Roman Domański and Giuma Fellah

Institute of Heat Engineering
Warsaw University of Technology. Warsaw, Poland

EXERGY AS A TOOL FOR DESIGNING
AND OPERATING THERMAL STORAGE UNITS

The paper presents the development, state-of-the-art, and applications of exergy analysis in the evaluation of the performance of thermal energy storage units. Formulas dealing with different sources of irreversibilities for complete charging-discharging cycles are introduced. The contributions of the most popular authors to the subject are focused on. Comparison of the present authors' results related to sensible heat storage units, utilizing a relatively simple iteration technique with that using the well-known optimization code „GRG2", shows an excellent agreement.

NOMENCLATURE

A — surface area of the heat exchanger duct, m^2

c_P — constant pressure specific heat, $\dfrac{J}{(kg \cdot K)}$

c_V — constant volume specific heat, $\dfrac{J}{(kg \cdot K)}$

k — specific heat ratio $\left(\dfrac{c_P}{c_V}\right)$

C — specific heat of storage material, $\dfrac{J}{(kg \cdot K)}$

V — velocity, $\dfrac{m}{s}$

Z — elevation, m

G — mass flow rate per unit cross-section of duct area, $\dfrac{kg}{(m^2 \cdot s)}$

$\bar{G}$ — dimensionless mass velocity $\left(G \dfrac{\sqrt{RT_0}}{P_0}\right)$

H — enthalpy per unit mass, $\dfrac{J}{kg}$

$\dot{m}_{s(R)}$ — mass flow rate of gas during charging (discharging), $\dfrac{\text{kg}}{\text{s}}$

M — mass of the storage material, kg

N — entropy generation number

NTU — number of heat transfer units $\left(\dfrac{U_s A}{(\dot{m}_s c_P)}\right)$

$y_{s(R)}$ — heat exchanger parameter, charging (discharging)

P — pressure, $\dfrac{\text{N}}{\text{m}^2}$

Pr — Prandtl number $\left(\dfrac{c_P \mu}{k}\right)$

PCM — phase change material

R — gas constant, $\dfrac{\text{J}}{\text{kg·K}}$

s — entropy per unit mass, $\dfrac{\text{J}}{\text{kg·K}}$

S — entropy, $\dfrac{\text{J}}{\text{K}}$

t — time, s

T — temperature, K

U — overall heat transfer coefficient, $\dfrac{\text{W}}{(\text{m}^2 \cdot \text{K})}$

I — exergy destruction, J

$\dot{Q}$ — rate of heat transfer, W

C^* — equivalent heat capacity, $\dfrac{\text{J}}{(\text{kg·K})}$

GREEK SYMBOLS

β — tare capacity $\left(\dfrac{T_I - T_{iR}}{T_{iR}}\right)$

μ — dynamic viscosity, $\dfrac{\text{N·s}}{\text{m}^2}$

θ — dimensionless time $\left(\dfrac{\dot{m}_s c_P}{(MC)}t\right)$

τ — dimensionless temperature $\left(\dfrac{T - T_0}{T_0}\right)$

η — first law efficiency

η_{2nd} — second law efficiency

ψ — exergy, $\dfrac{J}{kg}$

Ψ — exergy, J

SUPERSCRIPT

— — dimensionless quantity

SUBSCRIPTS

cv — control volume
gen — generation
opt — optimum
e — exit
i — inlet
I — initial
d — destruction
F — final
0 — ambient condition
g — gas
R — discharging
S — storage
m — melting
is — inlet-charging
iR — inlet-discharging
es — exit-charging
eR — exit-discharging

INTRODUCTION

Energy conservation is a key goal of our economy now and will continue to be in the future. Many contemporary scientists recognize that the second law of thermodynamics can help them discover where our available energy is inefficiently used [1]. Exergy is defined as the maximum-useful-reversible work that can be obtained from a system at a given state [2]. To reach this value, the system must be brought in a reversible manner to a state of thermodynamic equilibrium (thermally, mechanically, and chemically) with the common components of its surrounding nature [3,4]. It may be considered to be one of our natural resources, found in such forms as oil reserves, coal reserves, and uranium reserves [2]. Thus, destruction of exergy must be minimized through a detection and evaluation of irreversibilities in our thermal processes. This concept may be applied in the field of thermal storage units through conscious selection

of their designing and operational parameters such as number of heat transfer units, charging temperature, melting temperature (for phase-change thermal storage units), mass flow rate, mass ratio, etc.

1. THE PROGRESS IN EXERGY TECHNIQUE IN THE FIELD OF THERMAL STORAGE UNITS

The word „energy" can offer to us two different impressions. The first is dealing with energy as a quantity, which is conservative, the second — as a quality, which is not conservative. Because of these two impressions, Z. Rant in 1956 introduced a new word „exergy" to express the quality of energy [3,4]. The degradation in exergy can be expressed by Gouy-Stodola law. According to which, the exergy destruction for a certain process is evaluated by the product of the sum of entropy increase $\sum \Delta S$ of all the bodies taking part in the process, and the temperature of the surrounding nature T_0 [3,4]

$$I = T_0 \sum \Delta S \tag{1}$$

Exergy of a unit mass of flowing fluid may be expressed as [2]

$$\psi = \left[(H - H_0) - T_0(s - s_0) + g(z - z_0) + \frac{V^2}{2} \right] \tag{2}$$

while the rate of the entropy generation for a uniform-state, uniform-flow may be given by [2]

$$\dot{S}_{gen} = \frac{dS_{cv}}{dt} + \sum \dot{m}_e s_e - \sum \dot{m}_i s_i - \frac{\dot{Q}_0}{T_0} \tag{3}$$

here $\dot{Q}_0$ is the rate of heat transfer from surroundings to a system. The earliest contributions to the exergy concept are due to R. Clausius (1865), P. G. Tait (1868), W. Thomson (Lord Kelvin), J. W. Gibbs (1873), and J. C. Maxwell (1875). This early work, as well as the succeeding elaboration by G. Gouy (1889), A. Stodola (1898), G. A. Goudenough (1911), and G. Darrieus (1930), induced little concern [5,6]. The latest evolution of exergy analysis was established by F. Bosnjakovic in Europe and J. H. Keenan in the United States. In the 1950s and 1960s, contributions to the exergy concept were also made by Z. Rant, P. Grassmann, W. M. Brodyansky, E. A. Bruges, M. Tribus, E. F. Obert, R. A. Gaggioli, R. B. Evans, W. Fratzscher, J. Szargut, R. Petela, and K. F. Knoch, among others [5,6].

