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The general exergy method of heating/cooling technology design for optimization

Malick Kane[a], Daniel Favrat[b]

[a] *HES-SO//FR, University of Applied Science in Fribourg, Mechanical Engineering Department (ENERGY institute), Fribourg, Switzerland, malick.kane@hefr.ch,* ***CA***
[b] *EPFL, Ecole Polytechnique Fédérale de Lausanne, Switzerland, daniel.favrat@epfl.ch*

Abstract:

A general exergy based-design method for optimization of heat pump/refrigeration systems is proposed. It is based on a concept of overall temperature level of a flow-energy, to propose a general expression of overall exergy efficiency and losses of any heat pump/refrigeration cycle. Explicit and general relations of exergy efficiency and coefficient of performances are given to evaluate de performance of such cycles regarding the selection of working fluids, the characteristic of equipment (pinches on evaporators and condensers, performance characteristics of compressors and expansion valve) and the design methods for optimization.

Rigorously we introduce the overall and complete exergy efficiency for the most general cases where two energy services are provided, like producing simultaneously refrigeration and heating services or when the cycle is located in a temperature domain far from the atmospheric temperature. This complete exergy efficiency is determined by considering losses in the various components of the cycle and permits to analyse the various cases of heat pump systems including frigopump and thermopump with or without cogeneration systems. Such a method will facilitate the use of exergy theory in a way to highlight the existing link and relationship between energy and exergy losses of heat pump systems. Results of using such a method will be shown for simple and advanced cycles. Results show that the coefficient of performance of a heat pump/refrigeration installation does not necessarily depend on the reference atmospheric temperature but only on the intrinsic parameters relating to the choice of cycle, the operating conditions and the components of the machine. These parameters are obviously chosen according to the temperature levels of the available sources.

Keywords:

Thermodynamics; Exergy losses; Exergy efficiency; Effectiveness; design and optimisation method

1. Introduction

"Pumping" heat from a lower temperature to an upper temperature can be done by using a vapor compression heat pump/refrigeration cycle most commonly used in heating and cooling applications. It consists of a process for which refrigerant vapor is compressed in the compressor and then used to the condenser where it is first cooled or desuperheated and then condensed and finally slightly subcooled. The saturated or slightly subcooled refrigerant is then expanded in a valve then evaporated in an evaporator and the cycle begins again. Whether operating in frigopump mode (providing cooling services while dissipating energy to the environment), in thermopump mode (providing heating service to customers while capturing energy from environment) or in cogeneration mode (providing heating and cooling simultaneously) the process is accompanied with different losses, resulting from various irreversibilities that occurred on their components. Generally, different approaches to assess exergy performances of such heating/cooling systems can be distinguished and applied at different levels [1, 2]:

- The method of calculating the exergy efficiency by quantitatively evaluating the global exergy losses $\dot{L}$ on the basis of internal/external losses [3, 4] (exergy losses $\dot{L}_D$ called also exergy destruction inside the strictly defined system and external exergy losses $\dot{L}_E$ or exergy destroyed between the system and the atmosphere) or on the basis of an exergy balance carried out on the boundary of the system extended to the atmosphere in order to internalize and attribute all losses to the system [1, 5]. Such a method is useful to compare different options or configurations of systems providing the same service (of heating and cooling);
- The assessment made on the basis of an exergy balance carried out on each of the components in the whole system in order to identify and locate the major losses. Borel & Favrat [5] subdivide $\dot{L}$ into:

$$\dot{L} = \dot{L}_r + \dot{L}_t$$

Where $\dot{L}_r$ includes the dissipation exergy losses (pressure losses and friction) inside the system and $\dot{L}_t$ includes the heat transfer exergy losses. The performance of a system is then characterised by calculating exergy losses of various components and/or subsystems.

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In this article, we use the approach of components/subsystems level analysis to propose a **general exergy based-design method for optimization** of heat pump/refrigeration systems. Such method makes it possible to identify and locate the major losses in the whole system in order to take optimization measures and to determine the exergy efficiency characteristics of any heat pump system. New method [6] using explicit and general relations of exergy efficiency and coefficient of performances will be given to evaluate de performance of such cycles regarding the selection of working fluids and the characteristic of equipment (pinches on evaporators and condensers, performance characteristics of compressors and expansion valve).

Simplified effectiveness (or COP-coefficient of performance) models of heat-pumps/refrigeration cycles using perfect gases on the compressor and/or equation-fit models have been proposed in many studies. [7] performed models to calculate the COP by using a theoretical Lorentz efficiency multiplied by a constant exergy efficiency (of around 40% for boosters HP and 50% for the centralized HP). [8] gives a review of recent development in variable refrigerant flow (VRF) systems with models ranging from detailed physics-based models to equation-fit models. [9] proposed a modular simplified model using perfect gas in compression for a fast multi-year simulation. [10] performed a detailed simulation model based-exergy performance characteristics of heat-pumps and applied by in [11] to simulate different configurations of network integration where the performances in term of exergy efficiency on the substation are determined in function of the temperature of the network and the differential temperature on the primary network.

This detailed process exergy-based design model uses real working fluids with the possibility of evaluating the exergy efficiency in function of all key parameters that influence the system. This method allows to give precise recommendations on the optimization of a real process in operation or on the decision support for the conception or the design of a heating or cooling system.

