ASSESSMENT OF IRREVERSIBLE HEAT PUMP-BASED HEATING SYSTEMS

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Abstract. The paper compares the performances of two irreversible heat pump-based heating systems operating in variable conditions and, controlled by the variable temperatures of the heat sources (air or geothermal) and the variable heat loss of the heated space depending on the temperature of the environment. The first irreversible heating system uses an air (environment) to air (heated space) heat pump and, the second one employs a geothermal heat pump. For the temperature of the environment was adopted a scenarios along the heating period (winter) and for the geothermal heat source was employed an analytical solution describing the underground soil temperature evolution in time. The assessment of two irreversible heat pump-based heating systems evaluates the heat rates, the powers and, the COP by employing the entropy balance equation of the whole heat pump-based heating system.

Keywords: entropy balance equation; heat pump-based heating system

1. INTRODUCTION

Because of large heating efficiency and environmental protection, the interest in heat pump based heating systems is increasing today. There is a wide variety of these systems. For heating most heat pump systems use closed loop. The study [1]1, regarding the office building heating systems using ground coupled heat pumps with different borehole diameters, showed that the highest savings to investment ratio, during a period of thirty years, is 4.80. The paper [2] presents an open loop system that uses for cooling a water lake heat capacity. The thermodynamic analysis of ground-coupled heat pump based energy systems emphasizes the important influence of underground soil temperature profile and the geometry of the heat exchangers extracting the geothermal energy [4]. A general review [3], presenting the most common heat pumps based energy systems that are used today, is comparing their performances and operational parameters ***but for quasi-steady state operation***.

This paper compares the performances of two irreversible heat pump-based heating systems operating in variable conditions and, controlled by the variable temperatures of the heat sources (air or geothermal) and the variable heat loss of the heated space depending on the temperature of the environment. The first irreversible heating system uses an air (environment) to air (heated space) heat pump and, the second one employs a geothermal heat pump. For the temperature of the environment was adopted a scenarios along the heating period (winter) and for the geothermal heat source was employed an analytical solution describing the underground soil temperature evolution in time. The assessment of two irreversible heat pump-based heating systems evaluates all energy transfers and, the COP by employing the entropy balance equation of the whole heat pump-based heating system. The heat pump based heating systems are presented in figure 1.

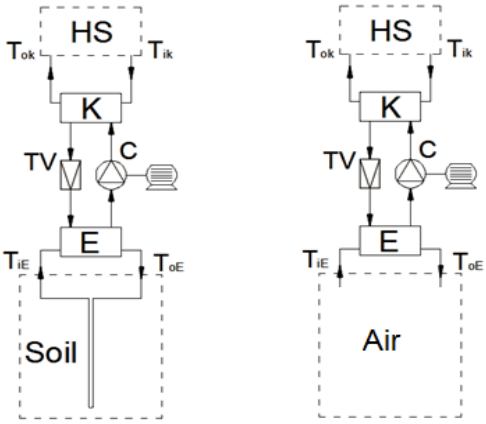

Fig. 1. The heating system scheme. HS – heated space; E – evaporator; K – condenser; C – compressor; TV – throttling valve; T_{iK} and T_{oK} – input and output temperatures for K; T_{iE} and T_{oE} – input and output temperatures for E.

2. MATHEMATICAL MODEL

2.1. Restrictive conditions

- The heated space temperature T_{HS} = 295.15K is constant.
- The ambient temperature is varying, 253.15K ≤ T_a ≤ 288.15K. For T_a it was considered during the winter (4320 hours) the time dependence (see figure 2):

$$T_a = 288.15 - 35 \cdot \sin\left(\frac{\pi \cdot t}{4320}\right) \qquad (1)$$

where t is the time in hours.

