Universal Entropy Efficiency of Thermal Processes

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Abstract

Objectives: The main task of the estimation of power processes of any technical system is to define the efficiency of the consumed energy usage. Method: The most often used efficiency criterion is the ratio of the useful energy to the consumed energy defined from the power flow balances and the environmental losses. This criterion is defined primarily on the basis of the energy conservation law (more often on the primary case, i.e., on the basis of the 1st law of thermodynamics) and is generally called the coefficient of efficiency (COE). Findings: The article presents the entropy coefficient of efficiency. The entropy efficiency of a building depends on the inside and outside temperatures and the heat carrier temperature in the heating system, but only to the extent to which the heat emission and the heat conductivity coefficients depend on these temperatures. Taking these values to be constant, one can determine that the building efficiency depends only on the thermo physical properties of the radiators, walls and heat carriers. The method can be applied to obtain a common criterion of efficiency of complex heating or/and cooling buildings. If thermal energy converts into work, then entropy efficiency is equal to ratio of real cycle efficiency to thermal efficiency of Carnot cycle both for power and heat pump cycles. Also this method works for endoreversible cycles. Improvements: Entropy efficiency EnEf is the universal parameter of thermodynamic efficiency including COE and COP. EnEf criterion can be applied even in the cases when COE and COP have no sense.

Keywords: Entropy Efficiency, Heat Pump Cycle, Heating a Building, Power Cycle, Thermal Energy

1. Introduction

The main task of the estimation of power processes of any technical system is to define the efficiency of the consumed energy usage. The most often used efficiency criterion is the ratio of the useful energy to the consumed energy defined from the power flow balances and the environmental losses. This criterion is defined primarily on the basis of the energy conservation law (more often on the primary case, i.e., on the basis of the 1st law of thermodynamics) and is generally called the Coefficient Of Efficiency (COE). The Coefficient Of Performance (COP) that is not reducible to COE is used for the development of heat pump cycles. One of the significant drawbacks of the balance methods is the limited possibility to consider the peculiarities of non-equilibrium processes. These drawbacks are well-known; however, COE is widely used in science and technology because such criteria are easy to use and understand. The COE and COP criteria do not match, and several researchers have attempted to develop a universal coefficient of energy efficiency.

The energy required for heating buildings in many countries is approximately 40% of the total energy consumed in the residential sector. A significant proportion of this energy is used to maintain the temperature of inner air. During the calculation of the energy efficiency of a building, researchers often use the specific criterion of energy consumption/floor area. Also, researchers use different thermo physical properties to describe energy performance of buildings. Standard specific heat consumption is interconnected with the thermal performance of walls. Standard heat consumption depends on the substantially of the temperature of the outer air, of the standards of insulation, and of the required temperature of the inner air. The standards of heat consumption essentially differ depending on the climatic conditions.
Thus, standard specific heat consumption cannot be a satisfactory criterion of efficiency of thermal consumption.

Another method of evaluation is to determine the specific fuel consumption per m$^2$ of space\textsuperscript{13}. Another method is the comparison of actual heat consumption with a minimum heat of sanitary standards\textsuperscript{10}.

Using the balance methods leads to contradictory numerical results. The coefficient of efficiency values can either be smaller or greater than 100 %. The heat dissipation to the environment while heating and the heat consumption while cooling can be referred to as such processes. The distinctive peculiarity of such processes is the indefiniteness of useful energy because the process itself is directed to maintain a predetermined inside temperature (different from the outside one). The quantity of delivered heat $Q_1$ is equal to the quantity of rejected heat $Q_2$. The application of generally accepted indices of efficiency for such processes is impossible because dissipating and consuming heat cannot be considered to be useful. For example, under comparative evaluation of the heating processes of different effective buildings (i.e., having different heat resistance of enclosure constructions and radiators), the result will be $1 - Q_{\text{loss}} / Q_{\text{radiator}} = 0$ . Additionally, the balance methods do not take into account the quality of the consumed energy\textsuperscript{14}.

