Techno-Economic Analysis of the Performances of High-Temperature Heat Pumps

MILOVAN JOTANOVIĆ*, GORAN TADIĆ*, JURIJ KROPE**, DARKO GORICANEC**

*Faculty of Technology, University of East Sarajevo
Karakaj bb, 75400 Zvornik, BOSNIA AND HERZEGOVINA
**Faculty of Chemistry and Chemical Engineering, University of Maribor
Smetanova ul. 17, 2000 Maribor, SLOVENIA
jotanovicm@mail.ru, gtadic.tf@gmail.com, jurij.krope@uni-mb.si, darko.goricanec@uni-mb.si

Abstract: - The authors of the paper are giving a detailed description of the operation principles of a one-stage and two-stage high-temperature heat pump. A computer programme was used to process the dependences of different evaporation temperatures regarding the heat flow, the heat pump strength and COP. An economic assessment of the investment into different heat pump designs at different evaporation temperatures of refrigerants was also conducted.

Key-Words: - Heat transfer, high temperature heat pumps, economic analysis

1 Introduction

For a long time now, people have experienced the negative consequences of the greenhouse effect. At various conferences and forums scientists are establishing that the only possible method of reducing air pollution is the efficient use of energy together with the development of new technologies and systems, as well as the use of renewable energy sources.

The consequences of an intensive exploitation of energy and energy dependence refer decidedly to the countries with poor sources of energy and which are classified into a group of the most environmentally endangered countries in Europe. The complicated situation in these countries does not allow them to deal with energy issues by simply fulfilling demands and desires of energy consumers.

When it comes to the operation of an energy plant, there is often a dilemma how to improve specific consumption of energy in processes, how to increase efficiency, how to direct and convert energy more efficiently, how to use waste heat and replace combustion of liquid and gaseous fuels with other sources of energy.

Researches have come up with an innovative solution of using waste low-temperature water (45°C) for the purpose of long-distance heating of buildings at the temperature of 90/70°C. In this respect, the Faculty of Chemistry and Chemical Engineering of University of Maribor developed an innovative high-temperature heat pump (HP) made by a Japanese company named Mycom in 2010 [1, 2, 3]. This is the first example in the world and has unseen application possibilities in industry, which is confirmed by the Intergovernmental platform for R&D in Europe reward.

2 The two-stage high-temperature heat pump with a flash expander (HP – FE)

Heat pump is a process device which is used for heating. Its operation principles are based on the removal of low temperature from the environment, which is then given on a higher temperature level. The sources for the removal of energy are air, water or ground. The operating principles of a single-stage heat pump are shown in [4, 5].

The two-stage high-temperature heat pump with a flash expander includes two compressors and two expansion valves. The two stages of compression are necessary due to the reduction of a high ratio of pressures, which has negative effects on the performances of the compressor (energy consumption, cooling...). Middle stage of compression is separated by the flash expander, and the same refrigerant is used for heat transfer in both stages. Two-stage heat pump with flash – expander is presented on Fig. 1.

Due to the two - stages of compression there is a possibility of using lower temperature sources (10 to 30°C) for heating. In such implementation it is important which refrigerant will be used, because it has to have good thermo-physical characteristics as well as properly determined pressure.
2.1 Equations for the Calculation

When performing calculation for the two-stage high-temperature heat pump with a flash expander it is important to determine correctly the intermediate pressure of the refrigerant, between the first and the second stage, which is equal to the following according to the general equation:

\[ P_M = \sqrt{P_{S,1} \cdot P_{T,2}} \quad \text{(Pa)} \quad (2.1) \]

where:
- \( P_{S,1} \) - vapour pressure of the refrigerant on the inlet side of the first stage of compression (Pa)
- \( P_{T,2} \) - vapour pressure of the refrigerant on the delivery side of the second stage of compression (Pa)

Intermediate temperature \( T_M \), which for the first stage of compression of the refrigerant it is the condensation temperature \( T_{K,1} \), and for the second stage of compression it is the temperature of evaporation \( T_{U,2} \) is calculated using the following equation:

\[ T_M = \frac{-B \pm \sqrt{B^2 - 4 \cdot A \cdot (C - \log \left( \frac{P_{\text{sat}}}{P_1} \right))}}{2 \cdot A} \quad \text{(K)} \quad (2.2) \]