Bejan [7,8] introduced the concept of entropy generation units as a means for evaluating the performance of heat exchangers. Saborio et al. [9] made an extension to the irreversibility minimization analysis applied to heat exchangers to include a term accounts for the exergy of the material of construction of the heat exchanger. They claim their analysis provides more realistic results. Bejan
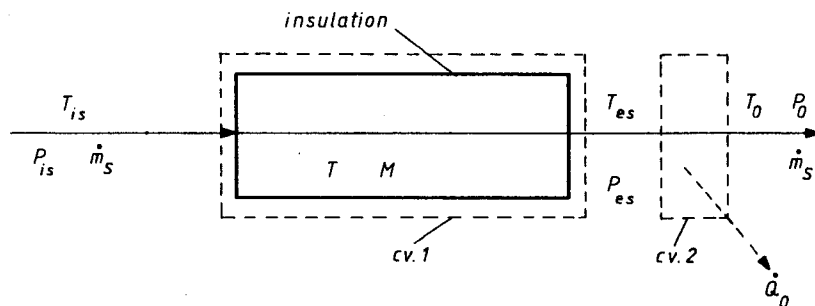

Fig. 1. Bejan's model

[10-12] presented a treatment of a simple sensible heat thermal energy storage unit, as shown in Fig.1. The analysis was based on the lump system of analysis and indicated the existence of two thermodynamic optima in designing sensible storage unit. The first, an optimum charging time θ_{opt}, beyond which the loss in exergy associated with steadily discharging T_{es} gas into the atmosphere becomes dominant. The second, an optimum number of heat transfer units NTU_{opt}, above which the loss of exergy due to friction in the working fluid side becomes dominant. Here we have

$$\theta = \frac{\dot{m}_S c_P t}{MC} \tag{4}$$

$$NTU = \frac{UA}{\dot{m}_S c_P} \tag{5}$$

His analysis was limited to a process of charging only, and pointed out thermal energy, to be stored at the instance of maximum second law efficiency, is only 50% to 70% of the maximum energy storage capability. He introduced an entropy generation number N, as a measure of thermodynamic irreversibilities, such that

$$N = \frac{T_0 \sum S_{gen}}{\Psi_i} \tag{6}$$

The second law efficiency can be written as

$$\eta_{2nd} = 1 - N \tag{7}$$

The sources of irreversibility in Bejan's system were due to finite temperature difference between the working fluid and storing material, frictional losses in the gas side, and exergy waste to atmosphere through a rejection of the working fluid to the surroundings at T_0. Bejan concluded that for best conditions the amount of exergy destruction is about 50% from the inlet exergy and the optimum charging time θ_{opt} is approaching the unity for the best second law efficiency. He claimed that the thermodynamic features of this simple system were also present in more complex versions of the sensible heat thermal energy storage systems, which could have led to the same trade-offs and a similar set of conclusions. The pioneer work for employing exergy analysis to the evaluation of phase-change, thermal energy systems, was done by Bjurström and Carlson [13]. The analysis was performed for both sensible and latent heat storage units. Uniform temperature distribution within the storage material was assumed and frictional loss in the gas side was neglected. The evaluation was made through variation of design parameter for both charging and discharging processes in order to find optimum parameters that minimize the exergy destruction. An energy analysis of the charging of the phase change unit pointed out that the melting temperature should be low. From a second law point of view the optimum melting temperature should be the geometric mean of the charging and surrounding temperature, i.e.

$$T_m = \left(T_{is} T_0\right)^{\frac{1}{2}} \tag{8}$$

An exergy balance was made, such that

$$\Psi_i = \Psi_s + \Psi_d + \Psi_e \tag{9}$$

where: Ψ_i — exergy supplied,

Ψ_s — exergy stored,

Ψ_d — exergy destroyed by irreversibilities,

Ψ_e — exergy wasted by rejecting the working fluid to surroundings.

They have found for a well-mixed sensible store or PCM store, having a heat exchanger with $NTU = 1$, the ultimate exergy fraction stored is about 30%. At the exergy maximum, about 40% of the energy supplied is stored. During discharging about 40% of the stored exergy is destroyed by transferring heat to the working fluid that cools the store. The overall second law efficiency is then about 12%. They claim that the main cause of these rather heavy losses is the heating up and cooling down of the heat storage material from a temperature that is initially different from that of the heat transfer fluids at the inlet of the heat exchanger. They have recommended that materials with low melting temperature should be superheated significantly ($T_m \approx 1.25 T_0$) and to stop charging the PCM (Phase-Change Material) with high transition temperatures

($T_m \approx 1.5 T_0$) even before the phase transition is reached. Krane [14] used the exergy techniques to optimize a thermal energy storage system with Joulean heaters. The discharging process is accomplished by the flow of air with a certain tare capacity ($\beta = \dfrac{T_I - T_{iR}}{T_{iR}}$). This work is focused on a specific class of thermal energy storage systems that has enjoyed the broad use in England and other European countries, and which is being considered for the universal use in the United States. A system of this type uses low-cost, off-peak electrical energy to heat a well-insulated mass of material with a huge potential for storing sensible thermal energy. The stored thermal energy is extracted later during a period of peak electrical power operation. The encouragement for its work is that the thermodynamic performances of thermal energy storage systems that employ Joulean heaters are very poor from the second law point of view, and must receive a close attention before this technology is used on a large scale
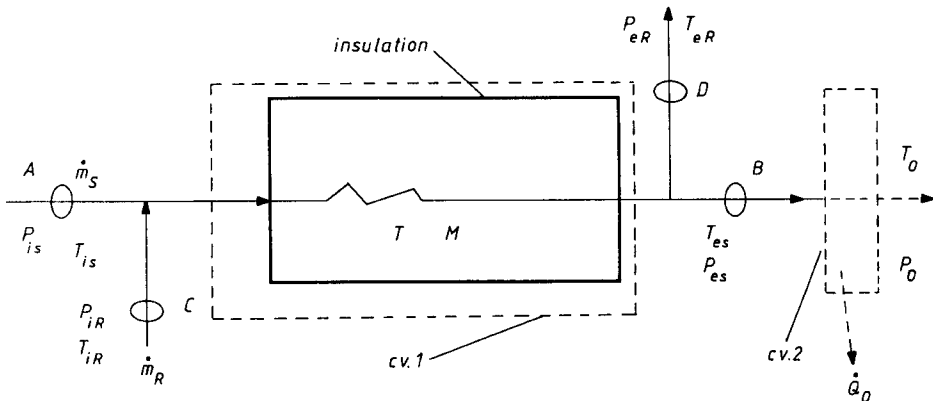