All real processes are naturally not at equilibrium; as a result, it is necessary to apply principals of non-equilibrium thermodynamics. The main terms of non-equilibrium thermodynamics are entropy and entropy production\textsuperscript{15,16}. Among the methods allowing for the consideration of the non-equilibrium nature of real processes, exergy analysis is frequently used\textsuperscript{17,18}. The exergy method accounts for the quality of the energy. The exergy method of the energy efficiency of a building often comes down to a comparison of the actual exergy losses with a minimum possible of sanitary standards\textsuperscript{7}. Additionally, the method of technically minimum exergy losses is used in\textsuperscript{19}.

Exergy analysis involves the fixing environmental parameters. The choice of the parameters can be performed by various criteria with different results. The exergy method must be used with the energy method simultaneously. However, the results of an exergy analysis often contradict the results of energy analysis\textsuperscript{20}. Exergy methods have several problems\textsuperscript{21}.

The definition of the energy efficiency of a heating process remains an unsolved problem. Because the application of the known criteria of efficiency for such process is not available or limited, it is necessary to develop another method that accounts for the non-equilibrium nature.

There are methods that account for the irreversibility of processes without a consideration of exergy. Prigozhin\textsuperscript{22}, on the basis of Onsager’s equations, proved the theorem regarding the minimum of entropy production. Ziegler\textsuperscript{23} represented the Maximum Entropy Production Principle (MEPP). Bejan\textsuperscript{24} developed the Entropy Generation Minimization principle (EGM). The Onsager equation related to cross-flows is not applicable in the consideration of heated spaces. The methods based on MEPP and EGM allow for producing significant results when describing non-equilibrium processes in the different fields of fundamental and application investigations.

The term “negentropy” or “negative entropy” was suggested in for information theory, which considers entropy to be a measure of disorder\textsuperscript{25,26}. In technical fields, the term negentropy is almost never used.

The application of the entropy production method will enable the development of a criterion that characterizes the degree of the thermal efficiency of buildings.

2. Heating a Building

Let us consider the simplest system of heating consisting of enclosure constructions (walls) and a radiator. This example of a heating system is shown in Figure 1.
resistance $R_r$, taking into account the heat exchange surface $A_r$. The building is characterized by the average temperature of the inside air $T_b$, which remains constant in the entire volume and by the thermal resistance coefficient $R_w$, which takes into account their surface $A_w$. The environment regarding the building is a thermostat with a given temperature $T_e$. The air inside of the building is considered to be equilibrium.

The system is supposed to be stationary. The heat carrier (zone 1) and inner (zone 3) and outer (zone 5) spaces are in a quasi-equilibrium state. Zones 2 and 4 are characterized with non-equilibrium heat exchange with the temperature change.\[\text{21}\]

Considering the stationary heat flow through the wall dividing two media, its power is described by the standard equation based on Fourier law:

$$Q = \frac{1}{R} (T_1 - T_2) \quad (1)$$

where $R$ is an average coefficient of the full thermal resistance, which takes into account the heat conductivity and the heat emission coefficients, the wall thickness and the heat exchange surface through which this heat flow runs, with dimensions of K/W; $T_1$ and $T_2$ are the temperatures of the media touching the wall.

The generally accepted formula of entropy production by this heat flow is as follows:

$$dS = \left(\frac{1}{T_2} - \frac{1}{T_1}\right) \cdot Q = \left(\frac{1}{T_2} - \frac{1}{T_1}\right) \cdot \frac{1}{R} \cdot (T_1 - T_2) \quad (2)$$

Let us rewrite the equation by introducing the parameter of the temperature ratio:

$$X = \frac{T_2}{T_1} \quad (3)$$

$$dS = \frac{1}{R} \frac{(1 - X)^2}{X} \quad (4)$$

The 1st stage of analysis is to define the building heat losses of a building equal to heat power of the radiators:

$$Q = \frac{1}{R_w} (T_b - T_e) = \frac{1}{R_r} (T_r - T_b) \quad (5)$$

where the average temperature of heat carrier $T_r$, which moves in the radiator and is in fact the internal source of energy for the system under consideration, is calculated.

The next stage is to define entropy production. Let us define entropy production in the process of heat transmission from heat element to inside air:

$$dS_r = \frac{1}{R_r} \frac{(1 - X_r)^2}{X_r} \quad (6)$$

where $X_r = \frac{T_b}{T_r}$.