2. The general exergy model of heat pump cycles

2.1. The concept of overall temperature for the calculation of flow-exergy

A fluid stream carrying a flow-energy ($\dot{Y} = \dot{M}\,\Delta h$) at constant pressure ($P$) between two states (1 and 2) can be characterized by its overall temperature level ($\bar{T}$) or logarithmic mean temperature. This temperature value is calculated on the basis of Gibbs equation of enthalpy variation ($dh = Tds + vdP$), for which we can thus write $vdP = 0$ and the integration will give a perfect relationship between enthalpy/entropy variations:

$$\frac{h_2 - h_1}{\bar{T}} = s_2 - s_1 \qquad (1)$$

$$\bar{T} = (T_2 - T_1)\,/\,ln\left(\frac{T_2}{T_1}\right) \qquad (2)$$

The logarithmic mean value of two different quantities is well known in Physics as well as in Thermodynamics, especially the calculation of a heat exchanger area with the method of mean temperature differences (heat transfer between fluids at constant pressure and without phase change. Considering Eq. (1) and the definition of co-enthalpy variation ($\Delta k = \Delta h - T_a \Delta s$) of the stream, the corresponding flow-exergy ($\dot{E}_y$) could then be given by the following equation:

$$\dot{E}_y = \dot{M}\,\Delta k = \left(1 - \frac{T_a}{\bar{T}}\right)\cdot \dot{Y} \qquad with \quad P = cste \qquad (3)$$

This concept of logarithmic mean temperature (called here overall temperature) has been used and applied by various authors [12, 13] for the modelling of a stream flow-exergy. Indeed, a fluid stream can be seen as a succession of matter carrying exergy (co-enthalpy) at a temperature level $\bar{T}$. The latter is simply given by the variation of enthalpy of the fluid to the entropy variation (($\bar{T} = \Delta h / \Delta s$). The exergy losses associated to the flow-energy is given as:

$$\dot{L} = \dot{M}\,T_a\,(s_2 - s_1) = \dot{M}\,\frac{T_a}{\bar{T}}(h_2 - h_1) \qquad (4)$$

In the case of a phase change fluid in evolution at constant pressure, Eq. (3) is still valid and the process is subdivided in different phases e.g. superheating, bi-phasic and subcooling phases. The expression given in Eq. (1) becomes:

$$\frac{h_2 - h_1}{\bar{T}} = (s_2 - s_l) + (s_l - s_g) + (s_g - s_1) \qquad (5)$$

Where s_l and s_g represent respectively the liquid and gas saturated entropy.

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2.2. Energy/exergy balances of a standard cycle

The vapor compression heat pump or refrigeration cycle that we considered (Figure 1-a) is so far an open system comprising a compressor, a condenser, an expansion valve and an evaporator, with various streams and with the following hypotheses: kinetic and potential energies neglected, heat devaluation in the condenser and in the evaporator but no dissipation in the heat exchangers (heat transfer processes occurring at constant pressures) and adiabatic compression and expansion with dissipation in the compressor and in the valve. Figure 1-b shows a T-s diagram of the refrigerant working fluid.

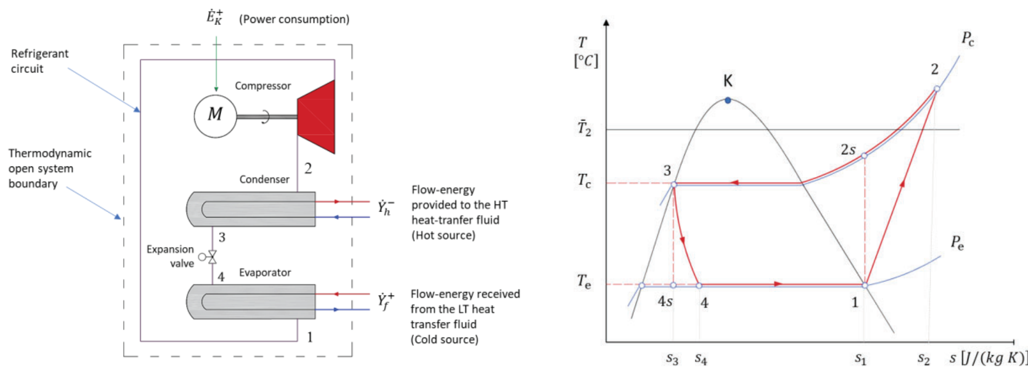

Figure. 1. Heating/cooling processes through a heat pump/refrigeration cycle
a) Schematic diagram, b) T-s diagram

The hot source is constituted by the HT heat transfer circuit in the condenser and the cold source is constituted by the LT heating circuit in the evaporator. The transformation (or flow) energy ($\dot{Y}_f^+$) received by the system from the cold source at a given temperature level ($\bar{T}_f$) is transferred through the heat pump/refrigeration system to the hot source ($\dot{Y}_h^-$) at a higher temperature level ($\bar{T}_h$). This is possible by consuming mechanical or electrical energy ($\dot{E}_K^+$) in the compressor. Note that the terms **hot** and **cold** here used to qualify the sources refer only to the temperature level of these sources. Not then to be confused with the **hot** and **cold streams** defined for heat exchangers where the cold stream is that of the flow to be heated by increasing its enthalpy level and the hot stream is that of the flow to be cooled by decreasing its enthalpy level. Thus,

- For the condenser, the hot stream is represented by the condensing refrigerant circuit at constant pressure (P_c) from state 2 to state 3. The overall temperature level of the stream is here noted ($\bar{T}_c$). The cold stream is represented by the HT heat transfer circuit heated in the condenser and supplying energy ($\dot{Y}_h^-$) to the hot source;
- For the evaporator, the hot stream is represented by the LT heat transfer circuit cooled in the evaporator and receiving energy ($\dot{Y}_f^+$) from the cold source. The cold stream is represented by the evaporating refrigerant circuit at constant pressure (P_c) from state 4 to state 1 ($\dot{Y}_e^-$).

These condensing or evaporating streams at constant pressure can be represented in the diagram T-s by their corresponding overall temperature values (or logarithmic mean temperature values at constant pressure), like $\bar{T}_c$ for the condenser and $\bar{T}_e$ for the evaporator. Figure 2 illustrates an example of this type of diagram with the advantage of showing the average pinches on heat exchangers characterizing the heat devaluation and the variations of entropies characterizing the dissipations in the compressor and the valve.