Fig. 2. Krane's model

throughout the world. The optimization code GRG2 is used. The calculations indicated that the systems examined destroyed about 60 to 80% of the supplied exergy. Krane [15] modified and extended the second law analysis of Bejan [10-12] to model an entire charging-discharging cycle of a sensible heat storage system as shown in Fig. 2. Results were obtained for 33 systems that represented ranges of independent variables that include most cases of practical interest. Air was assumed to be the working fluid for both charging and discharging processes. The calculations found that a typical optimum system destroyed about 70 to 90% of the supplied exergy. The optimization code GRG2 was used. The present authors studied the same model by developing a relatively simple iteration program. **It is substantial to distinguish that there exists a vast variety of considerations that must be accounted for in the design of an „optimum" storage system. Most of these considerations are**

**either thermodynamic or economic in nature. This work addressed only
the thermodynamic side of the design philosophy. An attempt to clarify the
significance of not disregarding the irreversibility aspect associated with
the charging-discharging cycle is sought and must not be considered to be
an attempt for global optimization.** The excellent agreement between the two
approaches, presented in Table 1, indicates that for such systems costly optimi-
zation codes may not be needed. Effects of charging time on entropy genera-
tions by each source of irreversibility in the system are shown in Fig. 3. This

Table 1

$\tau_{iR} = 0.0$;	$\tau_{is} = 1.0$;	$\dfrac{\dot{m}_R}{\dot{m}_S} = 1.0$;	$\bar{G} = 0.05$;	$\beta = 0.1$
Parameter	Results of R. Krane's system number 15 using GRG2 code		Results from the present authors' analysis	
$\theta_{s_{opt}}$	0.8634		0.86	
θ_R	2.6930		2.69	
N_Q	0.2159		0.2148	
$N_{S,T}$	0.2640		0.2646	
$N_{R,T}$	0.2142		0.2142	
N_P	0.0396		0.0401	
N	0.7337		0.7337	
NTU_{opt}	5.5330		5.6	
η_{1st}	0.5768		0.5768	

figure is constructed by using a simple program written in FORTRAN, and
a comparison with that given by Krane [15] shows an excellent agreement. It
presents the variations of different sources of irreversibility with charging time.
The deviation from the optimum charging time, which is equal to 0.863 (see
Table 1), causes an increase in the total entropy generation number (N). As
charging time is approaching zero, the largest source of irreversibility is due to
heat transfer between charging air and storage material, which is presented by
$N_{S,T}$. As expected, the contribution of this source decreases with time, since
storage material is heated up. For large storage period the contribution of heat
transfer between the existing working fluid and environment during the process
of charging, which is denoted by N_Q, becomes the dominant source of entropy
generation. The variations of the two other sources of irreversibility, N_P (due
to frictional effects) and $N_{R,T}$ (due to heat transfer between the storage mate-
rial and working fluid during the process of discharging), with charging time
are also presented in this figure. Whereas, Fig. 4 shows the variation of both

the first and second law efficiencies. It may be shown at θ_s equal to 5 that the first law efficiency approaches its limiting value of the unity. However, the second law efficiency in this case is only 11%. The same result was found by Krane, who pointed out the necessity of employing the second law technique in order to correctly optimize the thermodynamic efficiency of thermal storage systems.
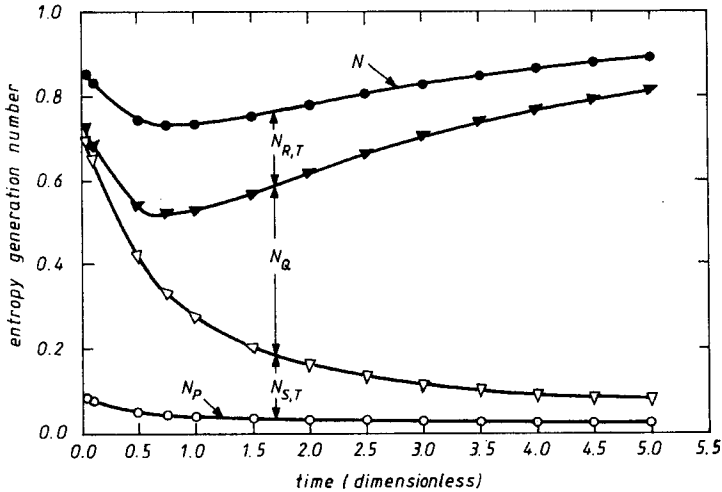

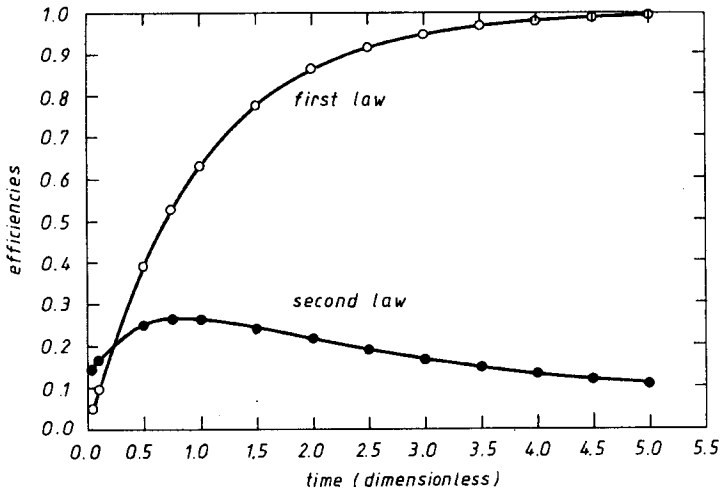Fig. 3. The influence of charging time on entropy generation number

Fig. 4. The variation of the first and the second law efficiencies with charging time

Another study of a complete charging-discharging cycle of phase-change material was done by Adebiyi and Russel [16]. They indicated at least two advantages to be gained by using a PCM in thermal energy storage systems.