The entropy production in the process of heat transmission from inside air to outside through enclosure constructions is as follows:

$$dS_w = \frac{1}{R_w} \frac{(1 - X_w)^2}{X_w} \quad (7)$$

where $X_w = \frac{T_e}{T_b}$.

In addition, let us introduce the minimum ratio of temperatures:

$$X_{\text{min}} = \frac{T_e}{T_r} \quad (8)$$

Let us introduce the term “thermally totally inefficient building” (Figure 2). Such a building does not have walls or has walls that have zero thermal resistance, which is mathematically equivalent. In this case, the temperature in the building will be equal to the environmental temperature ($T_b = T_e$).

Figure 2. Scheme of a thermally totally inefficient building. 1 – zone of the radiator; 2 – zone of the thermal resistance of the radiator; 3 – zone of the inner air; 4 – zone of zero thermal resistance of the walls; and 5 – zone of the outer air.
The entropy produced by the walls is accordingly equal to zero, and the total entropy production is defined by the thermal resistance of the radiator only and is a maximum:

\[ dS_w = dS_{\text{max}} = \frac{1}{R_e} \cdot \left(1 - X_{\text{min}}\right)^2 \]  

(8)

However, a “thermally totally efficient building” should be considered to be a building with adiabatic walls having an infinitely large value of thermal resistance and thus no heat losses. In this case, there are no heat flows, both to the inside and the outside air, and therefore, the entropy production is equal to 0. Because \( dS_{\text{max}} \leq dS \), the possibility arises to pass to the relative thermodynamic efficiency of the process considered:

\[ \frac{dS}{dS_{\text{max}}} = \frac{dS + dS_w}{dS_{\text{max}}} \]  

(9)

Let us modify the fraction in general form to:

\[ \frac{\left(1 - X\right)^2}{X} = \frac{1 - 2 \cdot X + X^2}{X} = \frac{1}{X} - 2 + X \]  

(11)

By substituting (6), (7) and (9) into (10) while accounting for the modification of (11), we obtain:

\[ \frac{dS}{dS_{\text{max}}} = \frac{1}{R_e} \cdot \left(\frac{T_e}{T_b} - 2 + \frac{T_b}{T_e}\right) + \frac{1}{R_w} \cdot \left(\frac{T_e}{T_e} - 2 + \frac{T_e}{T_e}\right) \]  

(12)

For the further simplification, let us multiply the numerator and the denominator of the obtained fraction by \( R_i \):

\[ \frac{dS}{dS_{\text{max}}} = \frac{\frac{T_e}{T_b} - 2 + \frac{T_b}{T_e} + \frac{R_e}{R_w} \cdot \frac{T_e}{T_e} - 2 + \frac{T_e}{T_e}}{\frac{T_e}{T_e} - 2 + \frac{T_e}{T_e}} \]  

(13)

After simplification of the received fraction, we obtain the following expression:

\[ \frac{dS}{dS_{\text{max}}} = \frac{T_e \cdot \left(T_e - T_b\right)^2 + T_e \cdot \left(T_b - T_e\right)^2 \cdot \frac{R_e}{R_w}}{T_b \cdot \left(T_b - T_e\right)^2} \]  

(14)

Next, divide numerator and denominator by \( \left(T_b - T_e\right)^2 \):

\[ dS = \frac{T_e \cdot \left(T_e - T_b\right)^2 + T_e \cdot \frac{R_e}{R_w}}{T_b \cdot \left(T_b - T_e\right)^2} \]  

(15)

From (1), it follows that \( \frac{T_e - T_b}{T_b - T_e} = \frac{R_e}{R_w} \), then:

\[ \frac{dS}{dS_{\text{max}}} = \frac{T_e \cdot \left(\frac{R_e}{R_w}\right)^2 + T_e \cdot \frac{R_e}{R_w}}{T_b \cdot \left(T_b - T_e\right)^2} \]  

(16)

Because \( T_e - T_e = (T_e - T_b) + (T_b - T_e) \) and \( \frac{T_e - T_b}{T_b - T_e} = \frac{R_e}{R_w} \), then:

\[ \frac{(T_e - T_e)^2}{(T_b - T_e)^2} = \left(\frac{R_e}{R_w} + 1\right)^2 \]  

(17)

