Cpm} = 0.9$.

To determine the energy-exergy balance withirreversibleworking processesof the heat pump is needed the characteristic temperatures of thermodynamic cycle:

≡ condensation temperature:
$T_C = T_H + \Delta T_C = 323K$;

≡ evaporation temperature:
$T_E = T_A - \Delta T_E = 268\ K$;

≡ condensate subcooling temperature:
$T_{SC} = T_C - \Delta T_{SC} = 318\ K$.

Next, using the characteristic temperatures, the heat pump cycle is drawn in the lg(p)-h diagram using CoolPack software [7]. The operating cycle of the heat pump follows the state points 12s33'41 (see figure 3).

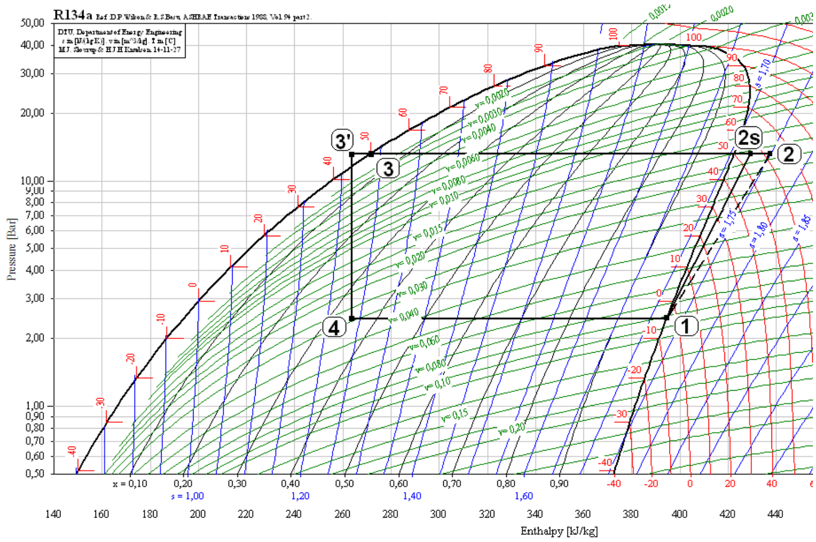


Figure 3. Operating cycle of heat pump represented in lg(p)-h diagram of R134a working fluid.

In Table 1 for each state point of the operating cycle are given the values of temperature T[K], pressure p[bar], specific enthalpy h[kJ/kg] and specific entropy s[kJ/(kg·K)]. These values were taken from the lg(p)-h diagram of R134a working fluid from Figure 3.

Table 1. Characteristic parameters of cycle state points

	T[K]	p[bar]	h[kJ/kg]	s[kJ/(kg·K)]
1	268	2.434	394.284	1.7249
2s	28.37	13.176	429.234	1.7249
3	323	13.176	271.530	1.2367
3'	318	13.176	263.712	1.2131
4	268	2.434	263.712	1.2380

As stated above, in actual operatingconditions the vapor compression process is carried out isentropic (adiabatic irreversible) and corresponds to process 1-2 in Figure 3. So, at compressor outlet (state point 2), specific enthalpy and entropy values may be determined based on the compression process efficiency: $\eta_{Cp} = \frac{h_{2s} - h_1}{h_2 - h_1} \approx \frac{T_E}{T_C} = 0.83$ .So, $h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_c} = 436.3924$ kJ/kg , from Figure 3 resulting $s_2 = 1.746$ kJ/(kg·K) .

In Table 2 are given the results of energy-exergy analysis of the air/water heat pump system.

Table 2. Results of energy-exergy analysis of air/water heat pump

No.	Heat pump sub-system	Specific energy/cycle		Specific exergy loss/cycle	
		Measure	Value [kJ/kg]	Measure	Value [kJ/kg]
1.	Compressor	W	42.10	e_{LCp}	5.9713
2.	Condenser	Q_{Cd-SC}	172.6804	e_{LC}	2.8639
3.	Expansion valve	-	-	e_{LEx}	7.0467
4.	Evaporator	Q_E	130.572	e_{LE}	7.3081

The values of specific exergy losses for all sub-systems of the heat pump are expressed as percentages from the compressor work value W and presented in Figure 4. So, the specific exergy loss of the working fluid in the expansion valve and evaporator are higher. When transferring heat in the evaporator (from ambient air to the evaporating working fluid), due to temperature gradient, a high value of exergy loss is expected [3]. Otherwise, due to small considered temperature gradient in the condenser ($\Delta T_C = 5\text{deg.}$), the value of exergy loss is also small.

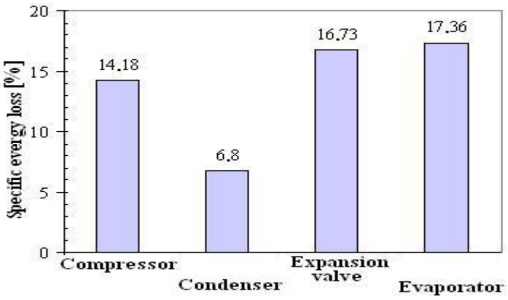


Figure 4. Specific exergy losses calculated for all sub-systems of the heat pump

Further, the following efficiencies are determined: heat pump efficiency (with relation 4): $\eta = 4.10$; heat efficiency for the reference Carnot cycle:

$\eta_C = \dfrac{T_H}{T_H - T_A} = 9.08$, from which results cycle reversibility degree $\mu = \dfrac{\eta}{\eta_C} = 0.45$.

Specific exergy of heat transmitted to water at temperature T_H and specific exergy loss due to this heat transfer irreversibility, e_{LC} , is: $|e| = |W_{min}| = |Q_{Cd-SC}| / \eta_C = 19\ \text{kJ/kg}$ .So, the heat pump exergetic efficiency is: $\eta_{ex} = |e| / |W| = 0.45$.

A verification may be done by: $\eta_{ex} = 1 - \sum_{j=1}^{4} \overline{e_{Lj}} = 0.45$ and $\eta = \eta_{ex} \cdot \eta_C \approx 4.10$.

To the obtained results, the associated energy-exergy flow diagram is shown in Figure 5, from which results that more than half of the work done by the compressor during a thermodynamic cycle (i.e. 55%) is consumed to cover exergy losses caused by the four internal and external irreversibility of processes $\sum_{j=1}^{4} e_{Lj} = 23.1\ \text{kJ/kg}$.

As shown above, the difference $|W| - \sum_{j=1}^{4} e_{Lj}$ represents minimum work required to achieve a reversed Carnot cycle between temperatures $T_A = 283\ \text{K}$ and $T_H = 318\ \text{K}$.

Also, this work ($|e| = 19\ \text{kJ/kg}$) represents the specific heat exergy used for heating water up to temperature of $T_H = 318\ \text{K}$.

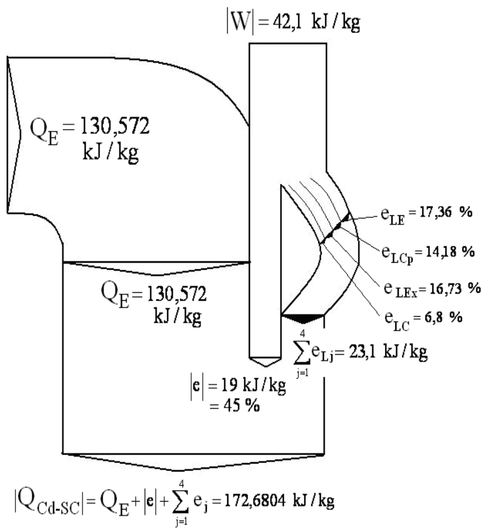


Figure 5. Energy - exergy flow diagram for the air/water heat pump (T_C=323K, T_E=268K).

4. CONCLUSIONS

In order to obtain a complete view of the energy exchanged during heat pumps working, the losses caused by irreversible processes are determined. Moreover, the energy-exergy flow diagram was drawn for the analyzed air/water heat pump system with the following operating conditions: outdoor air temperature of T_A=283K (10°C) and required water heating temperature of T_H=318K (45°C).Also, it was found that for the overall output heat of Q_{Cd-SC}=172.6804 kJ/kg generated by the heat pump system, the thermal efficiency is η=4.10, the reference Carnot cycle efficiency is η_C=9.08, while the cycle reversibility degree or the heat pump exergetic efficiency is η_{ex}=0.45.

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