Firstly, it has a better second law efficiency compared with a system that uses only sensible heat. This conclusion is disagreeing with that given by Bjurstrom and Carlson [13]. Secondly, less storage material is required compared with that for sensible storage units. Adebiyi [17] analyzed numerically the performance of bed thermal energy storage systems, based on both the first and second laws of thermodynamics utilizing phase-change materials. The temperature gradiènts inside the storage unit and along the bed length in the storage and working fluid were taken in the consideration. De Lucia and Bejan [18] documented the actual melting process for rather simple geometry. They considered a one-dimensional heating process for a rectangular element. The analysis was limited to a quasi-steady regime corresponding to small Stefan number with negligible pressure drop in the working fluid side. The first part of the paper dealt with the exergy destruction for a melting process ruled by a pure conduction, while the second part — with the exergy destruction for a melting process ruled by a natural convection. They found that for large number of heat transfer units ($NTU \approx 5$) the best second law efficiency was about 50%. They claimed, for the best second law efficiency, the melting temperature should be equal to the geometric mean of the surrounding and charging temperatures for both conduction and convection dominated melting. De Lucia and Bejan [19] studied the effects of liquid superheating during melting and irreversibility during solidification. They considered the question of whether the optimum phase--change temperature for optimum exergy utilization is always equal to the geometric mean of the charging and surrounding temperatures. They found that although the optimal melting temperature was, in general, different from the geometric mean, it was approximated fairly well by the geometric mean. The conclusions of their work can be summarized in the following points: (i) The optimum melting temperature for the best second law efficiency for a melting process, dominated by pure conduction, is less than the geometric mean that prevails in the limit of Stefan number approaching zero. The deviation between the two increases as the liquid Stefan number grows. (ii) The entropy generation number of solidification by conduction decreases monotonically as the melting temperature increases. (iii) The optimum melting temperature for complete charging-discharging cycle is less than that for the melting process alone. Taylor et al. [20,21] examined the performance of a flat-slab, sensible heat, thermal energy storage system. The physical design and operation of which have been optimized to minimize the creation of entropy by thermodynamic irreversibilities. The work included the entropy creation by transient heat conduction within the storage element. They claimed that the inclusion of a distributed storage element would result in more realistic values of the entropy generation number and these values would be appreciably lower than those predicted by lumped storage element model. In other words, lumped system models tend to overpredict the value of the entropy generation number. A second law optimization of phase-change thermal energy storage, using two or more materials, was documented by Lim et al. [22]. They investigated the

thermodynamic merits of using more than one PCM element for energy storage. The study was performed by a charging process with air as a working fluid and a discharging process by transferring heat to power machines. They conclude the following: (i) With two materials in series, an optimal melting point exists for each material; the upstream material has the higher of the two optimal melting points. (ii) The optimum performance of an arrangement where the melting point varies continuously along the source stream is equivalent to that of a scheme with unmixed fluid and a single PCM. (iii) The longitudinal and temporal variations of the temperatures and melted layer thickness in the scheme with unmixed fluid and single PCM has a negligible effect on the optimal melting temperature. M. A. Rosen [23] developed and discussed several definitions of energy and exergy efficiencies for closed systems for thermal energy storage. He concluds that exergy efficiencies are generally more meaningful and illuminating than energy efficiencies for evaluating and comparing thermal storage systems. Charach and Zemel [24] studied the latent heat storage unit in a shell-and-tube heat exchanger. A hot gas flowing through a cylindrical tube causes melting of a PCM surrounding the tube. The analysis was performed by assuming a conduction heat transfer dominating the melting process, and very low Stefan and Stanton numbers. The analysis concerned: irreversibilities due to temperature gradient in the gas side and within the melted layer, heat transfer between the working fluid and storage material, friction in the gas side. Charach [25] extended the analysis of Charach and Zemel [24] to a complete charging-discharging cycle. The assumption of quasi-steady solution and heat transfer by conduction dominating the phase-change processes was made. The analysis indicated that the second law efficiency for a complete charging-discharging cycle is less than that for a charging process alone. Saborio et al. [26] applied exergy method to predict the performance of latent heat thermal storage systems, based on a lumped system of analysis. According to them, this modeling allows a broad class of latent heat storage systems to be completely categorized with only two parameters, those are: a number of heat transfer units during charging and discharging processes, and a set of operating temperatures while still maintaining the main thermodynamic aspects associated with their operations. The model is later modified in three ways. First, the storage unit is considered to be formed of many separated containers. Each container exchanges heat only with the working fluid. Second, the system is considered to have PCMs with different melting temperatures. The third modification refers to the case when we have commercial materials. Those kinds of materials are not pure and melt, and solidify over a temperature range. An equivalent specific heat is defined as

$$C^* = \frac{\Delta H}{\Delta T} = \frac{dH}{dT} = \text{constant} \tag{10}$$

where: ΔH — change of enthalpy during phase-change period,
        ΔT — temperature range at which phase-change takes place.

For incompressible materials we may also write

$$\frac{dS}{dt} = \frac{1}{T}\frac{dH}{dt} = \frac{C^*}{T}\frac{dT}{dt} \qquad (11)$$

or

$$\Delta S = C^* \ln\left(\frac{T_{m,2}}{T_{m,1}}\right) \qquad (12)$$

where: $T_{m,1}$ — temperature at the begin of the melting process,

   $T_{m,2}$ — temperature at the end of the melting process.

The problem becomes identical to a sensible storage unit with initial temperature equal to $T_{m,1}$ and heated to a final temperature equal to $T_{m,2}$.

The results indicate that the efficiency of the basic model represents a higher boundary for the efficient operation of latent heat storage systems with negligible sensible storage capacity and a single PCM. This conclusion is in contrast with that given by Taylor et al. [20,21], which indicated a lump system of analysis over predict irreversibilities of charging-discharging cycle. Using multiple PCMs within latent heat storage systems results in improving second law efficiency. This conclusion agrees with that given by Lim et al. [22].

2. CONTRIBUTION OF THE PRESENT AUTHORS

The present authors have submitted two articles for publication. The first: *Exergy analysis for the optimum performance of phase-change thermal storage units*. The second article: *Exergy analysis for the evaluation of a thermal storage system employing PCMs with different melting temperatures*. Another article dealing with the performance of phase-change thermal energy storage unit from the second law point of view will be submitted for publishing soon. A sensible thermal energy storage unit based on lump system of analysis is studied by the present authors (see Fig. 2). The model is similar to that analyzed by Krane [15]. A relatively simple iteration program is elaborated to study the behavior of the model. The excellent agreement between the two approaches is shown in Table 1 and Figs. 3, 4. The analysis is extended to find the effect of charging temperature, mass flow rate, mass ratio, charging time, and tare capacity on the performance of a sensible storage unit. The following equations have been derived for the model.