By substituting (17) into the denominator of the fraction (16), we obtain:

\[ dS = \frac{T_e \cdot \left(\frac{R_e}{R_w}\right)^2 + T_e \cdot \frac{R_e}{R_w}}{T_b \cdot \left(\frac{R_e}{R_w} + 1\right)^2} \]  

(18)

For further simplification, let us express \( T_e \) from equation (1):

\[ T_e = T_b + T_b \cdot \frac{R_e}{R_w} - T_e \cdot \frac{R_e}{R_w} \]  

(19)

By substituting (19) into (18) we obtain:

\[ dS = \frac{T_e \cdot \left(\frac{R_e}{R_w}\right)^2 + \frac{T_e}{\frac{R_e}{R_w} + T_b} \cdot \frac{R_e}{R_w}}{T_b \cdot \left(\frac{R_e}{R_w} + 1\right)^2} \]  

(20)
\[
T_b \left( \frac{R_r}{R_w} + \left( \frac{R_r}{R_w} \right)^2 \right) = \left( \frac{R_r}{R_w} + 1 \right)^2
\]

\[
dS = 1 - \frac{1}{\frac{R_r}{R_w} + 1}
\]

Hence, it was found that there is a value that does not depend on the temperature (if taking into account that temperature does not influence the heat conductivity and heat emission coefficients): 27:

\[
\frac{dS_r + dS_w}{dS_{max}} = const
\]

This value increases with the growth of the total thermal resistance of the radiator and decreases with the growth of the total thermal resistance of the walls. For a totally inefficient building, the parameter takes is 0, and for a totally efficient one, it is 1.

Thus, the value of

\[
1 - \frac{dS}{dS_{max}} = \frac{dS_{max} - dS}{dS_{max}} = \frac{1}{\frac{R_r}{R_w} + 1}
\]

will range from 0 to 1.

The term of entropy efficiency was introduced in 28:

\[
\eta = \frac{\Delta N}{\Delta S} \leq 1
\]

where \(\Delta N\) is the negentropy change of the considered object, and \(\Delta N < \Delta S\).

In our case, it is suggested to associate a negentropy change with the difference of the maximally available entropy change and the given one:

\[
dN = dS_{max} - dS
\]

Then, we will obtain the entropy efficiency index in the following form:

\[
\eta_s = \frac{dN}{dS_{max}} = 1 - \frac{dS}{dS_{max}}
\]

Herewith, the negentropy production is equal to:

\[
dN = dS_{max} - (dS_r + dS_w)
\]

It is suggested to use the obtained index as an entropy efficiency coefficient of heating (or cooling) of a building:

\[
\eta_s = \frac{1}{\frac{R_r}{R_w} + 1}
\]

The physical sense of negentropy production by heating is to maintain the local temperature at a value higher or lower than the environmental temperature (\(T_b > T_e\)). Note that unlike ratio \(\frac{dN}{dS_{max}}\), the ratio \(\frac{dN}{dS} = \frac{dN}{dS_r + dS_w}\) is not constant and can have a value of either less than or greater than 1; therefore, it cannot be an efficiency criterion.

### 3. Results of Calculating Entropy and Negentropy Production

Consider the numerical simulation of the heated space. Let us assume the following thermophysical parameters: heat transfer coefficient of radiator is 11.5 W/(m²*K), full surface of the radiators is 20 m² 29 , full thermal resistance is \(R_r=0.00435\) (m²*K)/W, coefficient of thermal resistance of walls is 3.5 (m²*K)/W for Transbaikal's climate conditions 10 , and the surface of walls is 900 m². Such parameters keep the inner temperature at 20°C when the temperature of the outer air is -40°C and the temperature in the radiator is 87.1°C. The costs of thermal energy are 15428 W.

**Figure 3.** Outer, inner and radiator temperatures.
Variant 1. Temperature in the radiator automatically changes to maintain the inner temperature (Figure 3)

Because the thermophysical parameters of the walls and radiators are assumed to be constant, the heat load of the radiator varies linearly (Figure 4)

![Figure 4. The heat load of the radiator.](image1)

The influence of the change of the temperature in the radiator on the values of $dS_r$, $dS_w$, $dS$, $dS_{\text{max}}$ and $dN$ is shown in Figure 5.