Heat flows, of the first stage \( \Phi_{HP,1} \) and of the second stage \( \Phi_{HP,2} \) are calculated using the equations:

\[ \Phi_{HP,1} = q_{m,S_1} \cdot (h_{g,2} - h_{l,3}) \quad \text{(W)} \quad (2.3) \]
\[ \Phi_{HP,2} = q_{m,S_2} \cdot (h_{g,6} - h_{l,7}) \quad \text{(W)} \quad (2.4) \]

where:
- \( h_{g,6} \) - specific enthalpy of refrigerant vapour on the delivery side of the compressor of the second stage of the heat pump (J/kg·K)
- \( h_{l,3} \) - specific enthalpy of the liquid refrigerant in the condenser of the second stage of the heat pump (J/kg·K)

The heat number of the two-stage heat pump with a flash expander is calculated using the following equation:

\[ COP = \frac{\Phi_{HP,2}}{P_{K,1} + P_{K,2}} \quad / \quad (2.5) \]

Other parameters of the high-temperature heat pump with a flash expander and the dimensions of the compressor for both stages are calculated according to the same principle as for single-stage high-temperature heat pumps [6, 7].

2.2 Calculation results

The calculations of the two-stage high-temperature heat pump with a flash expander for refrigerants R-600a, R-290, R245fa and R-134a were conducted at a temperature of condensation \( t_k = 70\,^\circ\text{C} \), \( t_k = 80\,^\circ\text{C} \). Results are shown in Fig. 2 and Fig. 3 shows the dependences between COP and heat flow of evaporation temperature.

The main objective of the implementation of the two-stage heat pump with a flash expander is to try to reduce the strength which is necessary for the compressor to operate in order to reduce the ratio between pressures and increase the useful heat flow of the condensation part of the pump.

When conducting the calculation, the fact that the compressors in both stages are equal, was taken into consideration. The compressors were piston compressors WBH chosen from the catalogue of the manufacturer called MYCOM [8]. The volume flow of the compressor \( q_v = 637 \, \text{m}^3/\text{h} \), maximum operating pressure \( P_{k,\text{max}} = 2,0 \, \text{MPa} \) and maximum power of the compressor \( P_{k,\text{max}} = 145,0 \, \text{kW} \).
3 The two-stage heat pump with a heat exchanger (HP-HE)

The two-stage heat pump with a heat exchanger consists of two single-stage heat pumps. The stages are separated by the heat exchanger, which serves as the condenser for the first stage and as evaporator for the second stage. The stages are separated in such a way that refrigerants are not in contact. This means that each stage has its own refrigerant. The heat pump diagram is shown in Fig. 4. [11].

![Diagram of the two-stage heat pump with a heat exchanger (HP-HE)](image)

**Fig. 4: The two-stage heat pump with a heat exchanger (HP – HE)**

3.1 Equations for the calculation

Calculation of the characteristic parameters of the operation of the two-stage heat pump with a heat exchanger is more simply conducted according to the principle of the two single-stage pumps, where it is assumed that the heat flow of the condensation of the first stage $\Phi_{C,1}$, is equal to the heat flow of boiling of the second stage $\Phi_{B,2}$. It holds that:

$$\Phi_{C,1} = q_{m,s,1} \cdot (h_{g,2} - h_{l,3}) \ (W) \quad (3.1)$$

$$\Phi_{C,1} = \Phi_{B,2} \ (W) \quad (3.2)$$

It follows that the mass flow of the refrigerant on the inlet side of the compressor of the second stage of the heat pump is equal:

$$q_{m,2} = \frac{\Phi_{B,2}}{(h_{g,5} - h_{l,8})} \ (kg/s) \quad (3.3)$$
where:
\( h_{k,5} \) - specific enthalpy of the refrigerant which enters the compressor in the second stage of the heat pump (J/kg·K)
\( h_{l,8} \) - specific enthalpy of the liquid refrigerant which enters the evaporator in the second stage of the heat pump (J/kg·K)

Heat number of the two-stage heat pump with a heat exchanger is calculated:

\[
COP = \frac{\Phi_{HP,2}}{P_{K,1} + P_{K,2}} \quad (\text{K}) \quad (3.4)
\]

Condensation temperature of the first stage \( T_{K,1} \) and the boiling temperature of the second stage \( T_{U,2} \) of the refrigerant are calculated using the method of the average value between the temperatures of boiling of the first stage \( T_{U,1} \) and the temperature of the condensation of the second stage \( T_{K,2} \):

\[
T_M = \frac{T_{U,2} + T_{K,2}}{2} \quad (K) \quad (3.5)
\]

The possibility of using refrigerant in both stages requires that we determine the characteristics of the compressor for each stage separately according to the same principle as for the single-stage heat pump. [7,8,9].

