Exergetic analysis of heat pump units for various climate conditions

A A Malyshev1,3, O S Malinina1, S I Arendateleva2 and I M Zawadzki2

1Saint Petersburg National Research University of Information Technologies, Mechanics and Optics, 49, Kronverksky Av., Saint Petersburg, 197101, Russia
2Yaroslav-the-Wise Novgorod State University, ul. B. St. Peterburgskaya, 41 173003, Veliky Novgorod, Russia
3E-mail: malyshev46@list.ru

Abstract. The article deals with exergetic analysis of a heat pump for climate of Russian North-West. The dependencies between the parts of the pump and its general characteristic are shown. The influence of ambient temperature, condensation temperature, and low-grade heat source on the working efficiency of heat pump units is analyzed. The changes in thermodynamic characteristics of the heat pump at ambient temperature decreasing form 0°C to –15°C are identified. The coefficient of performance is seen to decrease two times, the coefficient of transformation – by 1.4 times; overall exergetic coefficient of performance increases by 18 %. Working efficiency and coefficient of performance of the unit in general, and the ones of compressor, compensator, evaporator and thermal expansion valve are determined. The results of the research are of use for increasing efficiency of heat pumps.

1. Introduction
It is established that heat pumps are energy-efficient thermotransformers using both traditional and renewable energy resources. The use of HP provides energy saving, economic benefits and a positive environmental effect [1, 2, 3, 4] by reducing CO2 emissions into the atmosphere.

The advantages of heat pump installations (HPI) are the versatility of the type of energy used and a wide range of capacities, including small and medium HPP (Heating Power Plants) [2].

According to [5], heat pump installations are most effective for decentralized heat supply systems near sources of power generation and, first of all, in cottages. In these cases, electric heating is excluded (as an additional means to the HPI), in which the energy consumption is 3–4 times higher than the power consumption of HPI. Currently, in world practice, heat pump installations are among the most dynamically developing areas of energy [6, 7, 8]. According to the article [1], global sales of air-to-water heat pumps account for more than 1.8 million units. The volume of the world market for HP of all types in value terms is estimated at more than $10 billion. In Russia, the forecast of the market for steam-compressor heat pumps (SCHP) for 2030 is the end of the implementation of the current “Energy Strategy” – 11,000–15,000 units. (500–700 MW).

The authors [9, 10, 11] present the results of a comprehensive analysis of the HP and their comparison with boiler systems, as well as the effective combination of heat pumps with electrically heated boilers.
It is shown that, taking into account the average Russian cost of generating electricity, the annual fuel consumption of the “SCHP + electric boiler” system will be less than that of a traditional boiler house, provided that the heating coefficient of the heat pump is at least 4.0 and also at least 65 % coverage of annual heat load will be provided by HP.

In the literary source [9] exergetic approach to the analysis of real cycles of HPI is used. Based on the analysis, a method has been developed to increase the efficiency of SCHP with temperatures of water sources of low-grade heat above +5°C.

In [12, 13], the efficiency of heat pump installations using renewable energy sources (RES) was analyzed. The energy of the sun and wind, as well as the heat of the soil is considered. The concept of application of renewable energy sources relevant for the south of Russia is developed.

In the majority of works devoted to the study of the efficiency of heat pumps, the region of positive temperatures of low-potential heat sources is considered, while data on the operation of HPI at negative temperatures is of considerable interest.

Work at negative temperatures is often associated with the use of air-source heat pumps, since in these cases it is difficult to use water as a source of low-grade heat. At the same time, air heat pumps have a seasonal loading factor on average 10–30% lower than when using water heat pumps [14], which limits their use. Air pumps are rarely seen as an alternative to geothermal heat pumps, and as an advantage of the former, there is no need to perform expensive drilling operations [15].