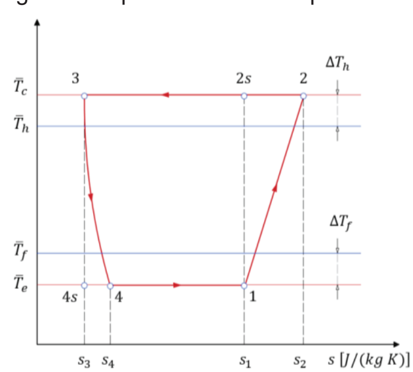

Figure. 2.: T-s diagram of a standard heat pump/refrigeration cycle

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For adiabatic components with steady state open operation and based on the energy balance equation (Eq. xx), we can give the following relations:

$$\dot{Y}_h^- = \dot{Y}_c^+ = \dot{M}\,(h_2 - h_3) \qquad \dot{Y}_f^+ = \dot{Y}_e^- = \dot{M}\,(h_1 - h_4) \tag{5}$$

$$\dot{E}_K^+ = \dot{Y}_h^- - \dot{Y}_f^+ = \dot{M}\,(h_2 - h_1) \tag{6}$$

2.3. Overall exergy efficiency and losses of the cycle

From the general expressions given in [6], the overall exergy efficiency of a vapor compression heat pump/refrigeration system can be given as a function of the temperature levels of the hot ($\bar{T}_h$) and cold ($\bar{T}$) sources:

$$\eta = 1 - T_a\left(\frac{\varepsilon_h}{\bar{T}_h} - \frac{\varepsilon_f}{\bar{T}_f}\right) \qquad with \qquad \varepsilon_h = \varepsilon_f + 1 \tag{7}$$

Where $\varepsilon_h = \dot{Y}_h^- / \dot{E}_K^+$ and $\varepsilon_f = \dot{Y}_f^+ / \dot{E}_K^+$ respectively represent the effectiveness (or coefficient of performance) for heating and cooling. Considering that the mass flow rate of refrigerant ($\dot{M}$) is the same throughout the different component of the standard cycle, we can here work with energy/exergy quantities per unit of mass of refrigerant. Effectiveness for heating and cooling can then be done by using the specific energies of cooling ($y_h^- = h_1 - h_4$), of heating ($y_f^+ = h_2 - h_3$) and of power consumption ($e_K^+ = h_2 - h_3$):

$$\varepsilon_h = \frac{h_2 - h_3}{e_K^+} \qquad \varepsilon_f = \frac{h_1 - h_4}{e_K^+} \tag{8}$$

From Eq. (7), we found the well-known and particular relationships between the overall exergy efficiency of a heat pump cycle, the coefficient of performance (effectiveness) and the ideal effectiveness based on the Carnot factor :

For a thermopump ($\bar{T}_f = T_a$):

$$\eta_h = \varepsilon_h\left(1 - \frac{T_a}{\bar{T}_h}\right) \qquad COP_h = \eta_h \cdot \frac{\bar{T}_h}{\bar{T}_h - T_a} \tag{9}$$

For a frigopump ($\bar{T}_h = T_a$):

$$\eta_f = \varepsilon_f\left(\frac{T_a}{\bar{T}_f} - 1\right) \qquad COP_f = \eta_f \cdot \frac{\bar{T}_f}{T_a - \bar{T}_f} \tag{10}$$

Considering that the mass flow rate of refrigerant ($\dot{M}$) is the same throughout the different component of the standard cycle, we can here work with energy/exergy quantities per unit of mass of refrigerant. Equations Eq. 5.12 can then be done by using the specific energies of cooling ($h_1 - h_4$), of heating ($h_2 - h_3$) and/or of power consumption ($e_K^+ = h_2 - h_3$). These values can directly be read in the P-h diagram of the cycle.

$$\varepsilon_h = \frac{h_2 - h_3}{e_K^+} \qquad \varepsilon_f = \frac{h_1 - h_4}{e_K^+} \tag{11}$$

Based on Eq. (7) and Eq. (8), Exergy losses per unit of mass (l_G) of the **complete heat pump or refrigeration system** are then:

$$l_G = (1 - \eta) \cdot e_K^+ = \frac{T_a}{\bar{T}_h}\,(h_2 - h_3) - \frac{T_a}{\bar{T}_f}\,(h_1 - h_4) \tag{12}$$

Two components of exergy losses can be distinguished: exergy losses that are rather related to both the temperature level of the hot source ($\bar{T}_h$) and the specific energy of heating ($h_2 - h_3$) and exergy losses that are related to the temperature level of the cold source ($\bar{T}_f$) and the specific energy of cooling ($h_1 - h_4$). The temperature levels of the sources therefore play an important role in the assessment of the overall exergy losses or irreversibilities in heat pumps. The exergy losses do decrease with the temperature level of the hot source while they increase with the temperature level of the cold source.

Such global exergy efficiency and losses given by Eq. (7) and Eq. (12) are very useful for comparing the performance of different options or configurations of heat pump systems but are not explicit enough to provide more specific recommendations on the optimization of the cycle and more particularly on the design and operation of the components but also on the choice and characteristics of the refrigerant.

2.4. Detailed exergy assessment of the heat pump cycle

Not that, the exergy losses by dissipation in the compressor and in the valve enter into the expression of the global exergy loss (Eq. 12) through the enthalpy values of h_2 and h_4. If the isentropic efficiency of the compressor (η_{Ks}) decreases, h_2 increases and so does the compressor dissipation loss. If the pressure at the condenser (P_c) increases, h_4 increases and so does the loss in the valve. In general, the temperature level of a source is initially a known parameter based on the need of heating ($\bar{T}_h$) or of cooling ($\bar{T}_f$). Thus, improving the performance of a heatpump/refrigeration cycle (COP) shall involve improving its overall exergy efficiency by minimizing exergy dissipation ($\dot{L}_r$) and devaluation ($\dot{L}_t$) losses in the system.