The dimensionless entropy generations during charging process are

(i) due to thermal effects

$$\left(\overline{S_{gen,S,T}}\right)_{cv1} = \ln\left(\frac{T_F}{T_1}\right) + \int_0^{\theta_S} \ln\left(1 - B_1 e^{B2\theta}\right) d\theta \qquad (13)$$

(ii) due to frictional effect

$$\left(\overline{S_{gen,S,P}}\right)_{cv1} = \theta_S \frac{k-1}{k} \ln\left(\frac{P_{is}}{P_0}\right) \tag{14}$$

where

$$B_1 = y_S\left(1 - \frac{T_1}{T_F}\right) \tag{15}$$

$$B_2 = -y_S \tag{16}$$

$$\theta_S = \frac{1}{y_S} \ln\left(\frac{T_{is} - T_1}{T_{is} - T_F}\right) \tag{17}$$

For the control volume (cv.2)

$$\left(\overline{S_{gen,Q}}\right)_{cv2} = \left[\theta_S \tau_{is} + \left(\tau_{is} - \tau_1\right)\left(e^{-y_S\theta} - 1\right) - \int_0^{\theta_S}\left(B_3 - B_4 e^{B_2\theta}\right)d\theta\right] \tag{18}$$

where

$$B_3 = 1 + \tau_{is} \tag{19}$$

$$B_4 = y_S\left(\tau_{is} - \tau_1\right) \tag{20}$$

The temperature at duct exit can be evaluated by using the following equation

$$\tau_{es} = \tau_{is} - \left(\tau_{is} - \tau_1\right)y_S e^{-y_S\theta} \tag{21}$$

The total entropy generation during charging is found by the summation of equations (13), (14), and (18), such that

$$\left(\overline{S_{gen,S}}\right) = \left(\overline{S_{gen,S,T}}\right)_{cv1} + \left(\overline{S_{gen,S,P}}\right) + \left(\overline{S_{gen,Q}}\right)_{cv2} \tag{22}$$

The dimensionless entropy generations during the discharging process are

(i) due to thermal effects

$$\left(\overline{S_{gen,R,T}}\right) = \ln\left(\frac{T_1}{T_F}\right) + \frac{\dot{m}_R}{\dot{m}_S} \int_0^{\theta_R} \ln\left(1 + Z_1 e^{Z_2\theta}\right)d\theta \tag{23}$$

(ii) due to frictional effects

$$\left(\overline{S_{gen,R,P}}\right) = \frac{\dot{m}_R}{\dot{m}_S}\theta_R\frac{k-1}{k}\ln\left(\frac{P_{iR}}{P_0}\right) \tag{24}$$

where

$$Z_1 = y_R\left(\frac{\tau_F - \tau_{iR}}{1 + \tau_{iR}}\right) \tag{25}$$

$$Z_2 = -\frac{\dot{m}_R}{\dot{m}_S}y_R \tag{26}$$

and

$$\theta_R = \frac{1}{\left(\dfrac{\dot{m}_R}{\dot{m}_S}\right)y_R}\ln\left(\frac{\tau_F - \tau_{iR}}{\tau_1 - \tau_{iR}}\right) \tag{26}$$

The foregoing integrations will be evaluated numerically.

The temperature at duct exit during the discharging process may be found, using the following equation

$$\tau_{eR} = \tau_{iR} + \left(\tau_F - \tau_{iR}\right)y_R e^{-\frac{\dot{m}_R}{\dot{m}_S}y_R\theta} \tag{27a}$$

$$T_I = T_{iR} + \beta\, T_{iR} \tag{27b}$$

where β is called the tare capacity.

The total entropy generation during the discharging process can be found by the summation of equations (23) and (24), such that

$$\left(\overline{S_{gen,R}}\right) = \left(\overline{S_{gen,R,T}}\right) + \left(\overline{S_{gen,R,P}}\right) \tag{28}$$

The total entropy generation for the complete charging-discharging cycle is evaluated by the summation of equations (22) and (29), such that

$$\left(\overline{S_{gen}}\right) = \left(\overline{S_{gen,S}}\right) + \left(\overline{S_{gen,R}}\right) \tag{29}$$

The analysis is limited to a turbulent flow in a smooth circular pipe, then the pressure drops may be given by the following equations [11]

$$\frac{P_{is}}{P_0} = 0.5 + \left[0.25 + Pr^{2/3}\left(1 + \tau_{is}\right)\bar{G}^2\, NTU_S\right]^{0.5} \tag{30}$$

$$\frac{P_{iR}}{P_0} = 0.5 + \left[0.25 + Pr^{2/3}\left(\frac{\dot{m}_R}{\dot{m}_S}\right)^{1.8}\left(1 + \tau_{iR}\right)\bar{G}^2 NTU_S\right]^{0.5} \tag{31}$$

$$G = 342\bar{G} \tag{32}$$

The supplied exergy during a charging-discharging cycle may be written as

$$\Psi_S = T_0 MC\theta_S\left[\tau_{is} - \ln\left(1 + \tau_{is}\right) + \frac{k-1}{k}\ln\left(\frac{P_{is}}{P_0}\right)\right] \tag{33}$$

$$\Psi_R = \frac{\dot{m}_R}{\dot{m}_S}T_0 MC\theta_R\left[\tau_{iR} - \ln\left(1 + \tau_{iR}\right) + \frac{k-1}{k}\ln\left(\frac{P_{iR}}{P_0}\right)\right] \tag{34}$$

$$\Psi = \Psi_S + \Psi_R \tag{35}$$

The entropy generation number can be evaluated as

$$N = \frac{T_0 S_{gen}}{\Psi} \tag{36}$$

2.1. Results of the present analysis

The influence of charging temperature on the best second law efficiency, and on the optimum number of heat transfer units can be shown in Figs. 5 and 6.
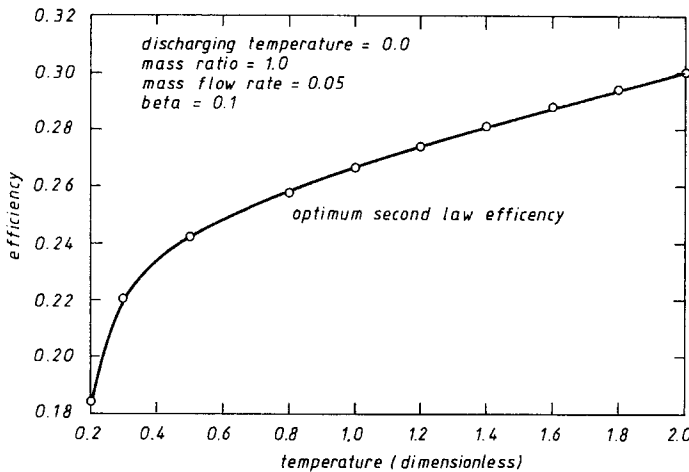
Fig. 5. The influence of charging temperature on optimum second law efficiency
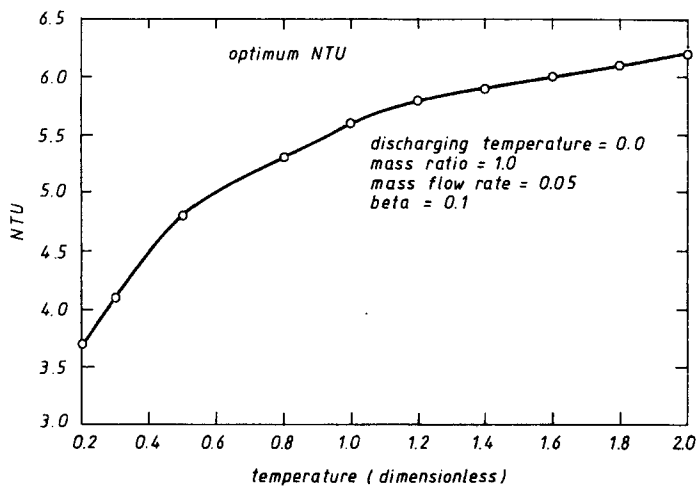
Fig. 6. The variation of the optimum *NTU* with charging temperature