Presented literature review shows:
- the value of heat pumps in the energy sector is constantly increasing;
- the area of use of HPI is expanding, including in the northern regions;
- the efficiency of a HPI is most often evaluated by methods based on the analysis of integral thermodynamic and technical and economic indicators;

It is necessary to evaluate the effectiveness of individual HPI units depending on their types, external conditions and tasks for the further development of technologies associated with the use of heat pumps, in addition to analyzing the integral indicators.

It seems that for such an assessment the use of an exergetic method of analyzing thermodynamic systems is quite reasonable.

Exergetic analysis was used when considering the energy complex of HPP-HPI. A significant increase in overall efficiency has been shown through the use of a heat pump installation.

2. Results and discussion
The aim of the work is to analyze the impact on the performance of a HPI and its individual elements of the following parameters:
- ambient temperature \( t_a \);
- condensation temperature \( t_c \);
- low potential heat source temperature \( t_s \).

The following initial data were taken for calculations:
- heat output \( Q_k = 10 \text{ kW} \);
- The water of the Gulf of Finland with the calculated temperature \( t_{s1} = 5°C \) is used as the low-potential source of heat;
- average room air temperature \( t_r = 22°C \) with the supply of heat through the fence;
- the difference between the condensation temperature and average air temperature in the room varied within \( \Delta T_{cr} = T_c - T_r = (12 \div 28) \text{ K} \);
- the temperature difference between the outside air and the boiling point of the refrigerant in the evaporator was assumed to be \( \Delta T_{ev} = T_a - T_0 = 7 \text{ K} \); \( \Delta T_{ev} = 10 \text{ K} \);
- temperature of the outside air (ambient) varied in the range \( t_a = (-5 \div -25)°C \)
- working media R134a.
The steam compressor heat pump cycle is shown in the figure 1.

![Figure 1. Steam compressor heat pump cycle.](image)

The results of the analysis of the influence of the ambient temperature on the thermodynamic parameters of the processes of HPI and its main elements under the specified conditions are summarized in Table 1 and presented as graphical dependencies (figure 2, a, b).

**Table 1.** The influence of the ambient temperature on the value exergetic loss processes in the elements of HPI.

| Name of elements of HPI | Ambient temperature, °C | -5  | -10 | -15 | -20 | -25 |
|-------------------------|--------------------------|-----|-----|-----|-----|-----|
| Compressor $DEx_{cm} = N_d - \Delta Ex_{cm}$, kW | 0.72 | 0.71 | 0.70 | 0.68 | 0.67 |
| Condenser $DEx_{cm} = [Ex_{cm}] - \Delta Ex_{cr}$, kW | 0.79 | 0.78 | 0.76 | 0.75 | 0.73 |
| Evaporator $DEx_{ev} = [Ex_s] - \Delta Ex_{ev}$, kW | 0.13 | 0.13 | 0.12 | 0.12 | 0.12 |
| Thermostatic valve $DEx_{tv} = [\Delta Ex_{tv}]$, kW | 0.34 | 0.33 | 0.32 | 0.32 | 0.31 |

![Figure 2. A graph of the dependence of the general exergetic efficiency of the heat pump (a), exergetic efficiency of the processes in the elements of the heat pump (b) on ambient temperature.](image)
From the obtained results it can be seen that the main losses are observed in the condenser. The reason is the choice of the accepted method of heat removal of condensation into the room. For a heat pump, an increase in thermal resistance on the air side leads to an increase in the condensation temperature of the refrigerant and a deterioration of the thermodynamic characteristics of the heat pump unit. For the considered conditions, the heating coefficient was $\mu = 3.51$.

Reducing the temperature difference between air and condensation will reduce irreversible losses and increase the efficiency of the heat pump as a whole. This can be achieved both by the development of the heat exchange surface and by intensifying heat transfer from the heated air.

It is necessary to assess the degree of influence of the temperature difference in the condenser on the thermodynamic parameters of the system to select the most appropriate direction for increasing the efficiency of a condenser. In addition, a sharp decrease in the exergetic efficiency of the evaporator with decreasing ambient temperature (figure 2, b) draws attention. This is due to the “depreciation” of exergy when the ambient temperature approaches the temperature of a low-potential source. The equalization of ambient temperature and low-potential source is the limiting case in which the thermodynamic efficiency of the evaporator equals zero.