Let us now determine the expression of exergy efficiency and losses by carrying out a detailed exergy assessment of losses in various components of the system (compressor, condenser, evaporator, valve). The global exergy losses ($\dot{L}_G$) here can be calculated by separating the dissipation exergy losses in the compressor and in the expansion valve ($\dot{L}_r = \dot{L}_K + \dot{L}_V$) and the devaluation (or heat transfer) exergy losses in the condenser and in the evaporator ($\dot{L}_t = \dot{L}_C + \dot{L}_E$). Then, the exergy efficiency of the cycle can be developed in function of these losses (dissipation and devaluation) per unit of power:

$$\eta = \ 1 - \left(\frac{\dot{L}_r}{\dot{E}_K^+}\right) - \left(\frac{\dot{L}_t}{\dot{E}_K^+}\right) \tag{13}$$

Let us then quantify these specific losses per unit of power.

Modelling dissipation exergy losses (compressor and valve)

Exergy losses in the compressor ($\dot{L}_K$):

For a compressor, adiabatic compression is aimed at, i.e. with perfect thermal insulation from the external environment to obtain the best exergy efficiency. In order to determine the compressor dissipation exergy losses $\dot{L}_K = \dot{M}\, T_a(s_2 - s_1)$ we first need to determine the variation of entropy in compression. According to Eq. (1) and to the fact that $s_1 = s_{2s}$ (Figure 2), the difference in entropy of the compression ($s_2 - s_1$) can also be determined in function of the overall discharge temperature ($\bar{T}_d$), when considering the refrigerant stream at constant pressure P_c from state 2 to state 2s (i.e., $vdP = 0$). The following equations can thus be defined:

$$s_2 - s_1 = \frac{h_2 - h_{2s}}{\bar{T}_d} = \frac{1 - \eta_{Ks}}{\bar{T}_d}\,(h_2 - h_1) \tag{14}$$

$$\bar{T}_d = (T_2 - T_{2s}) \,/\, ln\left(\frac{T_2}{T_{2s}}\right) \tag{15}$$

Where η_{Ks} represents the isentropic efficiency of the compression and $1 - \eta_{Ks} = (h_2 - h_{2s})/(h_2 - h_1)$ represents the part of dissipation losses in the compression. The variations in the kinetic and potential energies of the fluid are neglected in relation to the work energy consumed by the compressor. We can finally express the dissipation exergy losses in the compressor by the following simple relation:

$$\dot{L}_K = \dot{M}\,\frac{T_a}{\bar{T}_d}\,(1 - \eta_{Ks})(h_2 - h_1) \tag{16}$$

The power consumption is:

$$\dot{E}_K^+ = \dot{M}\,\left(\frac{\Delta h_{Ks}}{\eta_{Ks}}\right) \qquad with \quad \Delta h_{Ks} = h_{2s} - h_1 \tag{17}$$

Exergy losses in the expansion valve ($\dot{L}_V$):

In the case of the **adiabatic expansion** in a valve, i.e., with perfect thermal insulation thus avoiding external heat losses, the enthalpy variation of the fluid is equal to zero (isenthalpic expansion, $h_3 = h_4$). According to Eq. (4) and to the fact that $s_3 = s_{4s}$ (Figure 2), the difference in entropy of the compression ($s_4 - s_3$) can also be determined in function of the evaporation temperature ($\bar{T}_e$), when considering the evaporating pressure P_e from state 4s to state 4 (i.e., $vdP = 0$). The following equation can thus be defined:

$$\dot{L}_V = \dot{M}\,T_a(s_4 - s_3) = -\dot{M}\,\frac{T_a}{T_e}\,\Delta h_{Ks} \qquad with \quad \Delta h_{Vs} = h_{4s} - h_3 \tag{18}$$

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From these Eq. (16), Eq. (17) and Eq. (18), we can deduce the specific dissipation losses as a function of the isentropic efficiency, a ratio of isentropic enthalpy variations between the valve and the compressor (here called isentropic expansion/compression ratio) and the overall discharge temperature level of the fluid at the outlet of the compressor:

$$\frac{\dot{L}_r}{\dot{E}_K^+} = \frac{T_a}{\overline{T}_d}\left(1-\eta_{Ks}\right) + \frac{T_a}{T_e}\left(\frac{-\Delta h_{Vs}}{\Delta h_{Ks}}\right)\cdot \eta_{Ks} \tag{19}$$

Such expression of dissipation exergy losses per unit of power can be given by the following simple equation:

$$\frac{\dot{L}_r}{\dot{E}_K^+} = \frac{T_a}{\overline{T}_d}\left(1-\eta_{Gs}\right) \tag{20}$$

Where we introduce a cycle global isentropic efficiency (η_{Gs}) which takes into account the total dissipation losses or irreversibilities in the compressor and in the expansion valve and therefore is intrinsic to the heatpump cycle:

$$\eta_{Gs} = \eta_{Ks}\left[1-\left(\frac{-\Delta h_{Vs}/T_e}{\Delta h_{Ks}/\overline{T}_d}\right)\right] \tag{21}$$

This relationship Eq. (20) clearly expresses the link between the exergy losses, the fluid temperature level (via the overall discharge temperature ($\overline{T}_d$) at the outlet of the compressor) and the dissipation losses in the whole cycle ($1-\eta_{Gs}$). For the same percentage of dissipation losses in the cycle ($1-\eta_{Gs}$), the dissipation exergy losses are lower the higher the discharge temperature level of the compressor ($\overline{T}_d$). The fluid temperature level plays an important role in the exergy assessment.