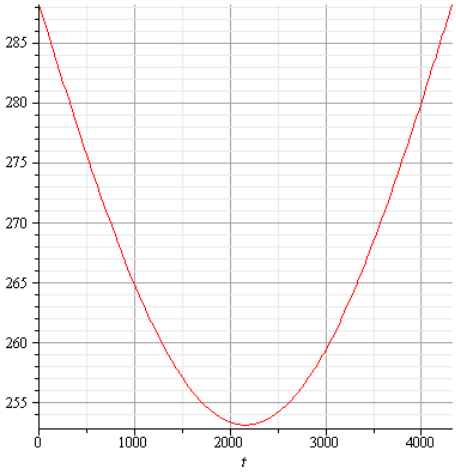

Fig. 2. $T_a = T_a(t)$

- The soil temperature time dependence was evaluated on preliminary modelling of continuous geothermal energy extraction (see figure 3):

$$T_S = 283.15 \cdot \exp\left(-0.0025 \cdot t^{0.375}\right) \qquad (2)$$

where t is the time in hours.

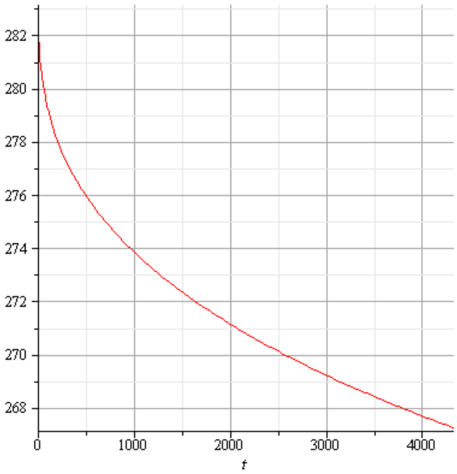

Fig. 3. $T_S = T_S(t)$

- Constant heat carriers heat capacities:
  - between HS and K, $\dot{C}_K$ = 500 W/K
  - between heat source (soil or environmental air) and E, $\dot{C}_E$ = 1000 W/K
- Heat pump heat exchangers (E, K and soil borehole) were presumed to have constant effectiveness (ε_E = 0.75 = const., ε_K = 0.75 = const., and ε_S = 0.95 = const.).

The heating system uses the air as heat carrier flowing in a loop between the condenser and the heated space. The air to air heat pump uses moreover air as heat carrier flowing in a loop between the evaporator and the environment.

2.2. Mathematical algorithm

The heated space ÷ condenser

The heated space needs a heating rate supplied by K equalizing the lost heat rate, see figure 4.

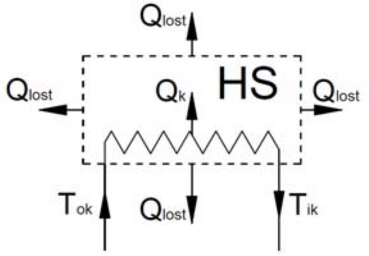

Fig.4. Thermal energy balance of the heated space

The heat balance equations:

$$\dot{Q}_{lost} = h_{HS} \cdot A_{HS} \cdot \left(T_{HS} - T_a\right) = \dot{Q}_K = \dot{C}_K \cdot \left(T_{oK} - T_{iK}\right) = \varepsilon_K \cdot \dot{C}_K \cdot \left(T_K - T_{iK}\right) \qquad (3)$$

$$\frac{\dot{Q}_{lost}\left(T_a\right)}{\dot{Q}_{lost}\left(T_{aN}\right)} = \frac{h_{HS} \cdot A_{HS} \cdot \left(T_{HS} - T_a\right)}{h_{HS} \cdot A_{HS} \cdot \left(T_{HS} - T_{aN}\right)} = \frac{\dot{Q}_K\left(T_a\right)}{\dot{Q}_{KN}\left(T_{aN}\right)} = \frac{\dot{C}_k \cdot \left(T_{oK} - T_{iK}\right)}{\dot{C}_k \cdot \left(T_{oKN} - T_{iKN}\right)} \qquad (4)$$

When T_a changes between +15°C and –20°C, the air input/output temperatures are: $T_{iK} = T_{iKN} = T_{HS}$ = 295.15K = const. and $T_{oK} = T_{oK}(T_a)$ function of the heating system design temperatures. It was imposed T_{oKN} = 313.15 K at T_{aN} = 253.15 K.