2.1. The influence of temperature difference in the condenser

It should be noted that when using average temperatures in the room $T_r$, the losses in the condenser include two components:

- actual losses in the condenser,
- loss when mixing air with different temperatures in the volume of the room.

With intensive methods of heat removal in condensers (with forced air circulation) it is advisable to separate these losses. At the same time, it is necessary to perform an analysis taking into account the existing Norms and Rules for the permissible temperature difference between the air supplied and leaving the premises. This task is multiparametric and requires the specification of the object. It was decided not to separate the indicated loss components for a general assessment, and to perform an analysis of the effect on thermodynamic indicators of the difference in condensation temperatures and the average air temperature in the room at the same ambient temperature $t_a = -10^\circ C$. The remaining independent parameters are taken as identical [18].

The results of the analysis of the influence of the temperature difference in the condenser on the value of the exergetic losses of the HPI are given in the table 2.

**Table 2.** The results of the analysis of the influence of the temperature difference in the condenser on the value of the exergetic losses of HPI.

| Name                                      | Value   |
|-------------------------------------------|---------|
| Temperature difference between condensation and average room $\Delta T_{ce} = T_c - T_r$, R |         |
| Ambient temperature $t_a$, $^\circ C$     | -20     |
| Average room temperature $t_r$, $^\circ C$| 22      |
| Boiling temperature $t_0$, $^\circ C$     | -2      |
| Condensation temperature $t_c$, $^\circ C$| 34, 38, 42, 46, 50 |
| Thermal load on the evaporator $Q_0$, kW  | 8.88, 8.37, 7.85, 7.33, 6.80 |
| Heat capacity $Q_{cd}$, kW                | 10.99, 10.59, 10.18, 9.75, 9.46 |
### Table 1: Key Parameters for Evaporator and Compressor

| Parameter                                    | Evaporator | Compressor |
|----------------------------------------------|------------|------------|
| Power consumed by the compressor, $N_d$, kW  | 2.22       | 2.24       |
| Heating coefficient, $\mu$                   | 4.95       | 4.93       |
| Refrigerant mass flow, $M_a$, kg/c           | 0.056      | 0.054      |

#### Evaporator

| Parameter                                    | Value       |
|----------------------------------------------|-------------|
| Evaporator inlet water temperature, $t_1$, °C| 5           |
| Evaporator outlet water temperature, $t_2$, °C| 5           |
| Mass flow of water in the evaporator, $M_s$ = $Q_0/c_s(T_{s1} - T_{s2})$, kg/c | 0.704       |
| Average heat capacity of the coolant, $c_v$, kJ/(kg·K) | 4.200       |
| Change in the specific exergy of water, $\Delta ex_s = c_s(T_{s2} - T_{s1}) - T_a \cdot \ln(T_{s2}/T_{s1})$, kJ/kg | -0.615      |
| Change in exergy of water flow, $Ex_s = M_s \cdot \Delta ex_s$, kW | -0.433      |
| Change in the specific exergy of the refrigerant flow, $\Delta Ex_{ev} = (h_1 - h_6) - T_a \cdot (S_2 - S_1)$, kJ/kg | 4.627       |
| Change in exergy of refrigerant flow, $Ex_{ev} = M_a \cdot \Delta Ex_{ev}$, kW | 0.263       |
| Loss of exergy in the evaporator, $DEx_{ev} = \Delta Ex_{ev} - \Delta Ex_t$, kW | 0.17        |
| Exergetic evaporator efficiency coefficient, $\eta_{ev} = \Delta Ex_{ev} / \Delta Ex_t$, | 0.60712     |