Modelling heat transfer exergy losses (condenser and evaporator)

From the general expressions given in [Kane, ecos 2023], the devaluation exergy losses on heat exchangers can be given as a function of the temperature levels of the streams.

Exergy losses in a condenser:

$$\dot{L}_C = \dot{M}\left[T_a\left(\frac{1}{\overline{T}_h}-\frac{1}{\overline{T}_c}\right)\right](h_2-h_3) \tag{22}$$

Exergy losses in an evaporator:

$$\dot{L}_E = \dot{M}\left[T_a\left(\frac{1}{\overline{T}_e}-\frac{1}{\overline{T}_f}\right)\right](h_1-h_4) \tag{23}$$

By combining equations Eq. (22), Eq. (23) and Eq. (8), we can determine the specific devaluation losses as a function of the effectiveness for heating and cooling and the overall temperature levels of the streams:

$$\frac{\dot{L}_t}{\dot{E}_K^+} = T_a\left[\left(\frac{1}{\overline{T}_h}-\frac{1}{\overline{T}_c}\right)\cdot\varepsilon_h + \left(\frac{1}{\overline{T}_e}-\frac{1}{\overline{T}_f}\right)\cdot\varepsilon_f\right] \tag{24}$$

Or then:

$$\frac{\dot{L}_t}{\dot{E}_K^+} = T_a\left[\frac{\varepsilon_h}{\overline{T}_h}\frac{\Delta T_h}{\overline{T}_c} + \frac{\varepsilon_f}{\overline{T}_f}\frac{\Delta T_f}{\overline{T}_e}\right] \tag{25}$$

The exergy losses by internal heat devaluation on the heat exchangers (condenser and evaporator) increase not only with the pinch differential temperatures of the hot (ΔT_h) and cold (ΔT_f) sources but are lower the higher the source temperature levels. The temperature therefore plays an important role in the assessment of the overall exergy losses or irreversibilities.

Let us know express the exergy efficiencies and effectiveness of a heatpump/refrigeration system explicitly in terms of all key parameters influencing the performance of the cycle: the overall discharge temperature level ($\overline{T}_d$) characterizes the type of the working fluid, the global isentropic efficiency (η_{Gs}) characterizes the losses by dissipation in the cycle and the pinch differential temperatures in the hot (ΔT_h) and cold (ΔT_f) sources characterize devaluation losses by heat transfer in the condenser and the evaporator.

2.5. Explicit relations between exergy efficiency and effectiveness

By developing the expression given in Eq. (24), the exergy loss by devaluation in a heat pump/refrigeration cycle can also be determined by the following relation:

$$\frac{\dot{L}_t}{\dot{E}_K^+} = T_a \left[\left(\frac{\varepsilon_h}{\bar{T}_h} - \frac{\varepsilon_f}{\bar{T}_f} \right) - \left(\frac{\varepsilon_h}{\bar{T}_c} - \frac{\varepsilon_f}{e_f} \right) \varepsilon_f \right] \tag{26}$$

Considering the general expression of exergy efficiency given by Eq. (7), we can deduct from Eq. (24) the expression of the system's devaluation exergy losses:

$$\frac{\dot{L}_t}{\dot{E}_K^+} = 1 - \eta - T_a \left(\frac{\varepsilon_h}{\bar{T}_c} - \frac{\varepsilon_f}{\bar{T}_f} \right) \tag{27}$$

By replacing equations Eq. (20) and Eq. (27) in Eq. (13) and by considering the general equation of overall exergy efficiency (Eq. 7), we found a simple expression between the effectiveness for heating ε_h) and cooling (ε_f), the overall temperature levels of the fluid in the evaporator (T_e), the condenser ($\bar{T}_c$) and the compressor ($\bar{T}_d$) and the global isentropic efficiency (η_{Gs}) defined above:

$$\frac{\varepsilon_h}{\bar{T}_c} - \frac{\varepsilon_f}{\bar{T}_e} = \frac{1 - \eta_{Gs}}{\bar{T}_d} \tag{28}$$

Such an expression Eq. (28) shows that the quantity or ratio $(1 - \eta_{Gs})/\bar{T}_d$ is in fact a decisive parameter for calculating the effectiveness (or coefficient of performance) of the cycle as a function of the evaporation and condensation temperature levels of the fluid. Considering the egality $\varepsilon_h = \varepsilon_f + 1$, we can finally deduce the following explicit expressions of effectiveness:

$$\varepsilon_h = \eta_{0h} \cdot \frac{\bar{T}_c}{\bar{T}_c - \bar{T}_e} \qquad with \qquad \eta_{0h} = 1 - \frac{\bar{T}_e}{\bar{T}_d}(1 - \eta_{Gs}) \tag{29}$$

$$\varepsilon_f = \eta_{0f} \cdot \frac{\bar{T}_e}{\bar{T}_c - \bar{T}_e} \qquad with \qquad \eta_{0f} = 1 - \frac{\bar{T}_c}{\bar{T}_d}(1 - \eta_{Gs}) \tag{30}$$

Where η_{0h} and η_{0f} correspond respectively to the exergy dissipation efficiency for the heating and the cooling application.

