EXERGY ANALYSIS OF TWO-STAGE WATER TO WATER HEAT PUMP

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Abstract: The paper presents an exergy analysis of the selected two-stage compression water to water heat pump system in heating mode of operation. The foreseen heat source is underground water of constant temperature level. The characteristic exergy values are calculated for the ambient temperature 0 [^{o}C] and ambient pressure 100 kPa. In addition, values of exergy losses, exergy efficiency, exergy loss ratio, exergy loss coefficient and exergy effectiveness (degree of thermodynamic ideality) are also calculated and analyzed. The objective to indicate components and working conditions with increased exergy losses was fulfilled and the obtained results should provide guidance for design and optimization of heat pumps that operate with underground water and are used for water heating applications.

***Key words**: Heat pump, Exergy analysis, Underground water, Heating*

1. INTRODUCTION

Heat pumps are becoming more and more primarily used within the heating systems of newly constructed buildings. Reason for this trend should not be particularly explained considering rising energy price and tendency of improving the energy efficiency of buildings and different systems. Additionaly, utilization of heat pump systems allows for efficient utilization of energy potential of low enthalpy ground waters. A two-stage water to water heat pump is observed in this paper. Calculation model have been taken from the reference [1] and have been run through the “Interactive Thermodynamics” software. Refrigerant R134a was used in the calculation as a work fluid. Heat source was underground water of temperature. extracted by the pump from depth of 100m. Mass flow of water was 2 kg/s. Heat pump was used for water heating in the central heating plant. Analysis of this heat pump system has been carried out only for the case heating the buildings The heating system worked with return water of temperature $T_{PW1} = 40$ [°C] and supply water of temperature $T_{PW2} = 60\left[°\mathrm{C}\right]$. Based on the T_{PW2} temperature, whose value is fixed, condensation temperature of the refrigerant was determined, and its value was $T_{CD} = 65\left[°\mathrm{C}\right]$. While conducting the exergy analysis, the following ambient conditions were used: air temperature $T_a = 0\left[°\mathrm{C}\right]$, atmospheric pressure $p_a = 100\left[\mathrm{kPa}\right]$.

2. CONFIGURATION OF A HEAT PUMP SYSTEM

The heat pump (Figure 1) comprises of two compressors with isentropic efficiency of 0.8. Refrigerant condensation is being conducted in the condenser at the expense of water heating from the temperature T_{PW1} to the temperature T_{PW2}. Inter-cooling of refrigerant between two compressions is taking place in the inter-cooler at the expanse of water heating from the temperature T_{W2} to the temperature T_{GW1}. Primary effects of this inter-cooling are actually compressor power savings, while the secondary effect is the water heating from the condition $W2$ to the condition $GW1$. After passing through the inter-cooler, water of the condition $GW1$ is being taken to the evaporation section of the heat pump, where it gives off heat to the Refrigerant and cooling to the value of T_{GW2}. Water flow through the heat exchangers and its lift from the depth of 100m are made possible by a water pump