To construct those figures, certain values are assigned to the operational parameters, such that:
— dimensionless discharging temperature 0.0 ($T_{iR} = T_0$),
— mass ratio 1.0,
— mass flow rate 0.05 ($G = 17.1$ kg$/(\mathrm{m}^2{\cdot}\mathrm{s})$),
— tare capacity (β) 0.1 ($T_I = 1.1 T_{iR}$).

The dimensionless charging temperature varies 0.20 ($T_{is} = 1.2 T_0$) and 2.0 ($T_{is} = 3.0 T_0$), while *NTU* varies between 1.0 and 10. The optimum efficiency jumps from a value of 0.182 at τ_{is} equal to 0.20 to a value of 0.24 at τ_{is} equal to 0.58. The improvement in the optimum second law efficiency after that is less sharp and reaches a value of 0.30 at τ_{is} equal to 2.0.

The influence of mass flow rate on optimum charging and discharging times, number of heat transfer units, and the second law efficiency are shown in Figs. 7, 8, and 9. Certain values are assigned to the operational parameters, such that:
— dimensionless charging temperature 1.0 ($T_{is} = 2.0 T_0$),
— dimensionless discharging temperature 0.0 ($T_{iR} = T_0$),
— mass ratio 1.0,
— tare capacity (β) 0.1 ($T_I = 1.10 T_{iR}$).

The dimensionless mass flow rate varies 0.01 ($G = 3.42$ kg$/(\mathrm{m}^2{\cdot}\mathrm{s})$) and 0.5 (171 kg$/(\mathrm{m}^2{\cdot}\mathrm{s})$). As expected, there are increases in the optimum charging time and the corresponding discharging time with the increase in mass flow rate. As a result of that, there is an improvement in the first law efficiency, since

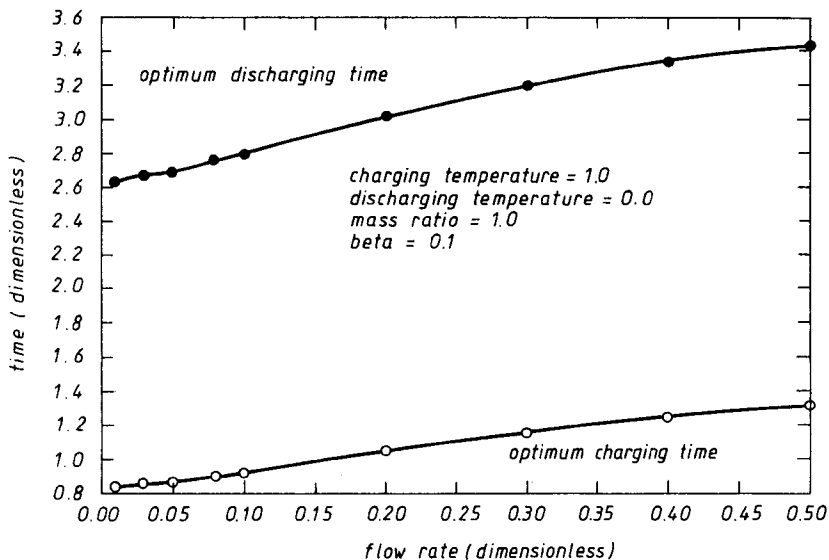$$\eta_{1st} = 1 - e^{-y_s \theta_s} \tag{37}$$

Fig. 7. The variation of the optimum charging (discharging) time with flow rate
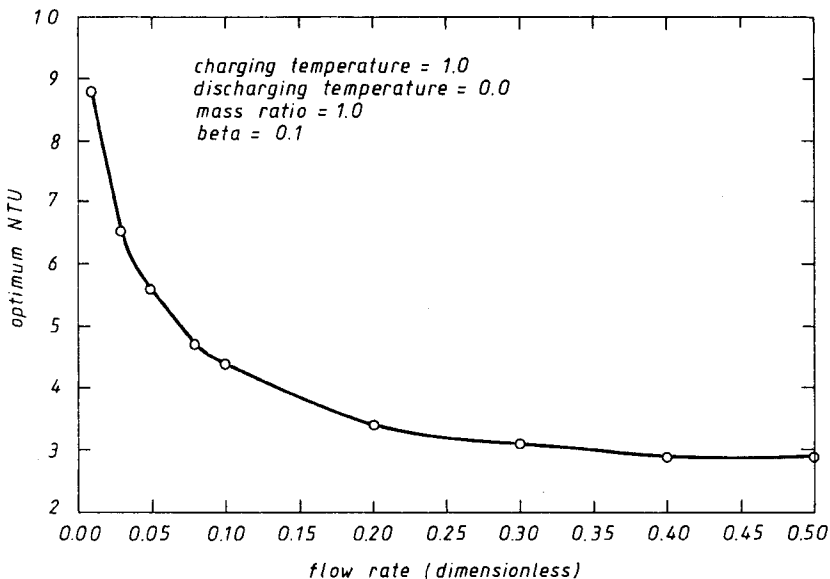