These expressions are similar to Eq. (9) and Eq. (10) for the coefficient of heating (COP_h) and for cooling (COP_f) given in function of the exergy efficiency for heating (η_h) and cooling (η_f) and the Carnot factors related to the temperature levels of the sources. The performances of a heat pump/refrigeration cycle are better than if it operates at higher temperatures. Dissipation exergy efficiency equal to unit ($\eta_{0h} = \eta_{0f} = 1$) corresponds to losses that are represented only by the internal heat transfer devaluation on heat exchangers. The final expressions for the exergy efficiency of a heat pump/refrigeration cycle are then determined by combining the equations of effectiveness for heating Eq. (9) and Eq. (29) and for cooling Eq. (10) and Eq. (30) and then become:

$$\eta_h = \eta_{0h} \cdot \frac{1 - \bar{T}_f/\bar{T}_h}{1 - \bar{T}_e/\bar{T}_c} \tag{31}$$

$$\eta_f = \eta_{0f} \cdot \frac{\bar{T}_h/\bar{T}_f - 1}{\bar{T}_c/\bar{T}_e - 1} \tag{32}$$

Where: $\bar{T}_c = \bar{T}_h + \Delta\bar{T}_h$ and $\bar{T}_e = \bar{T}_f - \Delta\bar{T}_f$ according to the diagram given in Figure 2.

These expressions Eq. (31) and Eq. (32) of exergy efficiency are explicit as a function of the various parameters that define the qualities that a heat pump/refrigeration cycle must have, namely: the temperature levels of the hot ($\bar{T}_h$) and cold ($\bar{T}_f$) sources, the pinch temperature difference characterizing the condenser (ΔT_h) and the evaporator (ΔT_f), the overall discharge temperature level of the compressor here expressing the type of the working fluid ($\bar{T}_d$) and the global isentropic efficiency (η_{Gs}) that expresses the dissipations in the compressor and the expansion valve.

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3. Results and discussion

3.1. The exergy-based design method for optimization

As already shown for the detailed exergy model above, the coefficient of performance of a standard heating (ε_h) or cooling (ε_f) heat pump cycle is determined by the explicit relations of the equations Eq. (29) and Eq. (30). The terms η_{0h} and η_{0f} are exergy efficiencies defined with respect to ideal Carnot efficiency expressed here as a function of the difference ($\Delta\overline{T} = \overline{T}_c - \overline{T}_e$) of condensation ($\overline{T}_c$) and evaporation temperature levels ($\overline{T}_e$). Indeed, it can be shown that there is a simple relationship between these two exergy efficiency values by considering their expressions given in Eq. (29) and Eq. (30):

$$\eta_{0h} - \eta_{0f} = \frac{\Delta\overline{T}}{\overline{T}_d} \cdot (1 - \eta_{Gs}) \tag{33}$$

Based on this formulation of Eq. (33), we express what we already knew in practice or from the theory of exergy, that the best heat pump or refrigeration cycles are those which will:

- Reduce the difference in temperature levels of the refrigerant cycle ($\Delta\overline{T}$);
- Minimize the dissipations at the level of the expansion valve and the compressor $(1 - \eta_{Gs})$
- Use a fluid with a higher discharge temperature at the compressor ($\overline{T}_d$) from the point of view of exergy analysis

For known evaporation and condensation temperature levels, these exergy dissipation efficiencies are fully determined if one knows the **overall isentropic efficiency** (η_{Gs}) and the compressor outlet **discharge temperature** ($\overline{T}_d$). The advantage is that they only depend on the parameters intrinsic to the refrigerant circuit. Knowing their values makes it possible to determine the cycle coefficient of performance in function of the evaporation and condensation temperature levels. Figure 3 shows an example of a refrigerant-ammonia ($R717$) with different temperature conditions (evaporation and condensation), the corresponding overall isentropic efficiencies and discharge temperatures. Knowing the temperature of evaporation ($T_e = -10°C$) and condensation ($T_c = 40°C$) streams for the heat pump, the overall isentropic efficiency can be determined, in the order of $\eta_{Gs} \cong 65\%$. The corresponding coefficient of performances based on Eq. (29) and Eq. (30) are in the order of $\varepsilon_h \cong 4.3$ for heating and $\varepsilon_f \cong 3.3$ for cooling.

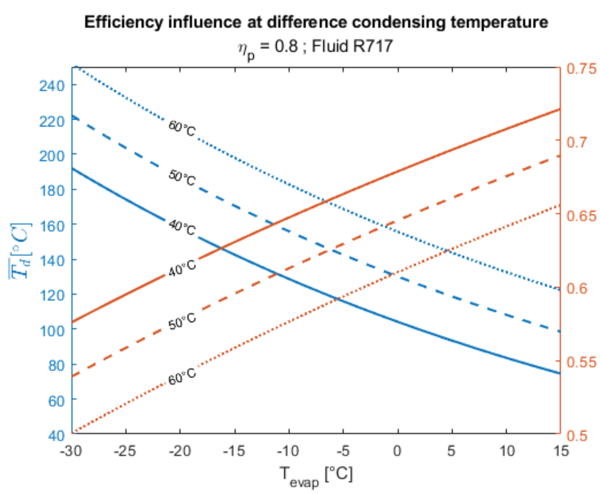

Figure. 3.: The effect of the evaporation temperature on the global dissipation efficiency of a heat pump cycle with parametric curves corresponding to the condensing temperature

Note that these quantities are a function of other parameters and are also dependent on each other:

- The difference in temperature levels of the cycle ($\Delta\overline{T}$) depends on the temperatures of the sources and the pinches at the evaporator and condenser and it is also influenced by the discharge temperature of the compressor;
- The overall isentropic efficiency of the cycle (η_{Gs}) defined above is also a function of the various characteristic parameters of the valve ($-\Delta h_{Vs}/\Delta h_{Ks}$), of the compressor ($\eta_{Ks}$) and of the refrigerant

fluid ($\bar{T}_d$). Figure 31 allows to highlight a dimensionless parameter enabling a fluid to be chosen if the operating conditions of evaporation and condensation and the isentropic efficiency of the compressor are known;

- Since the isentropic efficiency of the compressor is also not constant, it can be a function of various parameters including, for example, the built-in volume ratio (VR_i) and the compression ratio ($\pi_K = PR$) for volumetric compressors.