The dependences $\dot{Q}_K = \dot{Q}_K\left(T_a\right)$ and, $T_K = T_K(T_a)$ and, $T_{oK} = T_{oK}(T_a)$ are obtained using heat balance equations 1, 3 and 4, see figure 5:

$$T_{oK}\left(T_a\right) \cong T_{iK} + \frac{T_{HS} - T_a}{T_{HS} - T_{aN}}\left(T_{oKN} - T_{iKN}\right) = 421.643 - \frac{3}{7} T_a \qquad (5)$$

$$T_K = T_{iK} + \frac{T_{oK}\left(T_a\right) - T_{iK}}{\varepsilon_K} \qquad (6)$$

$$\dot{Q}_K = \dot{C}_K \cdot \left[T_{oK}\left(T_a\right) - T_{iK}\right] = \varepsilon_K \cdot \dot{C}_K \cdot \left[T_K\left(T_a\right) - T_{iK}\right] \qquad (7)$$

The soil ÷ evaporator (geothermal heat pump)

The heat balance equations are:

$$\dot{Q}_E = \dot{C}_E \cdot \left(T_{iE} - T_{oE}\right) = \varepsilon_E \cdot \dot{C}_E \cdot \left(T_{iE} - T_E\right) \quad (8)$$

$$\dot{Q}_S = \dot{C}_E \cdot \left(T_{iE} - T_{oE}\right) = \varepsilon_S \cdot \dot{C}_E \cdot \left(T_S - T_{oE}\right) = \dot{Q}_E \quad (9)$$

From equations 2, 8, and 9 it yields:

$$T_{iE} = \frac{\varepsilon_E \cdot \varepsilon_S \cdot T_E - \varepsilon_E \cdot T_E - \varepsilon_S \cdot T_S}{\varepsilon_E \cdot \varepsilon_S - \varepsilon_E - \varepsilon_S} \quad (10)$$

$$T_{oE} = \frac{\varepsilon_E \cdot \varepsilon_S \cdot T_S - \varepsilon_E \cdot T_E - \varepsilon_S \cdot T_S}{\varepsilon_E \cdot \varepsilon_S - \varepsilon_E - \varepsilon_S} \quad (11)$$

$$\dot{Q}_E = \frac{\dot{C}_E \cdot \varepsilon_E \cdot \varepsilon_S \cdot \left(T_E - T_S\right)}{\varepsilon_E \cdot \varepsilon_S - \varepsilon_E - \varepsilon_S} \quad (12)$$

The soil ÷ evaporator (air to air heat pump)

The heat balance equation is:

$$\dot{Q}_E = \dot{C}_E \cdot \left(T_{iE} - T_{oE}\right) = \varepsilon_E \cdot \dot{C}_E \cdot \left(T_{iE} - T_E\right) \quad (13)$$

From equations 1, and 13 it yields:

$$T_{iE} = T_a \quad (14)$$

$$T_{oE} = T_a - \varepsilon_E \cdot \left(T_a - T_E\right) \quad (15)$$

Heat pump irreversibility

For the irreversible heat pump cycle the entropy balance equation becomes [5,6]:

$$\frac{\dot{Q}_E}{T_E} Irr - \frac{\dot{Q}_K}{T_K} = 0 \quad (16)$$

$$\text{where } Irr = \frac{(COP+1)}{COP} \frac{T_E}{T_K} = COPR \frac{T_E}{T_K} \quad (17)$$

By using CoolPack we get for R717 (ammonia):

$$COPR = 1.5672 - 2.1225 \cdot \frac{T_K}{T_E} + 1.56564 \left(\frac{T_K}{T_E}\right)^2 \quad (18)$$

The entropy balance equation gives the reference temperature $T_E(t)$. This allows to evaluate T_{iE}, T_{oE} and $\dot{Q}_E$.

Power and, first law efficiency

$$\left|\dot{W}\right| = \dot{Q}_K - \dot{Q}_E \quad (19)$$

$$COP = \frac{\dot{Q}_K}{\left|\dot{W}\right|} \quad (20)$$

3. NUMERICAL RESULTS

The simulation was completed for a period of 4320 hours (180 days). All cycle features, energy exchanges, temperatures, are based on solving the entropy balance equation.

For the two heat pump based heating systems the time dependence of system parameters are shown in next figures.

For ground coupled heat pump system, heat carrier temperature output from condenser and temperature inside condenser are higher when ambiental temperature is lowest (see figure 5) and heat rates in evaporator and condenser are also highest when ambiental temperature is lowest (see figure 6).

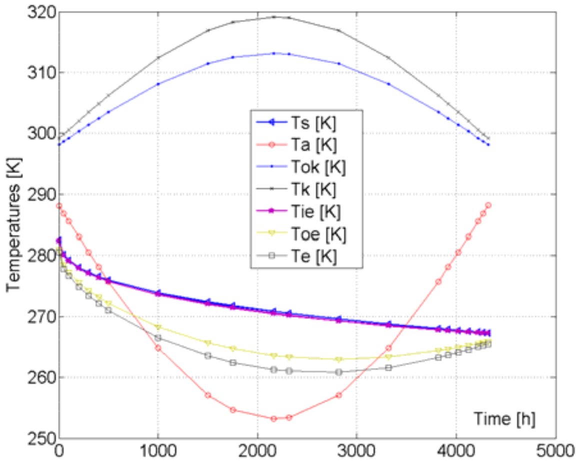

Fig. 5. Operational temperatures function of time (T_S, T_a, T_{oK}, T_K, T_{iE}, T_{oE}, T_E) – geothermal heat pump

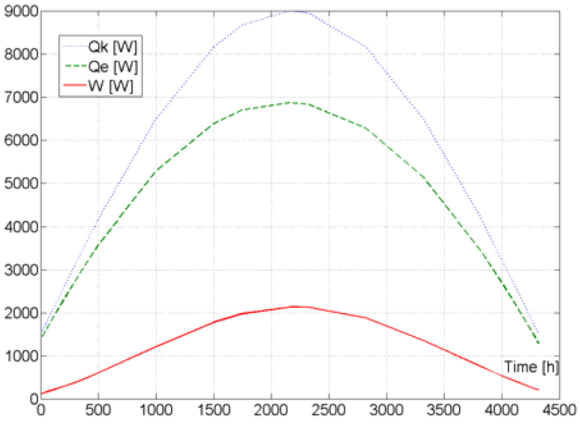

Fig. 6. Heat rates in evaporator and condenser and, power function of time – geothermal heat pump

COP of the ground coupled heat pump system starts from 12.86 and drops to 4.21 during the lowst ambiental temperature, but when ambiental temperature increases again after 180 days, maximm value of COP will be 7.12 (see figure 7). This evolution of COP shows that it is related to soil temperature, that at the end of our simulation drops to 267.27K.

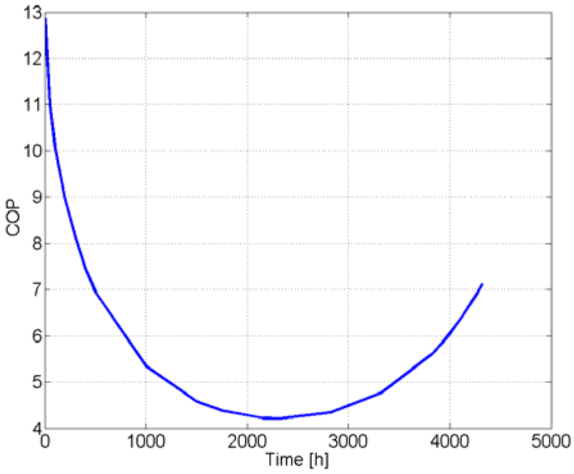

Fig. 7. Heat pump COP function of time for geothermal heat pump

For air to air heat pump system, temperatures are related to each other (see figure 8), heat rates are almost the same during the coolest period, but higher for positive ambiental temperatures (see figure 9). This leads to a COP of 18 at the start and end of the simulation. The work done by air to air heat pump for the lowest ambiental temperature is higher than the work done by the ground heat pump system, while heat rates are aproximately equal. This shows a COP for air to air heat pump system that is lower for the lowest ambiental temperature than the COP of the ground coupled heat pump system (see figure 10).

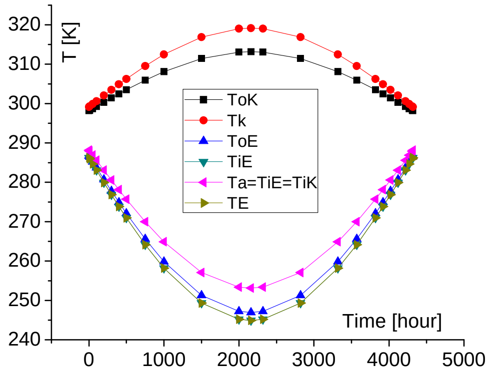

Fig. 8. Operational temperatures function of time (T_a, T_{oK}, T_K, T_{oE}, T_{iE}, T_E) – air to air heat pump

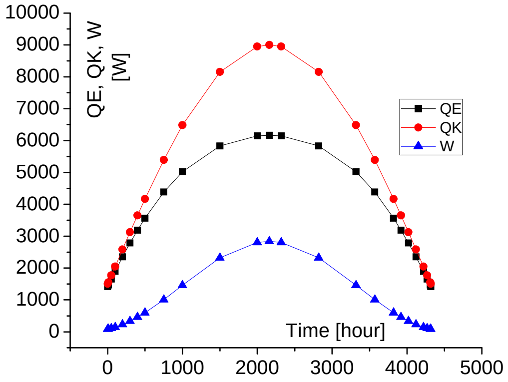

Fig. 9. Heat rates in evaporator and condenser and, power function of time – air to air heat pump

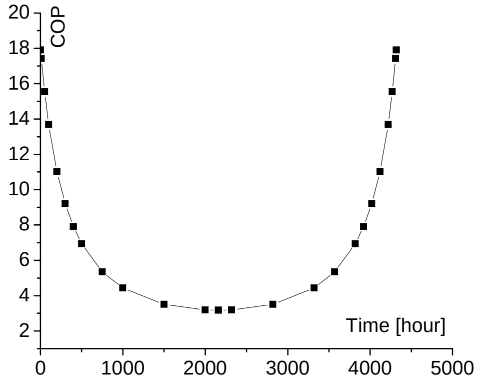

Fig. 10. Heat pump COP function of time for air to air heat pump

4. CONCLUSIONS

The paper deals with entropy balance equation in order to evaluate the time dependent performances of heating system, during the winter season, involving either a geothermal heat pump or air to air heat pump. The thermodynamic assessment considered restrictive conditions describing imposed time based evolution of heat pump external heat “reservoirs”. The entropy balance equation of the irreversible heat pump allowed complete linking of all variable in time parameters, temperatures, energy transfer rates and COP. The mathematical model of this assessment might be applied to other real time based restrictive conditions. The heat pump based heating systems seem to be very competitive to cogeneration based ones. Thus, for this assessment, we must consider the first law efficiency related to the fossil based heat energy operating the engine delivering the power for the heat pump, respectively:

$$FLE_{HP} = COP \cdot FLE_{engine}$$

Here FLE_{engine} is the first law efficiency of the engine delivering the power to the heat pump, from 0.35 (mean value) to 0.59 for combined cycles.

Nomenclature
T – temperature [K]
h – convection heat transfer coefficient [W/(m^{2}K)]
$\dot{Q}$ – heat rate [W]
$\dot{W}$ – power [W]
ε – heat exchanger efficiency [-]
$\dot{m}$ – mass flow [kg/h]
$\dot{C}$ – heat capacity of heat carrier [W/K]
Irr – irreversibility
COP – coefficient of performance

Subscripts:

a – ambient	i – input
E – evaporator	N – nominal
K – condenser	S – soil
o – output	HS – heated space

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