![Figure 5. Entropy and negentropy production for variant 1.](image2)

The relative values change, as shown in Figure 6.

Variant 2. The temperature of radiator is constant, so the inner temperature changes.

Influence of the change of the inner temperature on the values of $dS_r$, $dS_w$, $dS$, $dS_{\text{max}}$ and $dN$ is shown in Figure 7.

![Figure 7. Entropy and negentropy production for variant 2.](image3)

![Figure 8. Ratios for variant 2.](image4)
The relative values change, as shown in Figure 8.

Variant 3. When the outer temperature is constant, the inner temperature depends on the temperature of the radiator.

The data from the Figures 9 and 10 show that all values - dSr, dSw, dS, dS_max, and dN – depend on the temperatures of the heat carriers. The change of temperature affects all of the relative values, except dN/dS_max, which is constant and depends on the values of thermal resistance, as in shown (10-27).

![Figure 9. Entropy and negentropy production for variant 3.](image1)

![Figure 10. Ratios for variant 3.](image2)

The effect of thermal resistance is shown in Figure 11, when T_e=-40°C and T_b=18°C. The coefficient of thermal resistance of the radiator is constant (does not depend on temperature) and the surface of the radiator changes.

![Figure 11. The effect of heat transfer of walls.](image3)

where, k_w=1/R_w. Due to (27), the entropy efficiency approaches 1 because if R_r→0 and R_w→∞ (k_w→0), then the ratio R_r/R_w→0. This behavior means that the entropy efficiency increases when the thermal resistance of the walls and the heat transfer from the radiator increase. Otherwise, if R_r→∞ and R_w→0, then the ratio R_r/R_w→∞, its entropy efficiency approaches 0, and the building becomes thermally absolutely inefficient, as shown in Figure 2. Note that the determining factor is the ratio of the thermal resistance.

4. Thermal Cycles

Having got the general form of entropy efficiency it is possible to extend the borders of this method appliance looking at systems which convert heat into work and vice versa.

4.1 Power Cycle

The power cycle works within the temperature range from T_1 to T_2 (Figure 12). Heat delivers from the hot reservoir to the cycle in Q_1 quantity and be removed to cold reservoir in Q_2 one, useful work taking place during the process L=Q_1 - Q_2.

Entropy production in the process of expansion will be as follows:

\[
dS = -\frac{Q_1}{T_1} + \frac{Q_2}{T_2}
\]  

(28)
As is known ideal adiabatic process of expansion is isoentropic, i.e. \( \frac{Q_1}{T_1} = \frac{Q_2}{T_2} \) and \( dS = 0 \). Real process is characterized by inequation \( dS > 0 \).

In case if useful work does not take place, i.e. \( Q_2 = Q_1 \), all delivered high-grade thermal energy dissipates ineffectively into low-grade environment, entropy change being maximum:

\[
dS_{\text{max}} = \frac{Q_1}{T_2} + \frac{Q_1}{T_2} \tag{29}
\]

Difference between maximum and current entropy change will be called negentropy. This term will be associated with conversion of thermal energy into another form, in this case into mechanical energy:

\[
dN = dS_{\text{max}} - dS = \frac{Q_1}{T_2} - \frac{Q_2}{T_2} = \frac{L}{T_2} \tag{30}
\]

Entropy efficiency will be as follows:

\[
\frac{dN}{dS_{\text{max}}} = \frac{dS_{\text{max}} - dS}{dS_{\text{max}}} = 1 - \frac{dS}{dS_{\text{max}}} \tag{31}
\]

after that it is easy to get the following dependence:

\[
\frac{dN}{dS} = \frac{\frac{Q_1}{T_2} - \frac{Q_2}{T_2}}{\frac{Q_1}{T_2} + \frac{Q_2}{T_2}} = \frac{Q_2 - Q_1}{Q_2 + Q_1} \tag{32}
\]

\[
\frac{dN}{dS_{\text{max}}} = \frac{\left( \frac{Q_1}{T_2} - \frac{Q_2}{T_2} \right) T_2}{\left( \frac{Q_1}{T_2} + \frac{Q_2}{T_2} \right) T_2} = \frac{1 - \frac{Q_2}{Q_1}}{1 - \frac{T_2}{T_1}} = \frac{1}{\eta_c} \tag{33}
\]

Hence entropy efficiency of power cycle is equal to ratio of real cycle efficiency \( \eta_R \) to thermal efficiency of Carnot cycle \( \eta_C \) for the given temperature range. In other words, entropy efficiency shows how the real cycle approaches to ideal Carnot cycle. Sometimes this ratio \( \frac{\eta_R}{\eta_C} \) is called: “second law efficiency”\(^{30}\). But as usual “second law efficiency” means exergy efficiency.