Fig. 8. The variation of the optimum *NTU* with flow rate

The results show a decrease in the second law efficiency with the increase in mass flow rate, and the losses could be very large with large flow rates as a result of the frictional effect. However, small flow rates require large number of heat transfer units (large heat exchanger). The effect of mass ratio on optimum dimensionless charging and discharging time, number of heat transfer

units, and the second law efficiency is shown in Figs. 10, 11, and 12. Certain
values are assigned to the operational parameters, such that:
− dimensionless discharging temperature 0.0 ($T_{iR} = T_0$),
− dimensionless charging temperature 1.0 ($T_{is} = 2.0T_0$),
− mass flow rate 0.05 ($G = 17.1$ kg$/(\text{m}^2 \cdot \text{s})$),
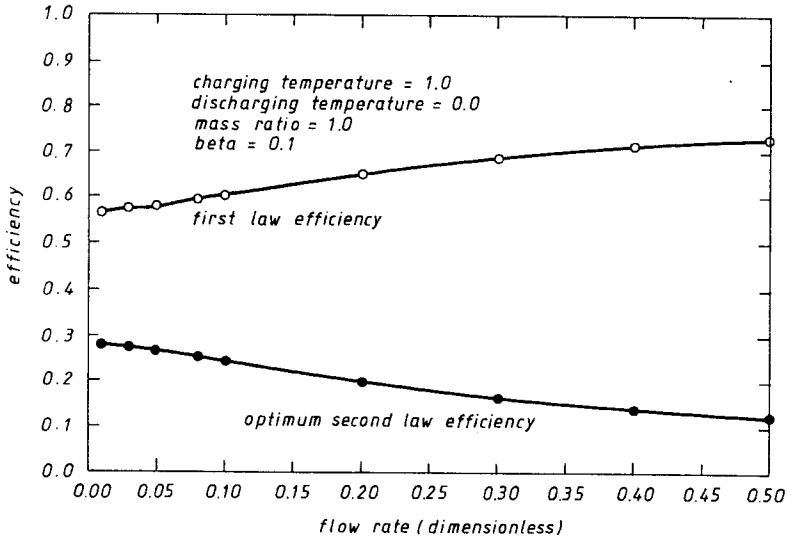− tare capacity (β) 0.1 ($T_I = 1.1 T_{iR}$).

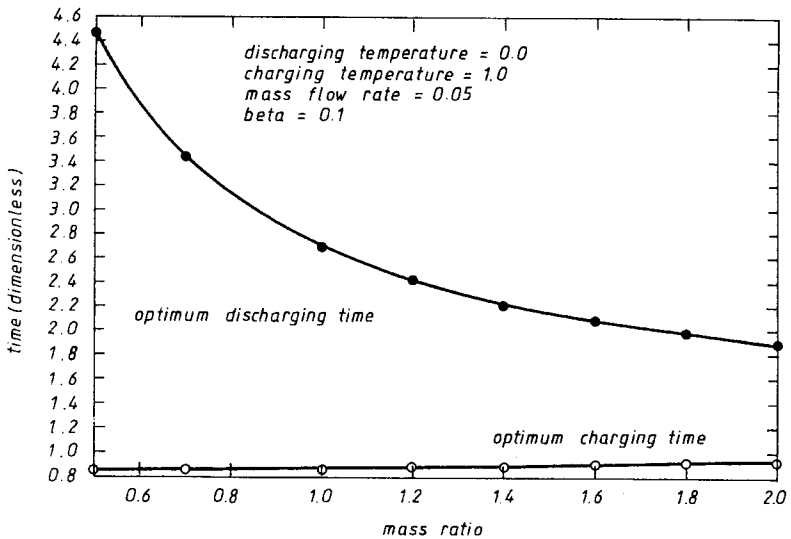Fig. 9. The variation of the optimum second law efficiency with flow rate

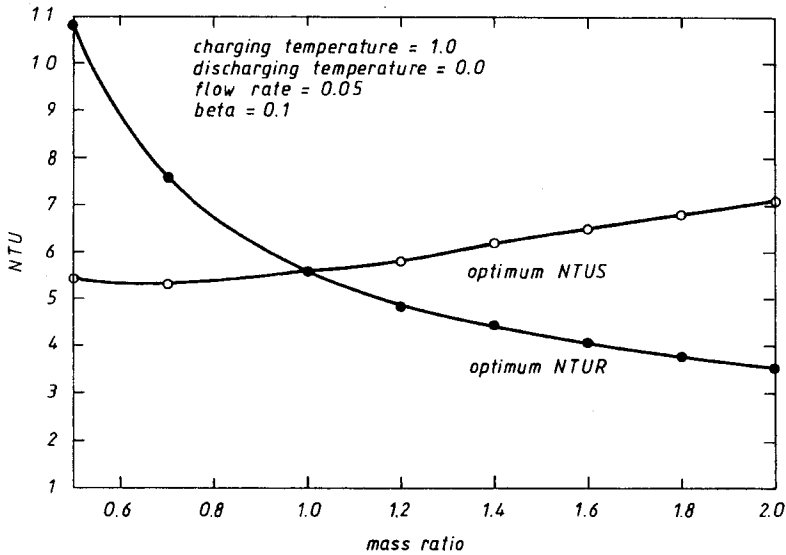Fig. 10. The influence of mass ratio on optimum charging (discharging) time

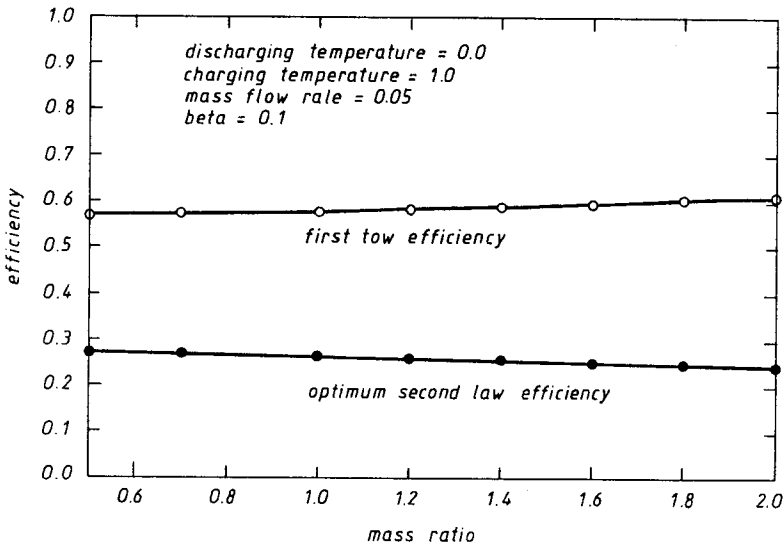Fig. 11. The variation of the optimum *NTU* with mass ratio

Fig. 12. The variation of the optimum second law efficiency with mass ratio

The results show that there is a big influence of the mass ratio on the optimum discharging time and on discharging number of heat transfer units, and that small mass ratio requires a large discharging time as well as large number of heat transfer units. Results also indicate that the influence of mass ratio on both the first and second law efficiencies is not very big. The influence of tare capacity β on the performance of the storage unit is shown in Figs. 13 and 14.