#### Compressor

| Parameter                                    | Value       |
|----------------------------------------------|-------------|
| Change in the specific exergy of the refrigerant flow, $\Delta ex_{cm} = (h_2 - h_1) - T_{ev} \cdot (S_2 - S_1)$, kJ/kg | 27.171      |
| Change in exergy of refrigerant flow, $Ex_{cm} = N_d \cdot \Delta ex_{cm}$, kW | 1.544       |
| Loss of exergy in the compressor, $Ex_{cm} = N_d \cdot \Delta Ex_{cm}$, kW | 0.676       |
| Exergetic compressor efficiency coefficient, $\eta_{cm} = \Delta Ex_{cm} / N_d$, | 0.696       |

#### Thermostatic Valve

| Parameter                                    | Value       |
|----------------------------------------------|-------------|
| Change in the specific exergy of the refrigerant flow, $\Delta ex_{rv} = -T_{ev} \cdot (S_6 - S_5)$, kJ/kg | -3.058      |

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**Note:** The values are given for different operating conditions, indicating the performance of the system under varying conditions.
## Loss of exergy in a thermostatic valve

\[ D\dot{E}_{rv} = \Delta E_{rv}, \text{ kW} \]

|         | 0.174 | 0.211 | 0.251 | 0.292 | 0.334 |
|---------|-------|-------|-------|-------|-------|

## Exergetic efficiency coefficient of thermostatic valve

\[ \eta_{rv} = 0 \]

## Condenser

| Change in the specific exergy of the refrigerant flow \( \Delta e_{x_c} \) | -28.741 | -30.503 | -32.147 | -33.677 | -35.087 |
| Change in exergy of refrigerant flow \( \Delta E_{x_c} \) | -1.634 | -1.698 | -1.748 | -1.784 | -1.803 |
| Indoor exergy flow \( \Delta E_{x_i} \) | 1.192 | 1.148 | 1.104 | 1.057 | 1.026 |
| Loss of exergy in the condenser \( D\dot{E}_{x_c} \) | 0.44 | 0.55 | 0.64 | 0.73 | 0.78 |
| Exergy efficiency of the condenser \( \eta_{cd} \) | 0.729 | 0.676 | 0.632 | 0.593 | 0.569 |

## General indicators

| Exergy flow to the room \( \Delta E_{x_r} \) | 1.192 | 1.148 | 1.104 | 1.057 | 1.026 |
| Total summed up exergy \( \Delta E_{x_gen} \) | 2.653 | 2.748 | 2.843 | 2.918 | 2.982 |
| Exergetic efficiency coefficient of the system \( \eta_{gen} \) | 0.449 | 0.418 | 0.388 | 0.362 | 0.344 |
| General thermodynamic losses \( D\dot{E}_{x_gen} \) | 1.462 | 1.600 | 1.739 | 1.861 | 1.956 |

## Relative components of thermodynamic losses

| In evaporator \( \Omega_{ev} \) | 0.116 | 0.100 | 0.086 | 0.075 | 0.066 |
| In compressor \( \Omega_{cm} \) | 0.462 | 0.424 | 0.399 | 0.377 | 0.365 |
| In thermostatic valve \( \Omega_{rv} \) | 0.119 | 0.132 | 0.144 | 0.157 | 0.171 |
| In condenser \( \Omega_{cd} \) | 0.302 | 0.344 | 0.370 | 0.390 | 0.398 |

From the calculation, it follows that reducing the temperature difference \( \Delta T_{ave} \) from 28 K to 12 K leads:

- to an increase in the performance of the heat pump by 16% while reducing power consumption by 16%;
- to an increase in the heating coefficient by 39%;
- to an increase of the total exergetic efficiency coefficient of the heat pump on.

### 2.2. Influence of low-potential source temperature on the efficiency of HPI

External air is taken as a source of low potential heat. Currently, many manufacturers produce units for air conditioning of residential, administrative and industrial premises with units that provide work on
the cycles of both the refrigerating machine and the heat pump with air condensers and evaporators. To reduce the well-known deficiencies in the operation of such units during periods of a year with a negative temperature, manufacturers use, depending on the conditions of use, various technical solutions (heat recovery of ventilation emissions, air circulation, combined heat supply, etc.).