The thermodynamic optimization of the cycle based on the exergy criterion given by the Eq. (33) therefore makes it possible to obtain a better compromise between these different parameters. Different actions are mentioned theoretically in the literature or actually carried out in practice to improve the coefficient of performance of heat pump or refrigeration cycles and whose results find their explanation based on this analysis of the Eq. (33).

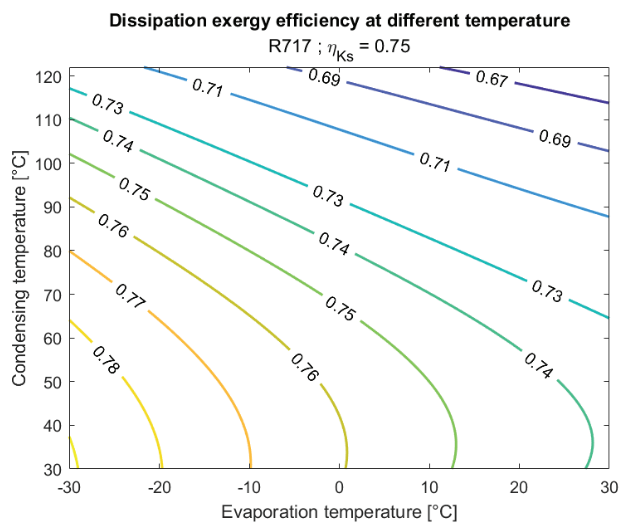

Figure. 4.: Dissipation exergy efficiency for heating in function of the condensing and evaporation temperatures

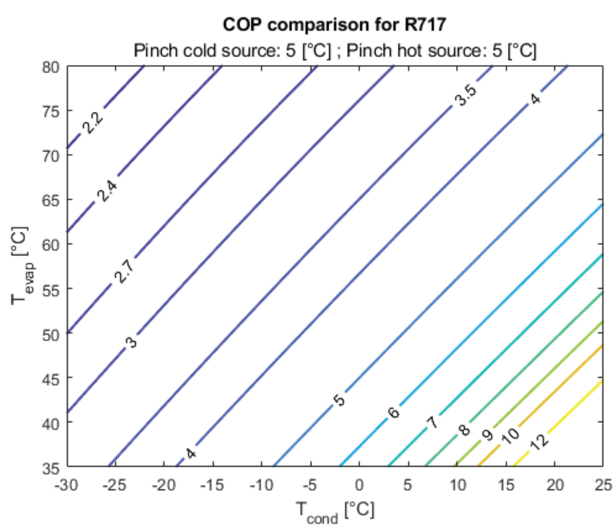

Figure. 5.: Coefficient of performance of heating, R717

These expressions and curves clearly show that the coefficient of performance of a heat pump/refrigeration installation does not necessarily depend on the reference atmospheric temperature but only on the intrinsic

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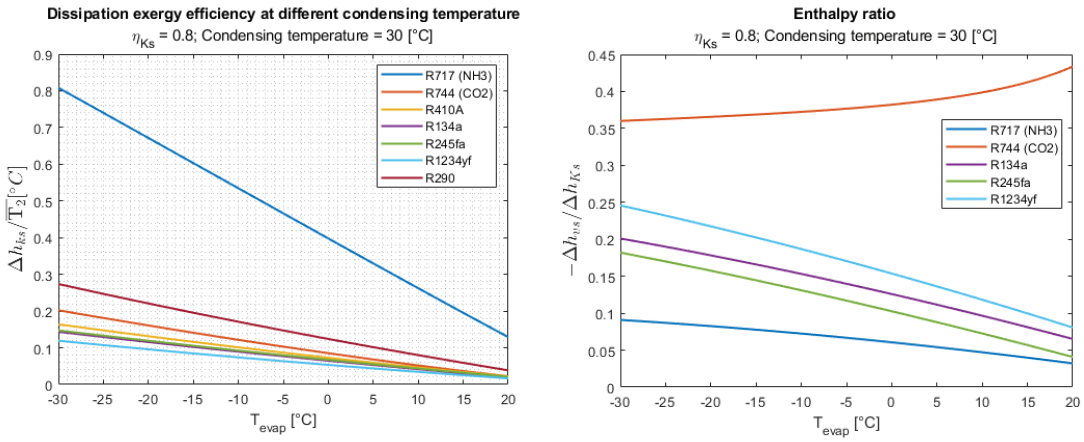
parameters relating to the choice of cycle, the operating conditions and the components of the machine. These parameters are obviously chosen according to the temperature levels of the available sources.

3.2. A dimensionless parameter to select the right working fluid

For the same evaporation and condensation temperatures, the specific power per unit of temperature of discharge compression ($\Delta h_{Ks}/\bar{T}_d$) is a decisive parameter for the selection of the working fluid for the compressor. On the other hand, for the valve, it is the isentropic expansion/compression ratio ($-\Delta h_{Vs}/\Delta h_{Ks}$) which is the determining parameter for the choice of fluid. Figure 6a and 6b show the variation of those parameters for various working fluids.



a) Specific power per unit of temperature b) Isentropic expansion/compression ratio

Figure. 6. Heat pump/refrigeration cycle with various working fluids

Refrigerants with the highest discharge temperature at the compressor e.g. with low molar mass refrigerants like ammonia also have the highest specific compression power per unit of discharge temperature ($\Delta h_{Ks}/\bar{T}_d$). The isentropic expansion/compression ratio ($-\Delta h_{Vs}/\Delta h_{Ks}$) is higher for CO2 than for other refrigerants.