### 4.2 Endoreversible Power Cycle

Endoreversible cycle uses temperatures of heat and cold sources which different from temperatures of the cycle. Let us consider the endoreversible power cycle which works within the temperature range from \( T_1 \) to \( T_2 \) and temperatures of reservoirs are \( T_{10} \) and \( T_{20} \). Let heat be delivered from the hot reservoir to the cycle in \( Q_1 \) quantity and be removed to cold reservoir in \( Q_2 \), useful work taking place during the process \( L = Q_1 - Q_2 \).

Then entropy production of heat transfer from hot reservoir to the cycle is

\[
dS_{10} = Q_1 \cdot \left( \frac{1}{T_1} - \frac{1}{T_{10}} \right) \tag{34}
\]

Entropy production of heat transfer from the cycle to cold reservoir is:

\[
dS_{20} = Q_2 \cdot \left( \frac{1}{T_{20}} - \frac{1}{T_2} \right) \tag{35}
\]

Entropy production in the cycle is:

\[
dS_{12} = \frac{Q_2}{T_2} - \frac{Q_1}{T_1} \tag{36}
\]

Maximum entropy production is:

\[
dS_{\text{max}} = Q_1 \cdot \left( \frac{1}{T_1} - \frac{1}{T_{10}} \right) + Q_1 \cdot \left( \frac{1}{T_{20}} - \frac{1}{T_2} \right) + Q_1 \cdot \left( \frac{1}{T_2} - \frac{1}{T_1} \right) \tag{37}
\]

because \( Q_2 = Q_1 \) and \( L = 0 \). And we obtain:

\[
dS_{\text{max}} = Q_1 \cdot \left( \frac{1}{T_{20}} - \frac{1}{T_{10}} \right) \tag{38}
\]

Entropy production in the process of expansion is:

\[
dS = dS_{10} + dS_{12} + dS_{10} = Q_1 \cdot \left( \frac{1}{T_1} - \frac{1}{T_{10}} \right) + Q_1 \cdot \left( \frac{1}{T_{20}} - \frac{1}{T_2} \right) + Q_1 \cdot \left( \frac{1}{T_2} - \frac{1}{T_1} \right) \tag{39}
\]

Negentropy production will be as follows:

\[
dN = dS_{\text{max}} - dS = Q_1 \cdot \left( \frac{1}{T_{20}} - \frac{1}{T_{10}} \right) - \frac{Q_1}{T_{10}} \cdot \frac{Q_1}{T_{20}} \cdot \frac{Q_1}{T_1} - \frac{Q_1}{T_1} \cdot \frac{Q_1}{T_{20}} - \frac{Q_1}{T_{20}} \cdot \frac{Q_1}{T_2} \tag{40}
\]
Now we obtain:

\[
\frac{dN}{dS} = \frac{Q_1 - Q_2}{Q_2 T_20 - T_10 Q_1} \quad (41)
\]

and entropy efficiency will take the next form:

\[
\frac{dN}{dS_{\text{max}}} = 1 - \frac{Q_2}{T_20} \quad (42)
\]

Hence entropy efficiency of endoreversible power cycle is equal to ratio of real cycle efficiency \( \eta_R \) to thermal efficiency of Carnot cycle \( \eta_C \) for the temperatures of reservoirs.