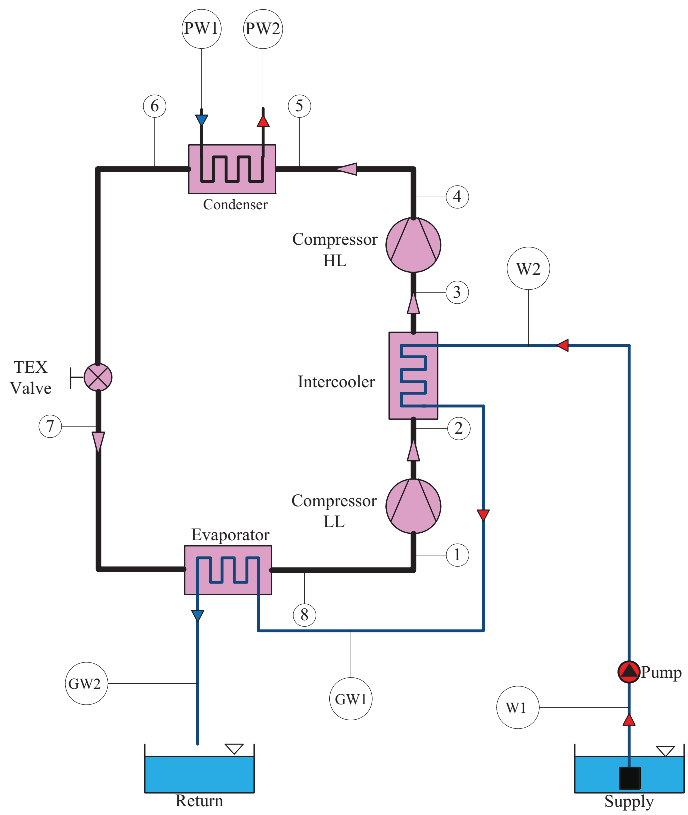

Figure 1 – Two-stage heat pump heating system

3. CALCULATIONS

While conducting the state properties calculation and the exergy analysis as well, certain assumptions have been made. For the Refrigerant evaporation temperature the following value was used: $T_{EV} = T_{W1} - 8[°C]$. It have been assumed that the temperature of water in state $GW2$ is by $3[°C]$ lower than the evaporation temperature. Heat losses to the environment by the system itself were neglected. Changes in kinetic and potential energy of refrigerant and water were also neglected. Refrigerant mass flow has been determined on the basis of evaporator and inter-cooler energy balance. On the other hand, water mass flow through the central heating system has been determined based on the desired change in water temperature in the condenser itself. It was assumed that the water temperature in the state $W2$ was equal to the water temperature in the state $W1$. Within the calculation itself, it has been assumed that the Refrigerant at the intakes of both compressors is in the sate of saturated vapor (state 1 and state 3). Intermediate pressure between two compressions was determined based on demand for maximum compressor power savings [3]. The exergy of refrigerant in certain states has been calculated on the basis of known expressions and proper magnitudes have been determined in detail and shown in Table 1, while the exergy indicators have been shown in Table 2.

4. EXERGY ANALYSIS

Exergy analysis of the evaporator

Exergy loss rate:

Exergy loss realized in some installation component defines the difference between the introduced exergy and the exergy which is leaving the same system component,

$$\left(\dot{E}x\right)_{loss,R} = \left(\dot{E}x\right)_{flow,in} - \left(\dot{E}x\right)_{flow,out} = (\dot{E}x_7 + \dot{E}x_{GW1}) - (\dot{E}x_8 + \dot{E}x_{GW2}) .$$

Exergy efficiency:

Exergy efficiency defines the ratio between desired exergy change and exergy change, or exergy input by which this change has been accomplished,

$$\eta_R = \left(\dot{E}x\right)_{desired,out} / \left(\dot{E}x\right)_{used} = \left(\dot{E}x_8 - \dot{E}x_7\right) / \left(\dot{E}x_{GW1} - \dot{E}x_{GW2}\right).$$

Exergy loss ratio:

Exergy loss ratio defines the ratio between exergy loss of the component and exergy loss of the entire heat pump system,

$$d_R = \left(\dot{E}x\right)_{loss,R} / \left(\dot{E}x\right)_{loss,sys} .$$

Exergy loss coefficient:

Exergy loss coefficient defines the ration between exergy loss of the component and power used within the system,

$$\lambda_R = \left(\dot{E}x\right)_{loss,R} / P_i \,.$$

Thermodynamic perfect degree:

Thermodynamic perfect degree defines the ratio between conserved and exergy introduced to the system,

$$\varepsilon_R = \left(\dot{E}x\right)_{out} / \left(\dot{E}x\right)_{in} = \left(\dot{E}x_8 + \dot{E}x_{GW2}\right) / \left(\dot{E}x_7 + \dot{E}x_{GW1}\right).$$

All of the expressions, described in the previous section and whose results are given in detail in the Table 1 and Table 2, are shown below.

Exergy analysis of the condenser

Exergy loss rate:

$$\left(\dot{E}x\right)_{loss,C} = \left(\dot{E}x\right)_{flow,in} - \left(\dot{E}x\right)_{flow,out} = (\dot{E}x_5 + \dot{E}x_{PW1}) - (\dot{E}x_6 + \dot{E}x_{PW2})$$

Exergy efficiency:

$$\eta_C = \left(\dot{E}x\right)_{desired,out} / \left(\dot{E}x\right)_{used} = \left(\dot{E}x_{PW2} - \dot{E}x_{PW1}\right) / \left(\dot{E}x_5 - \dot{E}x_6\right)$$