The analysis is made for:
— dimensionless discharging temperature 0.0,
— dimensionless charging temperature 1.0,
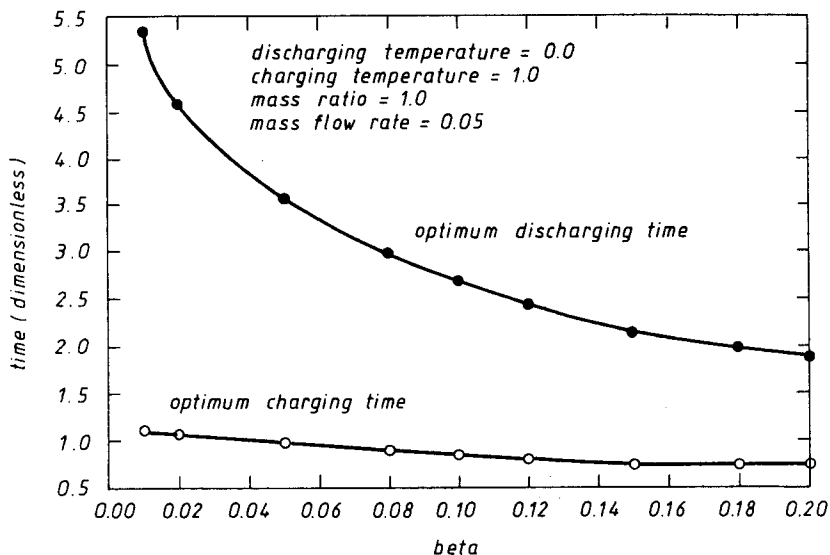— mass ratio 1.0,
— mass flow rate 0.05.

Fig. 13. The dependence of the optimum charging (discharging) time on the tare capacity
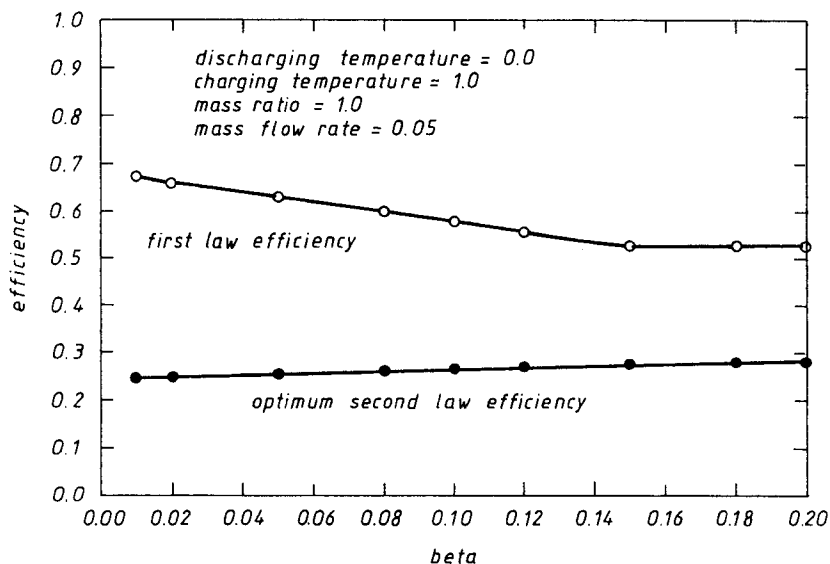

Fig. 14. The variation of the optimum second law efficiency with the tare capacity

The tare capacity varies between 0.01 and 0.2. The analysis shows that there is a big influence of beta on the optimum discharging time. Less influence is observed on optimum charging time and the first law efficiency and that — for the latter at last up to β equal to 0.15 ($T_I = 1.15\,T_0$). Beyond that the optimum charging time and the first law efficiency are invariable. There is a little gain in the second law efficiency with the increase in the tare capacity. This may be as a result of reducing entropy creation through a reduction in both charging and discharging times.

CONCLUSION

Thermal energy storage is of a great concern in practice where thermal energy generation and consumption occur at different times. The significance of the effective use of the available energy is now getting the close attention it has always deserved.

Lately, there has been an increasing attention in the second law analysis of thermal energy storage systems. The exergy method of analysis provides a true measure of effective energy use throughout its applications of both the first and second laws of thermodynamics. The present article deals with exergy as a concept and its uses in the evaluation of the performance of thermal energy storage units. The contributions of the most popular authors to the subject are presented in a great detail. The contribution of the present authors to the problem of optimizing sensible heat storage units, utilizing a relatively simple iteration technique is discussed. The results are in excellent agreement with that given by the well-known optimization code „GRG2".

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ANALIZA EGZERGETYCZNA JAKO NARZĘDZIE PRZY PROJEKTOWANIU I OCENIE PRACY UKŁADÓW MAGAZYNUJĄCYCH ENERGIĘ CIEPLNĄ

Streszczenie

W pracy przedstawiono krótko stan wiedzy i rozwój analizy egzergetycznej w zastosowaniu do oceny pracy układów magazynujących ciepło. Zostały omówione zależności opisujące różne przyczyny nieodwracalności procesów dla pełnego cyklu ładowanie-rozładowanie magazynu. Omówiono wkład pracy najbardziej znanych badaczy w tę tematykę. Porównano wyniki własne (uzyskane przy użyciu stosunkowo prostego programu z zastosowaniem techniki iteracyjnej) dotyczące układu magazynującego, wykorzystującego ciepło właściwe ze znanym optymalizacyjnym kodem numerycznym „GRG2", uzyskując bardzo dobrą zgodność.

ЭКСЕРГЕТИЧЕСКИЙ АНАЛИЗ В КАЧЕЦТВЕ ОРУДУЯ ПРИ ПРОЕКТИРОВАНИИ И ОЦЕНКЕ РАБОТЫ СИСТЕМ ТЕПЛОВЫХ АККУМУЯТОПОВ

Краткое содержание

В работе коротко представлены состояние знаний и развитие эксергитического анализа, применяемые при оценке работы тепловых аккумуяторов. Обсуждены зависимости, описыбающие разные причины необратимости процессов для зарядки и разрядки хранилища. Представлен трудовой вклад известных исследователей в эту область науки. Сравнены собственные результаты (полученные при использовании сравнительно простой программы, использующей итерационную технику), касающиеця системы хранения использыющей удельную теплоту с известным цифповым кодом „GRG2". Получено очень хорошее соответсвие.