An isentropic efficiency of $\eta_{Ks} = 80\%$ is considered for comparison. If this isentropic efficiency of the compressor decreases, the overall discharge temperature ($\bar{T}_d$) increases and so does the global dissipation loss (η_{Gs}). If the temperature at the evaporator (T_e) decreases, the loss in the valve increases too and so does the global dissipation loss (η_{Gs}). We identify an explicit dimensionless dissipation factor as a new criterion for estimating the exergy performance of the heatpump/refrigeration system:

$$a_s = \frac{-\Delta h_{Vs}/T_e}{\Delta h_{Ks}/\bar{T}_d} = \frac{1}{\eta_{Ks}} \cdot \left(\frac{\Delta s_V}{\Delta s_K}\right) \tag{34}$$

For same operating conditions of the cycle (evaporation and condensation temperatures), this dimensionless dissipation factor could be used as a good characteristic parameter to select the right working fluid. To illustrate this, we can visualize the effect of such a factor on the global isentropic efficiency (η_{Gs}) and for various refrigerants (Figure 7) and for different temperature levels (Figures 8).

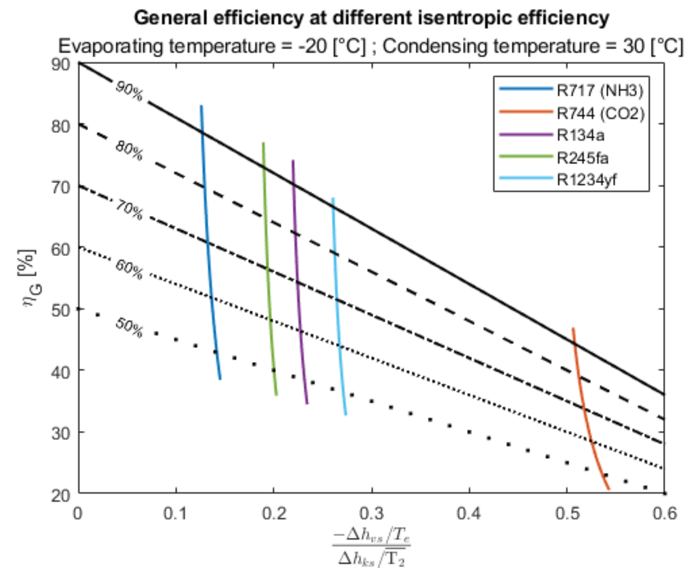


Figure. 7. Global isentropic efficiency in function of the dimensionless dissipation factor of a heatpump cycle and for various refrigerants (single stage cycle)

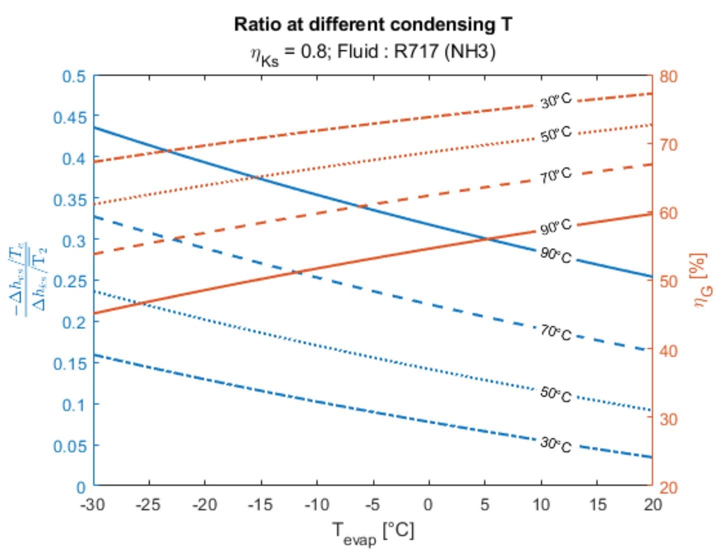


Figure. 8. Global isentropic efficiency in function of the dimensionless dissipation factor of a standard cycle working with R717-ammonia for different temperature levels

4. Conclusion

A **general exergy based-design method for optimization** of heat pump/refrigeration systems is proposed. It is based on a concept of overall temperature level of a flow-energy, to first propose a general expression of overall exergy efficiency and losses of any heat pump/refrigeration cycle.

Rigorously we introduced an overall and complete exergy efficiency for the most general cases where two energy services are provided, like producing simultaneously refrigeration and heating services or when the cycle is located in a temperature domain far from the atmospheric temperature. This complete exergy efficiency is determined by considering losses in the various components of the cycle and allows to analyse the various cases of heat pump systems including frigopump and thermopump with or without cogeneration systems. Main results of such a study are:

- Developing explicit and general relations of exergy efficiency and coefficient of performances of vapor compression heat pump/refrigeration systems. This allows to evaluate de performance of such cycles

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regarding the selection of working fluids and the characteristic of equipment (pinches on evaporators and condensers, performance characteristics of compressors and expansion valve);

- Providing an explicit dimensionless dissipation factor as a new criterion for estimating the exergy performance of a heatpump/refrigeration system. For same operating conditions of the cycle (evaporation and condensation temperatures), this dimensionless dissipation factor could be used as a good characteristic parameter to select the right working fluid with the definition of a global isentropic efficiency given in a function of the various characteristic parameters of the valve, the compressor and the heat exchangers;
- Developing a new exergy-based method for the design and optimization of heat pump/refrigeration systems. Such a method has the advantage of facilitating the use of exergy theory in a way to highlight the existing link and relationship between energy and exergy losses of heat pump systems.

Nomenclature

Roman symbols

E work energy, exergy, J

k co-enthalpy, J

L global exergy losses, J

$\dot{M}$ mass flow rate, kg/s

Q heat energy, J

Y flow or transformation energy, J

T temperature, K

U utility, energy service, J

Greek symbols

η exergy efficiency

ε effectiveness, coefficient of performance

Subscripts and superscripts

a ambient, atmosphere

C condenser

E evaporator

f cooling service

h heating service

K compressor

V Expansion valve

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