### 3.2 Calculation Results

Calculation of characteristic parameters was conducted for the following refrigerants:
- R-407c in the first stage and R-600a in the second stage,
- R-717 in the first stage and R-600a in the second stage,
- R-290 in the first stage and R-600a in the second stage.

The results are given in Fig. 5 and Fig. 6, which show the power which is necessary for the operation of the compressor and the heat flow of the heat pump. Calculations were conducted for different temperatures \( t_u \) of the refrigerant: from 5.0°C to 25.0°C and the constant temperature of condensation \( t_k = 70°C \).

The values of COP of the two-stage heat pump with a heat exchanger were calculated for the condensation temperatures of the refrigerant \( t_k = 60°C \) and \( t_k = 80°C \). Fig. 7 shows the dependence between the COP and the refrigerant boiling temperature.

For the calculation of all suggested combinations of refrigerants the compressors chosen were: a piston compressor with the flow of 300 m³/h for the first stage and a piston compressor with the flow of 700 m³/h for the second stage.

![Fig. 5: The first stage of the two-stage heat pump with a heat exchanger](image)

![Fig. 6: The second stage of the two-stage heat pump with a heat exchanger](image)
4 Economic justification of the application of the high-temperature heat pump

The decision on the investment is much easier and simpler if it is based on the calculated parameters for individual implementations of heat pumps [10]. The calculation included the models with most suitable refrigerants. The costs of the investment are covered from personal sources and from loans in 30% to 70% ratio. The present value of the investment costs $C_{INV}$ je was calculated according to the equation:

$$C_{INV} = C_0 + \sum_{j=0}^{N} a_n \cdot C_{HP} \cdot (1 + r)^j \text{ (EUR/year)}$$  \hspace{1cm} (4.1)

where:

- $C_0$ - own funds (EUR)
- $C_{HP}$ - the cost of the heat pump (EUR)
- $r$ - discount rate ( / )
- $N$ - lifetime of the system (years)

Annuity factor $a_n$ is calculated using the equation:

$$a_n = \frac{r_a \cdot (1 + r_a)^N}{(1 + r_a)^N - 1} \text{ ( / )}$$  \hspace{1cm} (4.2)

where:

- $r_a$ - discount stage annuity ( / )

Maintenance costs of the heat pump were evaluated at 2 % of the purchase price. Net present value of expenses with inflation rate included was calculated using the equation:

$$C_S = \sum_{j=0}^{N} \frac{0.02 \cdot C_{HP} \cdot (1 + r_j)^j}{(1 + r_j + r)^j} \text{ (EUR/year)}$$  \hspace{1cm} (4.3)

Net present value of electric power costs for the operation of the compressor of the pump were calculated using the equation:

$$C_{PS} = \sum_{j=0}^{N} \frac{C_E \cdot P_k \cdot t_1 \cdot t_2 \cdot (1 + r_j)^j}{(1 + r_j + r)^j} \text{ (EUR/year)}$$  \hspace{1cm} (4.4)

Net present value of the income from the heat produced considering the inflation rate is calculated using the equation:

$$C_T = \sum_{j=0}^{N} \frac{C_T \cdot \Phi \cdot t_1 \cdot t_2 \cdot (1 + r_j)^j}{(1 + r_j + r)^j} \text{ (EUR/year)}$$  \hspace{1cm} (4.5)

Net present value of the heat income, considering the investment costs and maintenance costs, as well as the consumption of electric energy for the operation of the compressor is calculated using the equation:

$$C = C_{P} - (C_{INV} + C_{S} + C_{PS}) \text{ (EUR/year)}$$  \hspace{1cm} (4.6)

where:

- $r_j$ - inflation rate ( / )
- $C_E$ - electric power cost (EUR/kWh)
- $C_T$ - the cost of heat for heating (EUR/kWh)
- $t_1$ - operation time per day (h/day)
- $t_2$ - operation time per year (days/year)

Table 1 contains data for the calculation of different implementations of heat pumps. It shows NPV at different temperatures of boiling of refrigerants, and at a constant condensation temperature $t_c=70\text{°C}$. 