Exergy loss ratio:

$$d_C = \left(\dot{E}x\right)_{loss,C} / \left(\dot{E}x\right)_{loss,sys}$$

Exergy loss coefficient:

$$\lambda_C = \left(\dot{E}x\right)_{loss,C} / P_i$$

Thermodynamic perfect degree:

$$\varepsilon_C = \left(\dot{E}x\right)_{out} / \left(\dot{E}x\right)_{in} = \left(\dot{E}x_6 + \dot{E}x_{PW2}\right) / \left(\dot{E}x_5 + \dot{E}x_{PW1}\right)$$

Exergy analysis of the compressors

Exergy loss rate:

$$\left(\dot{E}x\right)_{loss,CLL} = \left(\dot{E}x\right)_{flow,in} - \left(\dot{E}x\right)_{flow,out} = (\dot{E}x_1 + P_{CLL}) - \dot{E}x_2$$

$$\left(\dot{E}x\right)_{loss,CHL} = \left(\dot{E}x\right)_{flow,in} - \left(\dot{E}x\right)_{flow,out} = (\dot{E}x_3 + P_{CHL}) - \dot{E}x_4$$

Exergy efficiency:

$$\eta_{CLL} = \left(\dot{E}x\right)_{desired,out} / \left(\dot{E}x\right)_{used} = \left(\dot{E}x_2 - \dot{E}x_1\right) / P_{CLL}$$

$$\eta_{CHL} = \left(\dot{E}x\right)_{desired,out} / \left(\dot{E}x\right)_{used} = \left(\dot{E}x_4 - \dot{E}x_3\right) / P_{CHL}$$

Exergy loss ratio:

$$d_{CLL} = \left(\dot{E}x\right)_{loss,CLL} / \left(\dot{E}x\right)_{loss,sys}$$

$$d_{CHL} = \left(\dot{E}x\right)_{loss,CHL} / \left(\dot{E}x\right)_{loss,sys}$$

Exergy loss coefficient:

$$\lambda_{CLL} = \left(\dot{E}x\right)_{loss,CLL} / P_i$$

$$\lambda_{CHL} = \left(\dot{E}x\right)_{loss,CHL} / P_i$$

Thermodynamic perfect degree:

$$\varepsilon_{CLL} = \left(\dot{E}x\right)_{out} / \left(\dot{E}x\right)_{in} = \dot{E}x_2 / \left(\dot{E}x_1 + P_{CLL}\right)$$

$$\varepsilon_{CHL} = \left(\dot{E}x\right)_{out} / \left(\dot{E}x\right)_{in} = \dot{E}x_4 / \left(\dot{E}x_3 + P_{CHL}\right)$$

Exergy analysis of the intercooler

Exergy loss rate:

$$\left(\dot{E}x\right)_{loss,IC} = \left(\dot{E}x\right)_{flow,in} - \left(\dot{E}x\right)_{flow,out} = (\dot{E}x_2 + \dot{E}x_{W2}) - (\dot{E}x_3 + \dot{E}x_{GW1})$$

Exergy efficiency:

$$\eta_{IC} = \left(\dot{E}x\right)_{desired,out} / \left(\dot{E}x\right)_{used} = \left(\dot{E}x_{GW1} - \dot{E}x_{W2}\right) / \left(\dot{E}x_2 - \dot{E}x_3\right)$$

Exergy loss ratio:

$$d_{IC} = \left(\dot{E}x\right)_{loss,IC} / \left(\dot{E}x\right)_{loss,sys}$$

Exergy loss coefficient:

$$\lambda_{IC} = \left(\dot{E}x\right)_{loss,IC} / P_i$$

Thermodynamic perfect degree:

$$\varepsilon_{IC} = \left(\dot{E}x\right)_{out} / \left(\dot{E}x\right)_{in} = \left(\dot{E}x_2 + \dot{E}x_{W2}\right) / \left(\dot{E}x_3 + \dot{E}x_{GW1}\right)$$

Exergy analysis of the expansion valve

Exergy loss rate:

$$\left(\dot{E}x\right)_{loss,EXV} = \left(\dot{E}x\right)_{flow,in} - \left(\dot{E}x\right)_{flow,out} = \dot{E}x_6 - \dot{E}x_7$$

Exergy loss ratio:

$$d_{EXV} = \left(\dot{E}x\right)_{loss,EXV} / \left(\dot{E}x\right)_{loss,sys}$$

Exergy loss coefficient:

$$\lambda_{EXV} = \left(\dot{E}x\right)_{loss,EXV} / P_i$$

Thermodynamic perfect degree:

$$\varepsilon_{EXV} = \left(\dot{E}x\right)_{out} / \left(\dot{E}x\right)_{in} = \dot{E}x_7 / \dot{E}x_6$$

Total exergy loss rate

Exergy loss of the system defines the sum of all exergy losses within the systems:

$$\left(\dot{E}x\right)_{loss,sys} = \left(\dot{E}x\right)_{loss,R} + \left(\dot{E}x\right)_{loss,C} + \left(\dot{E}x\right)_{loss,IC} + \left(\dot{E}x\right)_{loss,CLL} + \left(\dot{E}x\right)_{loss,CHL} + \left(\dot{E}x\right)_{loss,EXV}$$

Input power

$$P_i = P_{Pump} + P_{CLL} + P_{CHL}$$

where is: P_{Pump} **-** power of the water pump, P_{CLL} **-** power of the low level compressor,

P_{CHL} - power of the high level compressor.

Power of the low pressure compressor and power of the high pressure compressor were determined on the basis of the known refrigerant enthalpy values before and after the compressor. Power of the supply pump has been calculated on the basis of the required geodesic altitude which have to be overcome by the pump (100m).

Table 1

Exergy Analysis Heat Pump Systems

State	Description	Fluid	Phase	Temperature	Temperature	Pressure	Specific enthalpy	Specific entropy	Mass flow rate	specific exergy	Exergy rate
				°C	K	bar	kJ/kg	kJ/kgK	kg/s	kJ/kg	kW
0	refrigerant	refrigerant	dead state	0	273.2	1	253	1.023	0	0.00	0.0
0w	water	water	dead state	0	273.2	1	0	0	0	0.00	0.0
1	compresor inlet	refrigerant	vapor	8	281.2	3.876	251.8	0.915	0.4205	28.30	11.9
2	compressor outlet-intercooler inlet	refrigerant	vapor	39.86	313.0	8.557	272.2	0.928	0.4205	45.15	19.0
3	intercooler outlet-compressor inlet	refrigerant	vapor	33.72	306.9	8.557	265.3	0.9059	0.4205	44.29	18.6
4	compressor outlet	refrigerant	vapor	70.25	343.4	18.89	285.1	0.9175	0.4205	60.92	25.6
5	condenser inlet	refrigerant	vapor	70.25	343.4	18.89	285.1	0.9175	0.4205	60.92	25.6
6	condenser outlet-valve inlet	refrigerant	liquid	65	338.2	18.89	145.7	0.5056	0.4205	34.03	14.3
7	valve outlet-evaporator inlet	refrigerant	mixture	8	281.2	3.876	145.7	0.5377	0.4205	25.26	10.6
8	evaporator outlet	refrigerant	vapor	8	281.2	3.876	251.8	0.915	0.4205	28.30	11.9
GW1	evaporator inlet	water	liquid	16.35	289.5	2	67.56	0.2402	2	1.95	3.9
GW2	evaporator outlet	water	liquid	11	284.2	1.5	45.26	0.1624	2	0.90	1.8
PW1	condenser inlet	water	liquid	40	313.2	2.5	167.2	0.571	0.69	11.23	7.8
PW2	condenser outlet	water	liquid	60	333.2	2.5	251.6	0.8323	0.69	24.26	16.8
W2	intercooler inlet	water	liquid	16	289.2	2	66.12	0.2352	2	1.88	3.8
W1	pump inlet	water	liquid	16	289.2	1	66.12	0.2352	2	0.89	1.8

Table 2

Exergy Analysis Heat Pump Systems

Component of the system	Exergy loss	Exergy efficiency	Exergy loss ratio	Exergy loss coefficient	Thermodynamic perfect degree
	Ex,loss	η	d	λ	ε
Compressor Low Level	1.49	0.826	0.15	0.08	0.93
Intercooler	0.21	0.409	0.02	0.01	0.98
Compressor High Level	1.33	0.840	0.14	0.07	0.95
Condenser	2.26	0.800	0.23	0.12	0.93
Expansion valve	3.69	-	0.38	0.20	0.74
Evaporator	0.82	0.609	0.08	0.04	0.94
HEAT PUMP System	9.81	0.38	1.00	0.52	0.66

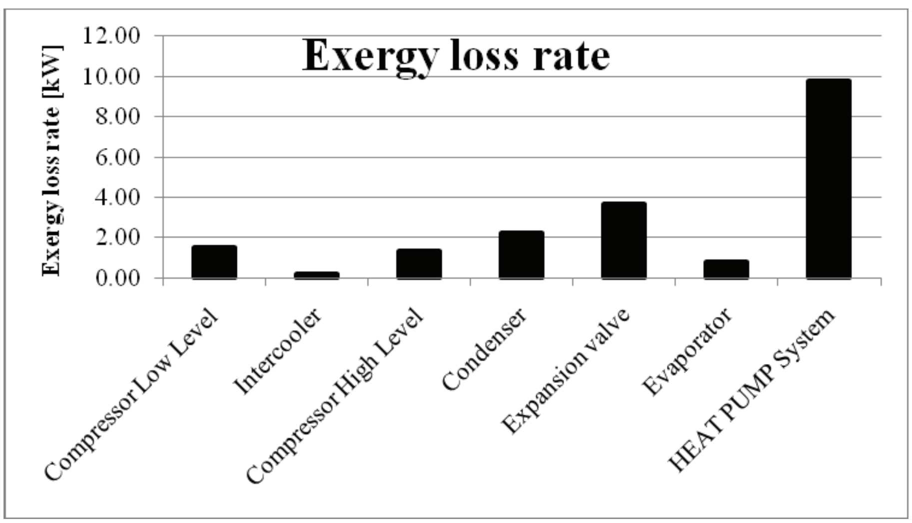

Figure 2 Exergy loss rate of equipment and system

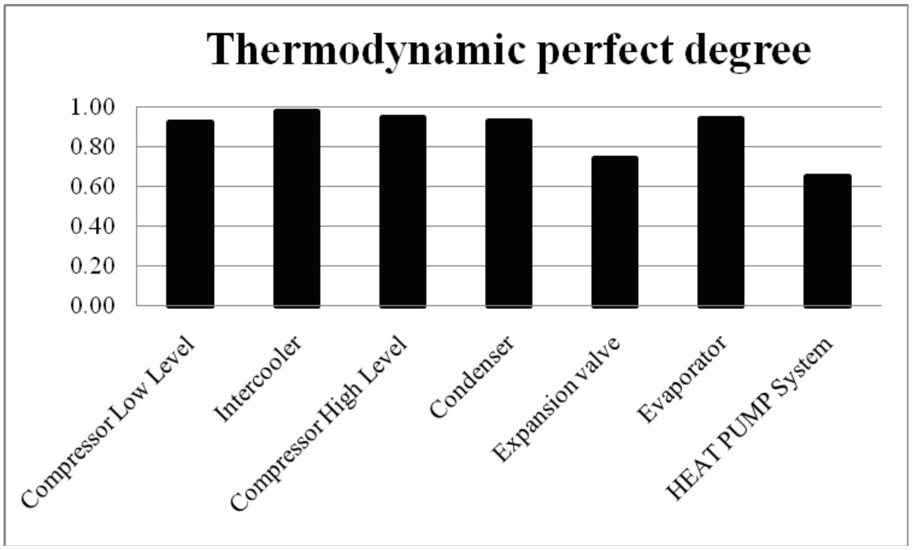

Figure 3 Thermodynamic perfect degree of the heat pump system and utilized components

5. CONCLUSION

Exergy analysis provides the possibilty to overview the entire system from the energy loss point of view and to determine and locate the hot spots when it comes to those losses. The approach enables evaluation of each component and process from the standpoint of its contribution to the overall exergy losses of the observed heat pump heating system. The analysis resulted with the values of thermodynamic perfect degree of the observed heat pump system and its components. Based upon calculated values and the results of the calculation shown in Tables 1 and 2, and in Figures 2 and 3, it was possible to make the following conclusions: Largest exergy losses and exergy loss ratio were at the expansion valve, while the smallest exergy loss and exergy loss ratio were in the inter-cooler. The lowest thermodynamic perfect degree was at the expansion valve and the largest was in the inter-cooler.

6. NOMENCLATURE

d-exergy loss ratio

P-power [kW]

p-pressure [bar]

T-temperature [°C]

$\dot{E}x$ *- rate of exergy*

η *-exergy efficiency*

λ *-exergy loss coefficient*

ε *-thermodynamic perfect degree*

Subscripts:

a-environment

in-input

out-output

sys-system

W-water

PW- plant water

GW-ground water

R-refrigerator

C-condenser

CLL-compressor low level

CHL-compressor high level

IC-intercooler

EXV-expansion valve

i-introduce

7. REFERENCES

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