The variant of the heat pump units without the use of these solutions leads to the lowest thermodynamic efficiency. At the same time, such a production of units with forced air circulation in the evaporator and condenser is typical and it is advisable to perform a thermodynamic analysis for these conditions.

The influence of $\Delta T_{\text{ave}}$ on the thermodynamic efficiency coefficient of the compressor and the condenser is shown in the figures 3 and 4.

![Figure 3](image1.png)

**Figure 3.** A graph of influence of heat pump heat capacity on ambient temperature.

![Figure 4](image2.png)

**Figure 4.** A graph of the dependence of the heating coefficient of the heat pump on the ambient temperature.
Figure 5. A graph of the dependence of the exergy efficiency coefficient of the heat pump on the ambient temperature.

The analysis performed reflects the significant influence of $T_{am}$ on the general thermodynamic characteristics of the heat pump. When $T_{am}$ от 0°C до –15°C, $\Delta T_{ev} = 7$ K and $\Delta T_{ev} = 10$ K they change accordingly:
- productivity decreases by 2 times;
- heating coefficient decreases by 1.4 times;
- common exergetic efficiency coefficient increases by 18%.

The cycles of the heat pump and the refrigerating machine are reverse, for which an increase in the difference in boiling points and condensation leads to an increase in the internal thermodynamic losses of the cycle, a decrease in power: the refrigerating of evaporator or the thermal of condenser, cooling and heating coefficients. At the same time, specific decisions of individual elements of the equipment can influence the nature of the change in exergetic efficiency, ensuring its changes in any direction, including the presence of optimum in a certain mode. Its only principal difference for the heat pump in comparison with the refrigerating machine is the following fact. For a chiller, any thermodynamic loss leads to a decrease in exergetic efficiency.

For a heat pump, internal losses lead to an increase in thermal power supplied to the room by a condenser and having an exergetic potential relative to the environment (outside air).

As we noted, the option of using an external low-potential source with a temperature above $T_{ac}$ is more expedient. The results of its comparison with the above data when using water with a temperature of $T_{s1} = 5^\circ$C and $\Delta T_{ev} = 7$ K, of outdoor air with a temperature of $T_{am} = -10^\circ$C and $\Delta T_{ev} = 10$ K with other things being equal:
- an increase in productivity by 77%,
- an increase of heating coefficient by 54%,
- an increase in the total exergetic efficiency by 29%.

The accepted temperature differences in the apparatus are close to those recommended in industrial heat exchangers of heat pumps.
For the option of supplying heat from the outside air in the evaporator in the temperature range from
–15°C to 0°C while decreasing the temperature difference $\Delta T_{ev}$ from 10 K to 7K similar indicators will
be:
– an increase in productivity by $(14 \div 17)$%,
– an increase of the heating coefficient by $(7 \div 8)$%,
– an increase of total exergetic efficiency coefficient by $(7\div8)$%.

3. Conclusion
1. The study established quantitative laws of the influence on the thermodynamic efficiency of a heat
pump with a piston compressor and its elements, parameters of low-potential heat sources (outside air
and external with a nonzero exergetic potential), temperature differences in heat exchange processes at
negative ambient temperatures.
2. Increase in the total exergetic efficiency coefficient of the heat pump with a decrease in the ambient
temperature is characteristic for the selected compressor in all the modes considered. The reasons are
associated with an increase in the exergetic value of the external low-temperature source and the heat
supplied to the room when the $T_{oc}$ decreases.
3. The fundamental difference between the influence of internal losses on the thermodynamic
efficiency of a heat pump and a refrigerating machine is shown.
4. The inexpediency of application of devices for heating air in rooms operating in natural convection
mode in heat pumps is substantiated.
5. The results of the study can be used to improve the efficiency of heat pumps.

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