Fig.7: COP of the two-stage heat pump with a heat exchanger
Table 1: Data for the calculation of the economic analysis of different implementations of the two-stage heat pump (HP)

<table>
<thead>
<tr>
<th>Implementation of the two-stage heat pump</th>
<th>HP - FE</th>
<th>HP - HE</th>
<th>HP - HE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>R-600a</td>
<td>R-717/R-600a</td>
<td>R-407C/R-600a</td>
</tr>
<tr>
<td>Temperature of boiling ( t_U ) [°C]</td>
<td>20,0</td>
<td>20,0</td>
<td>20,0</td>
</tr>
<tr>
<td>Temperature of condensation ( t_K ) [°C]</td>
<td>70,0</td>
<td>70,0</td>
<td>70,0</td>
</tr>
<tr>
<td>Heat flow of the heat pump ( \Phi_{HP} ) [kW]</td>
<td>473,0</td>
<td>674,2</td>
<td>631,1</td>
</tr>
<tr>
<td>Compressor strength ( P_C ) [kW]</td>
<td>111,1</td>
<td>161,6</td>
<td>116,9</td>
</tr>
<tr>
<td>Operation time per day ( t_1 ) [h/day]</td>
<td>18</td>
<td>18</td>
<td>18</td>
</tr>
<tr>
<td>Operation time per year ( t_2 ) [days/year]</td>
<td>120</td>
<td>120</td>
<td>120</td>
</tr>
<tr>
<td>Heat pump lifetime ( N ) [years]</td>
<td>20</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Own funds ( C_{co} ) [EUR]</td>
<td>28,200</td>
<td>31,800</td>
<td>31,800</td>
</tr>
<tr>
<td>Cost of the heat pump ( C_{HP} ) [EUR]</td>
<td>94,000</td>
<td>106,000</td>
<td>106,000</td>
</tr>
<tr>
<td>Electric power costs ( C_{E} ) [EUR/kWh]</td>
<td>0,07</td>
<td>0,07</td>
<td>0,07</td>
</tr>
<tr>
<td>Cost of heat for heating ( C_{HT} ) [EUR/kWh]</td>
<td>0,0325</td>
<td>0,0325</td>
<td>0,0325</td>
</tr>
<tr>
<td>Discount rate ( r ) [%]</td>
<td>7,00</td>
<td>7,00</td>
<td>7,00</td>
</tr>
<tr>
<td>Inflation rate ( r_f ) [%]</td>
<td>1,20</td>
<td>1,20</td>
<td>1,20</td>
</tr>
</tbody>
</table>

5 Conclusion
The main subjects of the research are a two-stage heat pump with flash expander and a two-stage heat pump with heat exchanger. A new computer program was developed in accordance with mathematical model, which enables the determination of characteristic parameters (pressure, temperature, heat flow, COP, compressor power, etc.); on the basis of economic analyses with net present value methods the estimation of investment rentability was done.

With a two-stage heat pump with a flash expander, the best operating conditions are with the refrigerant R-600a. For the proposed implementation of the heat pump, the ratio between the pressures of the compressed refrigerants decreases. Boiling of the refrigerant at the temperature of \( t_U = 20°C \) is economical. The condensation temperature is \( t_K = 70°C \). Then the heat flow is 408,0 kW, the consumption of energy for the operation of both compressors is 111,1 kW. It is suggested to use the same compressor in the first and the second stage as with the single-stage pump.

The two-stage heat pump with a heat exchanger consists of two single-stage pumps. Its advantage is that we can use, in each stage, different refrigerants [11].

The analysis of thermo-physical properties of refrigerants established that the operation of a two-stage heat pump with a heat exchanger is the cheapest using refrigerants R-407c in the first stage and R-600a in the second stage of the heat pump. If the condensation temperature of the refrigerant in the second stage is \( t_K=70°C \), the operation of the heat pump of the first stage is economically justifiable at the boiling temperature of \( t_U = 10°C \). In the first stage, the type of the compressor which is recommended is WA or WBH, in the second stage only WBH. The heat flow of the two-stage heat pump with a heat exchanger in that case is 489,4 kW, the total consumption of energy for the operation of both compressors is 113,9 kW.

References: