# ENTROPY-CYCLE METHOD FOR ANALYSIS OF REFRIGERATION MACHINE AND HEAT PUMP CYCLES

by

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*Real and ideal cycles of any complexity can be compared through the definition of the irreversibilities and ways of their minimization in the thermodynamic analysis. The paper considers the use of entropy-cycle method for thermodynamic analysis of refrigerating machines and heat pumps. Using this method preconditions of practical use for analysis, synthesis, and optimization have been created. The methodology of the entropy-cycle method can be use in many areas of the scientific work and educational process.*

Key words: *ideal cycle, real cycle, irreversibility, entropy, exergy*

## Introduction

The thermodynamic analysis is an instrument to define the performance of any energy conversion system (including refrigeration machines and heat pumps) and indicate the way of its improvement. All elements that render significant influence on the efficiency have to be perfect. The thermodynamic analysis allows to find technical ways to increase energy efficiency and show how to do this, whereas the thermoeconomic analysis reveals the benefit at given economic situation when the fuel, capital, operational, and maintenance costs are known. The creation of a real schemes of compressor refrigeration machines and heat pumps have been considered from this point of view. The entropy method [1-6] is a calculation and analysis of the entropy generation in all parts of the system under consideration. Some parts (subsystems) can be excluded from the system, but they should be in connection with the system during the operation. The calculation of the total entropy generation and exergy destruction of the system is necessary for the definition of the real work produced in real direct thermodynamic cycles (power systems) or consumed in reverse ones (refrigeration machines and heat pumps). To solve the present problem we will introduce the "ideal cycles" (*i. e.* reversible cycle-samples [7-8]) and create a methodology to compare the ideal and real cycles. The real processes differ from ideal ones because of some technical reasons [8-9]. For example:

- it is not possible to realize a theoretical process in real conditions of operation,
- any energy conversion system working by ideal cycle cannot be efficient for a long time, and

– any energy conversion system working by ideal cycle is dangerous (there is an opportunity for breakdown) and unreliable in behavior.

The deviation of the real cycle from the ideal one (from the viewpoint of exergy analysis) is due to exergy destructions. The exergy destructions can be defined by a procedure of step-by-step calculations of all elements of an energy conversion system. The equation of exergy balance for $k$-th element of the system is:

$$E_{\mathrm{F}k} \quad E_{\mathrm{P}k} \quad \Delta E_k \tag{1}$$

where the value $\Delta E_k$ is the sum of exergy destruction $E_{\mathrm{D}k}$ and exergy losses $E_{\mathrm{L}k}$ in $k$-*th* element [6]. The concept of exergy destruction describes irreversibilities as a result of the replacement of an ideal element on the real one. For an ideal element $E_{\mathrm{D}k} = 0$. The concept of exergy losses describes the heat contact between an element and the environment ($E_{\mathrm{L}k}$). The value $E_{\mathrm{D}}$ for the $k$-th element of an energy conversion system is determined according to the exergy theory by Gouy-Stodola theorem [5-6]:

$$E_{\mathrm{D}k} \quad T_{\mathrm{env}} S_{\mathrm{gen}k} \tag{2}$$

Using the entropy-cycle method (ECM) every one can answer thoroughly to the question about the greatest possible energy (exergy) efficiency of an energy conversion system and show the ways for its further improvement. The ECM can be used for analysis, synthesis of schematic solution, and optimization of any energy conversion system. Let us consider step-by-step the basic points of the entropy method, cycles method and, at the end, the ECM (proposed by authors) as ones of the second law methods of applied thermodynamics.

## Entropy method

If an energy conversion system consists of a set of elements $k = 1, 2, \ldots, n$, then using the additive property of the entropy, we can write:

$$S_{\mathrm{gen\ tot}} \quad {}_{k\ \ 1}^{\ \ n} S_{\mathrm{gen}\,k} \tag{3}$$

Similarly, the value of exergy destruction, eq. (2), is:

$$E_{\mathrm{D\,tot}} \quad {}_{k\ \ 1}^{\ \ n} E_{\mathrm{D}k} \tag{4}$$

and

$$E_{\mathrm{D\,tot}} \quad T_{\mathrm{env}} \, {}_{k\ \ 1}^{\ \ n} S_{\mathrm{gen}\,k} \tag{5}$$

The value $S_{\mathrm{gen}k}$ can be written in general form as algebraic sum of entropy and heat flows through the boundary of the $k$-th element, fig. 1:

$$S_{\mathrm{gen}k} = -\sum_{\mathrm{in}_i=1}^{\mathrm{IN}} m_{\mathrm{in}_i k} s_{\mathrm{in}_i k} + \sum_{\mathrm{out}_i=1}^{\mathrm{OUT}} m_{\mathrm{out}_i k} s_{\mathrm{out}_i k} - \sum_{\mathrm{add}_i=1}^{\mathrm{ADD}} \frac{Q_{\mathrm{add}_i}}{T_{\mathrm{add}_i}} \tag{6}$$

and

$$S_{\mathrm{gen\ tot}} = -\sum_{k=1}^{n}\sum_{\mathrm{in}_i=1}^{\mathrm{IN}} m_{\mathrm{in}_i k} s_{\mathrm{out}_i k} + \sum_{k=1}^{n}\sum_{\mathrm{out}_i=1}^{\mathrm{OUT}} m_{\mathrm{out}_i k} s_{\mathrm{out}_i k} - \sum_{k=1}^{n}\sum_{\mathrm{add}_i=1}^{\mathrm{ADD}} \frac{Q_{\mathrm{add}_i}}{T_{\mathrm{add}_i}} \tag{7}$$

where $m_{\mathrm{in}_i}$ and $m_{\mathrm{out}_i}$ are inlet ($\mathrm{in}_i$) and outlet ($\mathrm{out}_i$) flows; $\mathrm{in}_i = 1, 2, \ldots, \mathrm{IN}$; $\mathrm{out}_i = 1, 2, \ldots, \mathrm{OUT}$. The values of $Q_{\mathrm{add}_i}$ are heat flows from external bodies; $\mathrm{add}_i = 1, 2, \ldots, \mathrm{ADD}$. $T_{\mathrm{add}_i}$ are temperatures corresponding to heat flows $Q_{\mathrm{add}_i}$.

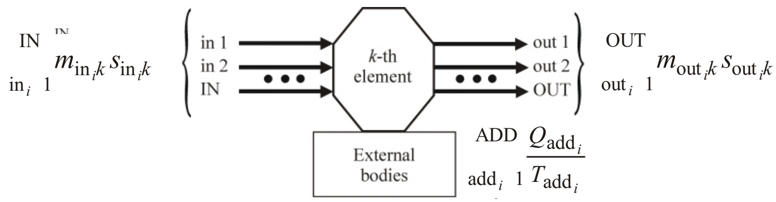


**Figure 1. Thermodynamic model of *k*-th element**

The value of the real cycle work can be determined:

– for direct cycle (produced work)

$$W_{\mathrm{real}} = W_{\mathrm{ideal}} - E_{\mathrm{D\,tot}} \tag{8a}$$

– for inverse cycle (consumed work)

$$W_{\mathrm{real}} = W_{\mathrm{ideal}} + E_{\mathrm{D\,tot}} \tag{8b}$$

To take into account the influence of the exergy destruction in $k$-th element of energy conversion system on the exergy destruction of the system, the contribution of exergy destruction (entropy generation) can be written in the form:

$$y^*_{\mathrm{D}k} = \frac{E_{\mathrm{D}k}}{E_{\mathrm{D\,tot}}} = \frac{S_{\mathrm{gen}\,k}}{S_{\mathrm{gen\ tot}}} \tag{9}$$

then

$$y^*_{\mathrm{D\,tot}} = \sum_{k=1}^{n} y^*_{\mathrm{D}k} = 1 \tag{10}$$

The total ratio of exergy destruction is:

$$y_{\mathrm{D\,tot}} = \frac{\sum_{k=1}^{n} E_{\mathrm{D}k}}{E_{\mathrm{F\,tot}}} = \sum_{k=1}^{n} y_{\mathrm{D}k} \tag{11}$$

The exergy efficiency of energy conversion system is:

$$\varepsilon = \frac{E_{\mathrm{P\,tot}}}{E_{\mathrm{F\,tot}}} = 1 - \frac{\sum_{k=1}^{n} E_{\mathrm{D}k}}{E_{\mathrm{F\,tot}}} = 1 - y_{\mathrm{D\,tot}} \tag{12}$$

## Cycles method

The Cycles Method (CM) was proposed for a first time in Odessa State Academy of Refrigeration in the 1950-s by Martinovsky [7] and an extension of the method was made later by Melzer [10]. The thermodynamic efficiency of the cycle in any stage of the thermodynamic analysis is:

$$\eta = \frac{COP_{\text{examinated cycle}}}{COP_{\text{ideal cycle}}} \tag{13}$$

The CM is a step-by-step transition from ideal cycle to real cycle by introducing of irreversibilities stemmed from the real conditions of each process. The basic points of the CM are *ideal cycle* → *model cycle* → *real cycle*.

The ideal cycle is a reversible cycle composed of the minimal necessary number of components (for *ideal cycle* $\eta = 1$). The traditional *ideal cycle* is Carnot-cycle. The *model cycle* is the cycle with the minimal necessary number of components including all irreversibilities. The *real cycle* describes the real system containing full set of basic and additional components and their irreversibilities. The CM is perfect base for research, as well as a strong methodological tool. The CM is widely used in educational process. The students can easily understand why the real cycle differs from ideal one, and which are the reasons for irreversibilities to occur. The implementation of the CM for analysis of real thermodynamic cycle of simple vapor-compressor refrigeration machine is shown in fig. 2. Let us consider each stage of creation of real thermodynamic cycle of vapor-compressor refrigeration machine and heat pump.

*Cycle 1. Ideal cycle* (Carnot-cycle). Temperatures of heat reservoirs are $T_{\mathrm{env}}$ and $T_{\mathrm{cold}}$:

– specific cold capacities

$$q_{\mathrm{cold}} = \mathrm{area(b\text{-}1\text{-}4\text{-}a)} = T_{\mathrm{cold}} \Delta s \tag{14a}$$

– specific heat capacities

$$q_{\mathrm{hot}} = \mathrm{area(b\text{-}2\text{-}3\text{-}a)} = T_{\mathrm{env}} \Delta s \tag{15a}$$

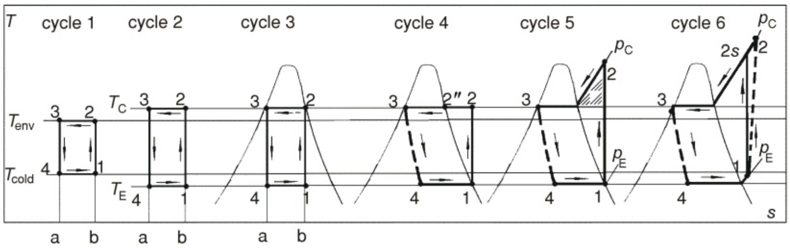


**Figure 2. Cycles method of thermodynamic analysis of vapor-compressor refrigeration machine (temperature of heat sources are $T_{cold}$ and $T_{env}$) or heat pump (temperature of heat sources must be change to $T_{env}$ and $T_{hot}$)**

– work of *cycle 1*

$$w_{cycle\,1} \quad \text{area}(1\text{-}2\text{-}3\text{-}4) \quad (T_{env} \quad T_{cold})\Delta s \tag{16a}$$

– *COP* of *cycle 1* (inverse reversible Carnot-cycle)

$$COP_{cycle\,1} \quad \frac{q_{cold}}{w_{cycle\,1}} \quad \frac{T_{cold}}{T_{env} \quad T_{cold}} \tag{17a}$$

– thermodynamic efficiency

$$\eta \quad 1 \tag{18a}$$

*Cycle 2. Endo-reversible Carnot-cycle*. The temperatures of the heat reservoirs ($T_C$ and $T_E$) are basis of work temperatures of refrigerating machine. The Cycle 2 corresponds to schematic solution of simple refrigeration machine with 4 elements: compressor, condenser, expander, and evaporator, fig. 3a. Heat transfer from a heat reservoir to a

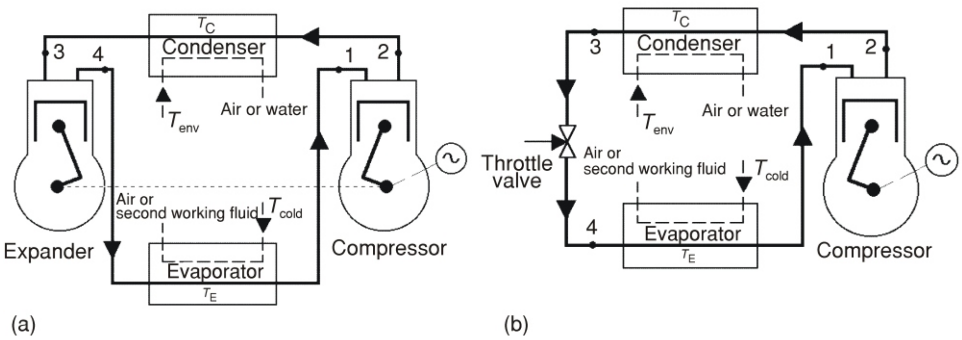


**Figure 3. Refrigeration machine (heat pump) schematic solution:**
*(a) with expander (correspond to cycle 1, cycle 2, and cycle 3 on fig. 2); (b) with throttle valve (correspond to cycle 4, cycle 5, and cycle 6 on fig. 2)*

working fluid of a refrigeration machines is possible if any finite temperature difference exists. Thus, the heat exchange surface must have real sizes. The basic working temperatures of the refrigeration machine are as follows:

– evaporation temperature of the working fluid $T_E \quad T_{cold} \quad \Delta T_E$, and
– condensation temperature of the working fluid $T_C \quad T_{env} \quad \Delta T_C$

where $\Delta T_E$ and $\Delta T_C$ depend on the design of the evaporator or condenser, respectively. Let us rewrite eqs. (14a)-(18a) taking into account the values $\Delta T_E$ and $\Delta T_C$:

– cold capacities

$$q_E = \text{area(b-1-4-a)} \quad T_E \Delta s \tag{14b}$$

– specific heat capacities

$$q_C = \text{area(b-2-3-a)} \quad T_C \Delta s \tag{15b}$$

– work of *cycle 2*

$$w_{\text{cycle 2}} \quad \text{area(1-2-3-4)} \quad (T_C \quad T_E)\Delta s \tag{16b}$$

– *COP* of *cycle 2* (inverse endo-reversible Carnot-cycle)

$$COP_{\text{cycle 2}} \quad \frac{q_E}{w_{\text{cycle 2}}} \quad \frac{T_E}{T_C \quad T_E} \tag{17b}$$

– thermodynamic efficiency

$$\eta \quad \frac{COP_{\text{cycle 2}}}{COP_{\text{ideal cycle}}} \quad 1 \tag{18b}$$

*Cycle 3. Endo-reversible Carnot-cycle for real working fluid. Cycle 1* and *Cycle 2* are depicted not to take into account the real working fluid properties (on *T-s* diagram the saturated curves are missing). The processes 4-1 and 2-3 are isothermal and therefore the Carnot-cycle should be located in the zone of damp vapor (between saturated curves). From the classical thermodynamics viewpoint, *Cycle 2* and *Cycle 3* are similar and:

$$COP_{\text{cycle 2}} \quad COP_{\text{cycle 3}} \tag{19}$$

The *Cycle 3* cannot be used in the practice because of the following reasons:

*Reason 1*: compression process 1-2 in the zone of damp vapor is not possible. There is a theory of hydraulic impact, fig. 4

– the process 1-2 is *dry compression*, *i. e.* it starts and finishes in the zone of the superheat vapor. Dry compression is an obligatory condition of trouble-free and long-term compressor work,
– the process 1'-2' is *moisture compression*. This compression regime is a safe mode of work but not recommended for practical operation, and
– the process 1"-2" is *hydraulic impact*. This is the roughest failure for the refrigeration machines and heat pumps.

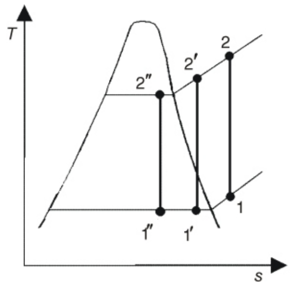


**Figure 4. Theory of hydraulic impact**

*Reason 2*: adiabatic expansion in the zone of the damp vapor is impossible – the theory of replacement the expander with throttle valve, fig. 5. The expander is an element for adiabatic expansion. The expander is similar to the compressor by capital and operational costs. The potentially work that can be produced by expander is:

$$w_{\text{EX}} \quad \text{area}(3-4_s\text{-d}) \tag{20}$$

The work of refrigeration machine cycle with expander is:

$$w_{\text{cycle}} \quad w_{\text{CM}} \quad w_{\text{EX}} \tag{21}$$

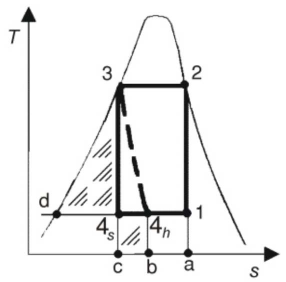


**Figure 5. Cycle with expander (1-2-3-$4_s$) and cycle with throttle valve (1-2-3-$4_h$)**

In vapor compression refrigeration machines the work produced in the expander, $w_{\text{EX}}$, is less than the work necessary to overcome the friction in the expander. Thus, the expander should be replaced by any other element for expansion, for example, by throttle valve, fig. 3b. The replacement of the expander with a throttle valve is positive from capital and operation costs viewpoint. Let us consider this replacement from thermodynamics viewpoint. Process 3-$4_s$ is expansion, process 3-$4_h$ is throttling. $T_{4s} \quad T_{4h}$, but, $h_{4s} \quad h_{4h}$, *i. e.* the specific cold capacities of refrigeration machine with throttle valve is less than the specific cold capacities of refrigeration machine with expander:

$$\begin{gathered}\Delta q_{\text{E}} \quad \text{area}(\text{a}-1-4_s-\text{c}) \quad \text{area}(\text{a}-1-4_h-\text{b}) \\ \text{area}(\text{b}-4_h-4_s-\text{c})\end{gathered} \tag{22}$$

The cycle work of refrigeration machine with throttle valve is equal to the compressor work:

$$w_{\text{cycle}} \quad w_{\text{CM}} \quad \text{area}(1-2-3-\text{d}) \tag{23}$$

*COP* of refrigeration machine with throttle valve is, fig. 5:

$$COP_{\text{cycle 3}} \quad \frac{q_{\text{E}}}{w_{\text{cycle 3}}} \quad \frac{q_{\text{E}}}{w_{\text{CM}}} \quad \frac{h_{4h} \quad h_1}{h_2 \quad h_1} \tag{24}$$

*Cycle 4*. Cycle with throttle valve, *dry compression* and heat transfer processes at isothermal conditions. The *cycle 4* is also impossible in practice because of the presence of two parts in the compression process:

– 1-2 is adiabatic compression (possible in practice), and
– 2-2 is isothermal compression (impossible in practice).

*Cycle 5. Model cycle*. The model cycle for vapor-compressor refrigeration machines and heat pumps is Plank-cycle [11]. In the damp vapor zone, the isobars coincide with the isotherms, therefore to construct the cycle it is necessary to determine values of the pressures according to the isotherms (for each working fluid):

$$T_{\text{E}} \quad p_{\text{E}} \quad \text{and} \quad T_{\text{C}} \quad p_{\text{C}}$$

In this case there is an additional irreversibility. This is the additional heat from superheat vapor after compressor (shaded area in *cycle 5* in fig. 2). It is obvious that the temperature at the end of compression, $T_2$, differs from the temperature of condensation, $T_{\text{C}}$, $T_2 > T_{\text{C}}$. The Plank-cycle is impossible in practice since it is impossible to fix strongly point 1 and point 3 in the saturated curves (in real operation conditions).

*Cycle 6. Real refrigeration machine cycle. Cycle 6* is shown in fig. 2 where point 1 is in the zone of superheat vapor, point 3 is in the zone of overcooled liquid, the real compression process is 1-2 not similar to the theoretical compression process 1-2*s*. The character of the polytrack 1-2 depends on various design characteristics of the compressor (the most important factor is the cooling system of the cylinders).

## Entropy-cycle method

Figure 6 depicts a graphical interpretation of the ECM. Let us consider the ECM for refrigeration machines and heat pumps separately.

### *Refrigeration machine*

Figure 6a depicts a real cycle of vapor-compressor refrigerating machine 1-2-2*s*-2 -3-4-1. The work of the cycle is:

$$w_{\text{real}} \quad q_{\text{C}} \quad q_{\text{E}} \quad \text{area(e-2-2 -3-a-b-4-1-d-e)} \quad h_2 \quad h_1 \tag{25}$$

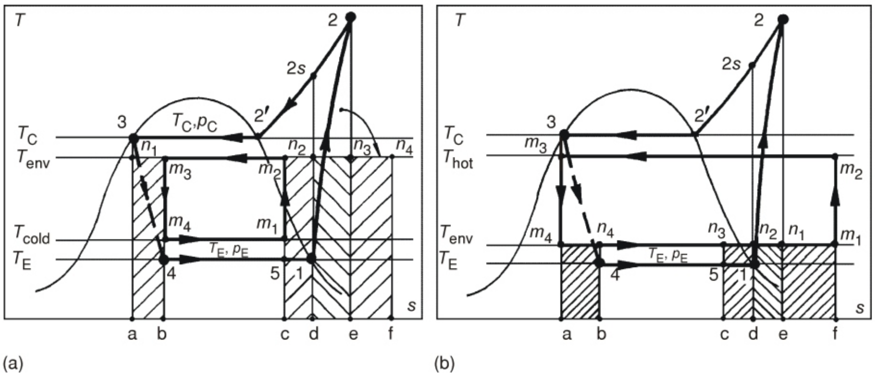


**Figure 6. Entropy-cycle method for inverse real cycles: (a) refrigeration machine; (b) heat pump**

Let us construct ideal Carnot-cycle between the same temperatures of the heat sources ($T_{cold}$, $T_{env}$) as real one and to define the exergy destruction of the real cycle. The ideal Carnot-cycle is $m_1$-$m_2$-$m_3$-$m_4$. The general equation for comparison of the cycles is as follows: the positive effect (*product* in exergy analysis) must be constant [7-8]. Thus, the specific cold capacity of the ideal Carnot-cycle must be equal to specific cold capacity of real cycle [7-8]:

$$\text{area}(4\text{-}5\text{-}m_1\text{-}m_4\text{-}4) = \text{area}(\text{c-d-1-5-c}) \tag{26}$$

then

$$h_1 - h_4 = T_{cold}(s_5 - s_4) \tag{27}$$

$$s_5 - s_4 = \frac{h_1 - h_4}{T_{cold}} \tag{28}$$

The work of an ideal Carnot-cycle is the minimal possible work:

$$w_{min} = \text{area}(m_1\text{-}m_2\text{-}m_3\text{-}m_4) \tag{29}$$

The specific extra work of real cycle in comparison with the ideal Carnot-cycle with condition eq. (26) is:

$$e_{D\,tot} = w_{real} - w_{ideal} = \text{area}(\text{a-b-}m_3\text{-}m_2\text{-c-e-2-2-3-a}) \tag{30}$$

or

$$e_{D\,tot} = e_F - e_P = w_{real} - e_{cold} \tag{31}$$

where

$$e_{cold} = q_{cold}\left(\frac{T_{env}}{T_{cold}} - 1\right) \tag{32}$$

For the refrigeration machine cycle (heat pump also) shown in fig. 6, it is possible to introduce a value of specific exergy destruction (traditionally not used in general exergy theory [5-6]) since the mass flow rate of the working fluid through all elements remains constant. Let us create a methodology to determine the exergy destruction of each element of simple vapor-compressor refrigeration machine by eq. (6) and fig. 1. The total number of the elements is $n = 4$.

*Compressor*: $k = 1$, $m_{in1} = m_{out1} = 1$; $s_{out1} = s_2$, $s_{in1} = s_1$, $Q_{add1} = 0$
then $s_{gen1} = s_2 - s_1$
and $e_{D1} = T_{env}(s_2 - s_1) = \text{area}(d\text{-}e\text{-}n_3\text{-}n_2\text{-}d)$

*Condenser*: $k = 2$, $m_{in2} = m_{out2} = 1$; $s_{out2} = s_3$, $s_{in2} = s_2$,
$Q_{add2} = q_k = h_2 - h_3$; $T_{add2} = T_{env}$

then $$s_{gen2} = s_3 - s_2 + \frac{h_2 - h_3}{T_{env}}$$

Let us reconstruct the area equal to extra work trough irreversibility (exergy destruction) in the condenser. The exergy destruction in the condenser is area($n_1$-$n_3$-2-2 -3-$n_1$). It is equal to an additional area(e-f-$n_4$-$n_3$-e) to the ideal Carnot-cycle. The unknown value of $s_f$ can be determined from $h_2 - h_3 = T_{env}(s_f - s_3)$. Thus, $s_f = s_3 + (h_2 - h_3)/T_{cp}$. The value of $s_f$ is necessary to determine $s_{gen2}$, therefore $s_{gen1} = s_f - s_2$ and finally $e_{D2} = T_{env}(s_f - s_1) =$ area(e-f-$n_4$-$n_3$-e).

*Throttle valve*: $k = 3$, $m_{in3} = m_{out3} = w$; $s_{in3} = s_3$, $s_{out3} = s_4$, $Q_{add3} = 0$
then $s_{gen3} = s_4 - s_3$
and $e_{D3} = T_{env}(s_4 - s_3) =$ area(a-b-$m_3$-$n_1$-a).

*Evaporator*: $k = 4$, $m_{in4} = m_{out4} = 1$; $s_{out4} = s_1$, $s_{in4} = s_4$, $Q_{add4} = q_E = (h_1 - h_4)$;
$T_{add4} = T_{cold}$

then $$s_{gen4} = s_1 - s_4 - \frac{h_1 - h_4}{T_{cold}}$$

The value $s_{gen4}$ for the ideal cycle is $s_{gen4} = s_1 - s_5$, therefore $e_{D4} = T_{env}(s_1 - s_5) =$ area(c-d-$n_2$-$m_2$-c). Finally, for the real refrigeration machine:

$$w_{real} = e_{cold} + e_{D1} + e_{D2} + e_{D3} + e_{D4} = h_2 - h_1$$

If the values of $s_{genk}$, $e_{Dk}$ are known, then the values of $y^*_{D\,tot}$, $y_{D\,tot}$ can be calculated and $\varepsilon$ can be calculated for the real cycle.

*Heat pump*

Figure 6b depicts the real cycle of vapor-compressor heat pump 1-2-2s-2 -3-4-1. The work of the cycle is:

$$w_{\mathrm{real}} = q_{\mathrm{C}} - q_{\mathrm{E}} = \text{area(e-2-2 -3-a-b-4-1-d-e)} = h_2 - h_1 \tag{33}$$

Let us construct an ideal Carnot-cycle to define the value of $E_{\mathrm{Dtot}}$ of an real heat pump cycle. The temperature levels of the heat sources for heat pump are $T_{\mathrm{hot}}$ and $T_{\mathrm{env}}$. The positive effect of a heat pump is the heat of condensation, therefore the specific heat capacity of an ideal Carnot-cycle must be equal to the specific heat capacity of the real:

$$q_{\mathrm{C}} = h_2 - h_3 = T_{\mathrm{hot}}(s_f - s_3) \tag{34}$$

where

$$s_f = s_3 + \frac{h_2 - h_3}{T_{\mathrm{hot}}}$$

The ideal Carnot cycle is $m_1$-$m_2$-$m_3$-$m_4$. The extra work (exergy destruction) of a real heat pump cycle is determined by eq. (31). To analyze the heat pump cycle by ECM we will keep the methodology proposed for refrigeration machine cycle. The total number of the elements is $n = 4$.

*Compressor*: $k = 1$, $m_{\mathrm{in1}} = m_{\mathrm{out1}} = 1$; $s_{\mathrm{out1}} = s_2$, $s_{\mathrm{in1}} = s_1$, $Q_{\mathrm{add1}} = 0$
then $s_{\mathrm{gen1}} = s_2 - s_1$
and $e_{\mathrm{D1}} = T_{\mathrm{env}}(s_2 - s_1)$ = area(d-e-$n_1$-$n_2$-d).

*Condenser:* $k=2$, $m_{\mathrm{in2}}=m_{\mathrm{out2}}=1$; $s_{\mathrm{out2}}=s_3, s_{\mathrm{in2}}=s_2$, $Q_{\mathrm{add2}}=q_k=h_2-h_3$; $T_{\mathrm{add2}}=T_{\mathrm{hot}}$

then $e_{\mathrm{D2}} = T_{\mathrm{env}}\left[(s_3 - s_2) + \dfrac{h_2 - h_3}{T_{\mathrm{hot}}}\right]$, where $\dfrac{h_2 - h_3}{T_{\mathrm{env}}} = s_f - s_3$,

and $e_{\mathrm{D2}} = T_{\mathrm{env}}(s_f - s_2)$ = area(e-f-$m_1$-$n_1$-e).

*Throttle valve*: $k = 3$, $m_{\mathrm{in3}} = m_{\mathrm{out3}} = w$; $s_{\mathrm{in3}} = s_3$, $s_{\mathrm{out3}} = s_4$, $Q_{\mathrm{add3}} = 0$
then $s_{\mathrm{gen3}} = s_4 - s_3$
and $e_{\mathrm{D3}} = T_{\mathrm{env}}(s_4 - s_3)$ = area(a-b-$m_4$-$n_4$-a).

*Evaporator*: $k = 4$, $m_{\mathrm{in4}} = m_{\mathrm{out4}} = 1$; $s_{\mathrm{out4}} = s_1$, $s_{\mathrm{in4}} = s_4$, $Q_{\mathrm{add4}} = -q_{\mathrm{E}} = -(h_1 - h_4)$; $T_{\mathrm{add4}} = T_{\mathrm{env}}$

then $e_{\mathrm{D4}} = T_{\mathrm{env}}\left[(s_1 - s_4) - \dfrac{h_1 - h_4}{T_{\mathrm{env}}}\right] = \text{area}(4\text{-}1\text{-}n_2\text{-}n_4\text{-}4)$

This area can be reconstructed by replacing it with the equivalent area (e-d-$n_2$-$n_3$-e). The value of $s_C$ is determined from $T_{env}[(s_1 \quad s_4) \quad (h_1 \quad h_4)/T_{env}] \quad T_{env}(s_1 \quad s_c)$, thus $s_c \quad s_4 \quad (h_1 \quad h_4)/T_{env}$. The specific exergy of the heat produced in a heat pump is $e_{hot} \quad q_C[1 \quad (T_{env}/T_{hot})]$. The value of the real specific work consumed in a heat pump cycle is:

$$w_{real} \quad w_{ideal} \quad \sum_{k \quad 1}^{n} e_{Dk} \quad e_{D1} \quad e_{D2} \quad e_{D3} \quad e_{D4} \tag{35a}$$

or

$$w_{real} \quad (h_2 \quad h_3) \; 1 \quad \frac{T_{env}}{T_{hot}} \quad T_{env}[(s_2 \quad s_1) \quad (s_3 \quad s_2)]\frac{T_{env}}{T_{hot}} \\ T_{env}[(s_4 \quad s_3) \quad (s_1 \quad s_4)] \quad h_2 \quad h_1 \tag{35b}$$

## Conclusions

The Entropy-Cycle Method created for compressor refrigeration machines and heat pumps can be transformed for heat using refrigeration machines and heat pumps (*i. e*. ejection and sorption) [12]. This paper considers the usefulness of the detailed methodology of the method. All irreversibilities can be described qualitatively and quantitatively in a real cycle to carry out real engineering calculations and design optimization. The Cycle Method is a general method for thermodynamic analysis in educational process of Refrigeration Machine Department of Odessa State Academy of Refrigeration. This method is used in the course "Theoretical bases of refrigeration engineering" for the bachelors. The Entropy Cycle Method is one of the methods of thermodynamic analysis of all types of refrigeration machines and heat pumps used for scientific work in Refrigeration Machine Department.

## Nomenclature

*COP* coefficient of performance, [ ]
*E* exergy, [W]
*e* specific exergy, [J/kg]
*h* specific enthalpy, [J/kg]
*m* mass flow rate, [kg/s]
*p* pressure, [Pa]
*Q* heat rate, [W]
*q* specific heat, [J/kg]
*S* entropy, [J/K]
*s* specific entropy, [J/kgK]
*T* temperature, [K]
*W* work, [J]
*w* specific work, [J/kg]
*y* – exergy destruction ratio, [–]
*y** – contribution of exergy destruction (entropy generation), [–]

*Greek symbols*

$\varepsilon$ exergy efficiency, [ ]
$\eta$ – thermodynamic efficiency, [ ]

*Subscripts*

C condenser
CM compressor
cold cold source
D destruction
E evaporator
EX expander
env environmental
F – fuel
gen generated
hot hot source
*i* flow
in inlet
*k* element of system
L losses
*n* number of elements
out outlet
P – product
tot – total

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