### 4.3 Heat Pump Cycle

The heat pump cycle works in the temperature range from \( T_2 \) to \( T_1 \) where \( T_2 < T_1 \) (Figure 13). Delivered heat equals \( Q_2 \); work spent on compressor driving equals \( L \); removed heat takes value \( Q_1 = L + Q_2 \). Entropy production in the compression process is as follows:

\[
Q_1 = Q_2 + L + Q_2 \quad (43)
\]

If external work irreversibly converts into low-grade heat and dissipates, temperature raise does not take place \( (T_2 = T_1) \). In this case entropy production is maximum:

\[
dS_{\text{max}} = -\frac{L}{T_2} = \frac{Q_1}{T_2} - \frac{Q_2}{T_1} \quad (44)
\]

Difference between maximum and current entropy change will be similarly called negentropy. In this case it will be associated with heat passage from low-grade reservoir to high-grade one with account of post-heating by means of conversion work into heat:

\[
dN = dS_{\text{max}} - dS = \frac{Q_1}{T_2} - \frac{Q_2}{T_2} + \frac{Q_2}{T_1} - \frac{Q_1}{T_1} \quad (45)
\]

Entropy efficiency in general case will take the similar form of previous one:

\[
\frac{dN}{dS_{\text{max}}} = 1 - \frac{dS}{dS_{\text{max}}} \quad (46)
\]

and further:

\[
\frac{dN}{dS} = \frac{T_1 - T_2}{T_2 - \frac{Q_1}{Q_1}} \quad (47)
\]

\[
\frac{dN}{dS_{\text{max}}} = \left( \frac{Q_1}{T_1} - \frac{Q_1}{T_2} \right) \frac{T_1}{Q_1} = \frac{T_1 - T_2}{T_1 - T_2} \frac{Q_1 - Q_2}{Q_1} \frac{Q_2}{Q_1} \eta_{\text{COP}} \quad (48)
\]

Hence entropy efficiency of heat pump cycle is equal to ratio of real cycle \( \eta_{\text{COP}} \) to the reversed Carnot cycle \( \eta_{\text{COP}} \) for the given temperature range. As in the previous example, entropy efficiency shows how perfect the real cycle is towards the reversed Carnot cycle in the given temperature range. It is interesting that \( T \) and \( Q \) are swapping in ratios \( dN/dS \) for power cycle (32) and for heat pump cycle (46).

### 5. Conclusions

The increase of the thermal resistance of the walls increases the entropy efficiency, but the effect is higher for a lower full thermal resistance of the radiators. A large surface of radiators allows for a low temperature of the heat carrier; a low temperature is thermodynamically advantageous because the heating of the space expends a low-potential thermal energy. The negative effect of this case is the increase of the hydraulic resistance of the heating system, with the corresponding increase of energy consumption for pumps, which increases the metal consumption. This aspect requires further study and is not considered in this article.

The entropy efficiency of a building depends on the inside and outside temperatures and the heat carrier temperature in the heating system, but only to the extent to which the heat emission and the heat conductivity coeffi-
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cients depend on these temperatures. Taking these values to be constant, one can determine that the building efficiency depends only on the thermophysical properties of the radiators, walls and heat carriers. The obtained formula is valid for an isothermal building inside of which there are no heat flows. The method can be applied to obtain a common criterion of efficiency of complex heating or/and cooling buildings.

If a significant change of the flow regime occurs, this affects the conditions of heat transfer from the heat carrier to the wall of radiator, which changes the thermal resistance of the radiator and the entropic efficiency of the entire system. Additionally, the entropic efficiency is reduced because of internal flows if one considers a complex building with several spaces of different temperatures in each. These calculations require further study.

If thermal energy converts into work, then entropy efficiency is equal to ratio of real cycle efficiency to thermal efficiency of Carnot cycle both for power and heat pump cycles. Also this method works for endoreversible cycles.

Based on the above, it is possible to make a conclusion that entropy efficiency EnEf is the universal parameter of thermodynamic efficiency including COE and COP. EnEf is not a classic coefficient of energy efficiency. EnEf criterion can be applied even in the cases when COE and COP have no sense. The suggested entropy method of thermodynamic efficiency definition of power engineering objects gives an opportunity to calculate relative degree of thermal efficiency of different systems by means of one universal method. The non-equilibrium nature of thermal processes in the terms of the local equilibrium concept is taken into account. The EnEf criterion increases when an actual entropy production decreases and requires knowledge of the maximum entropy production. Therefore, the entropy efficiency principle considers both EGM and MEPP. This method makes it possible to use the non-linear dependences of heat flows from the temperatures involved. Universal entropy efficiency criterion can be used in different fields of sciences.

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