Reviewing State-of-the-Art Exergy Analysis of Various Types of Heat Exchangers – Part 1: Principles, Double-Pipe and Shell & Tube Heat Exchangers

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ABSTRACT: *The heat exchanger, an integral component in diverse processes, finds extensive application across industrial and domestic sectors. Functioning as a mechanical apparatus, it is designed to transfer or exchange heat between different mediums, thereby enhancing energy efficiency by redirecting surplus heat from unnecessary systems to those in need. Heat exchangers have become essential equipment in various end-user applications due to their environmentally friendly nature and their ability to boost overall energy efficiency in systems. The global market for heat exchangers has undergone significant changes in recent years, with manufacturers increasingly emphasizing efficiency and performance improvements. The enhanced performance of heat exchangers, driven by technological advancements, contributes to heightened energy consumption efficiency in the systems where these devices are employed. Exergetic assessments play a crucial role in improving heat exchanger efficiency from a thermodynamic standpoint. This study provides a comprehensive review of scientific papers, examining the exergetic aspects of various heat exchanger types. The literature survey explores the impact of parameters such as entropy generation, cumulative exergy destruction, nanofluids, geometry, and two-phase fluids on heat exchanger exergetic performance. It also discusses the effectiveness of different optimization approaches on the second law's efficiency. Primarily, the study comprehensively reviews four types of heat exchangers—double-pipe, plate, cross-flow, and shell & tube—and briefly explains new types of heat exchangers. Part 1 of this study, presented in this manuscript, focuses on the fundamentals of exergy analyses in heat exchangers, with an emphasis on double-pipe and shell & tube heat exchangers. Future research directions aim to explore advanced materials with superior properties, innovative geometries for optimal performance, integration with renewable energy sources, smart technologies for adaptive control, machine learning applications for predictive modeling, and the potential of miniaturization and microscale heat exchangers. These endeavors seek to propel the field towards greater efficiency, sustainability, and adaptability across various applications.*

KEYWORDS: *Exergy, Second law of thermodynamics, Exergoeconomic, Double-pipe heat exchangers, Shell & Tube heat exchangers.*

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1021-9986/2023/11/3958-3981 24/$/7.04

INTRODUCTION

Energy saving is one of the vital goals of the global economy. Using energy systems more efficiently leads to a reduction in energy consumption of the systems [1]. Heat exchangers, as useful equipment in the processes, are widely used in vehicles, air conditioning systems, and industrial sections. So, the optimal design of heat exchangers leads to energy-saving approaches. Efficient energy usage has become an important issue due to reducing world fossil fuel resources and increasing energy costs [2, 3]. Therefore, determining heat exchangers' operating conditions and thermodynamic considerations is very important. The heat exchangers' performance is affected by various parameters like [4]:

- Increase heat duty or decrease surface area;
- The initial cost of building a heat exchanger;
- Pumping capacity or operating cost;
- Maintenance cost;
- Safety;
- Reliability;
- And compact fabrication.

Technically, heat exchangers use the heat gradient concept to exchange energy between the streams. *Bejan* represents four main types of exergy destruction in heat exchangers [5-8]:

- Exergy loss caused by finite temperature variations;
- Exergy dissipation due to fluid friction;
- Exergy degradation resulting from the materials and construction of heat exchangers [9];
- Exergy degradation arising from heat exchange with the surrounding environment of the heat exchanger.

Today, thanks to the effective insulation implemented in heat exchangers, the heat loss to the environment is virtually insignificant, resulting in negligible exergy destruction attributed to it. Nevertheless, the remaining three sources of exergy destruction can be calculated using thermodynamic and heat transfer principles. To conduct a comparative assessment of different heat exchangers, it is crucial to quantify the system's quality, indicating its level of excellence. Reversibility acts as a valuable criterion for measuring the quality of system performance. The first law of thermodynamics, also known as the energy conservation theorem, governs energy transfer during processes. As per the first law, energy cannot be created or destroyed, but its form can undergo changes [10]. Nevertheless, the first law of thermodynamics does not account for process quality, whereas the second law of thermodynamics assesses the quality of energy transfer. The second law is characterized by two statements: the Kelvin–Planck and the Clausius statements. According to the Kelvin–Planck perspective, the thermal efficiency of a heat engine cannot reach 100%. In accordance with the Clausius approach, the Coefficient of Performance (COP) for a heat pump or refrigerator is consistently less than infinity [11]. Irreversibility in a process leads to exergy destruction, which can be investigated by the second law. So the second law is useful for optimizing different and complex heating systems [12]. Heat exchangers, by their nature, exhibit irreversibility. Therefore, in the optimization and assessment of their performance, the design of heat exchangers takes into account various aspects of the second law of thermodynamics. Within engineering thermodynamics, three crucial concepts, all rooted in the second law, play a significant role: the minimization of entropy generation, exergy analysis, and exergo-economics [4]. Entropy, as a property of a system, is not conservative like energy. The entropy variation can be calculated as below [13]:

$$S_2 - S_1 = \int \frac{dQ}{T} + S_{gen} \tag{1}$$

Where in the above equation, S is entropy, kJ/kg.K, T is the temperature of boundaries, K, Q is heat transfer from the boundaries, kJ, and subscript 1, 2, and gen are related to the first, second, and entropy generation in the process. As can be understood from the above equation, the entropy generation of the adiabatic and reversible process is equal to zero. Ideally, a process with an entropy variation of zero means the highest quality of energetic interactions (reversible process). The higher entropy generation means the poorer quality of energy transfer occurs. Irreversibilities occur for various reasons, including heat transfer at finite temperature differences, friction, expansion, and mixing. In terms of thermodynamic formulation, irreversibility is obtained from the following equation [13]:

$$\dot{I} = T_0 \dot{S}_{gen} \tag{2}$$

Where in the above equation, $\dot{I}$, is irreversibilities during the process, kW/kg, T_0 is the temperature of a reference temperature and equals 298.15 K, and $\dot{S}_{gen}$ is entropy generation during the process, kW/kg.K. All thermal processes have an entropy greater than zero, and heat exchangers are no exception. The analysis of the second law aims to minimize the production of entropy within a system and attain optimal thermodynamic performance [11]. *Bejan* has done extensive research on irreversibility in heat exchangers, and his method is used as an approach to minimize irreversibilities in heat exchangers. By increasing the irreversibilities in a system, the exergy efficiency decreases. In addition to the general irreversibilities mentioned earlier, the heat exchanger and working fluid type affect the entropy generation rate and the system's efficiency [14, 15].

Exergy can be considered the highest level of ability to produce work in a process. Irreversibility indicates the amount of work that could have been done, but for some reason, it did not. Despite the energy, the amount of exergy also depends on environmental conditions. In an equilibrium state, the exergy of a system is zero. Therefore, as long as the system or environment conditions do not change, the exergy changes are zero. The exergy of a system depends on temperature, pressure, and composition changes. Also, exergy can be transferred by heat, work, and mass transfer [16, 17]. This method has a beneficial advantage, specifying the exergy destruction in each device and the flow associated with the system's irreversibility [18].

The present study introduces and reviews the research that examines parameters that affect the exergetic conditions of different heat exchangers. Also, it should be noted that exergy loss is different from exergy destruction. In this study, the term *exergy loss* is used to mean the heat loss (heat emission to the environment) present in the system, but the term *exergy destruction* refers to irreversibilities in a system.

Exergy loss and exergy destruction are fundamental concepts in thermodynamics that characterize the diminishing availability and quality of energy within a system. Exergy loss denotes the overall reduction in exergy during a process, encompassing both reversible and irreversible components. It arises due to inherent inefficiencies in real-world systems, such as friction and non-ideal heat transfer, leading to a decrease in the potential for performing useful work. On the other hand, exergy destruction specifically refers to the irreversible degradation of exergy within a system, resulting from processes that convert exergy into less useful forms, often manifesting as heat dissipation. Both exergy loss and exergy destruction are critical considerations in the analysis of energy systems, providing insights into the efficiency and irreversibilities inherent in various processes. It should be noted that based on the literature, the dead state for exergy analysis is considered as 298 K and 1 atm [19].

Our research is divided into two sections: although both segments contribute to a thorough review of exergy analysis in heat exchangers, "Part 1" concentrates on fundamental principles and specific types such as double-pipe and shell & tube, while "Part 2" broadens the scope to encompass various heat exchangers like Plate, Cross flow, and Others. Additionally, it discusses the current state and challenges in the field.

THERMODYNAMIC ANALYSIS OF HEAT EXCHANGERS

The parameters and formulas used in the thermodynamic assessments of the heat exchangers are introduced in this section. It should be noted all heat exchangers obey from mass conservation methodology [20].

Heat transfer

There are various approaches to investigate the first law of thermodynamics in heat exchangers. These methods include Logarithmic Mean Temperature Difference (LMTD), ε-NTU method, P-NTU method, Ψ-NTU method, and P_2-P_1 method. Most of the analyses performed on the heat exchangers are based on the ε-NTU method. This method is a powerful tool for analyzing different heat exchangers, comparing them with each other, and selecting the best one for the desired application. In the ε-NTU method, there is no need for the outlet streams' temperatures in the calculation; only the input stream temperatures are necessary for the analysis of the heat exchanger. The heat exchanger effectiveness is defined as follows [21, 22]:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{3}$$

Where in the above equation, ε is the effectiveness of the heat exchanger, $\dot{Q}$ and $\dot{Q}_{max}$ are the actual and maximum heat transfer in the heat exchanger, kW, respectively. The actual heat transfer for single-phase process can be obtained as below [23]:

$$\dot{Q} = \left(\dot{m}c_p\right)_h (T_{h@in} - T_{h@out}) = \left(\dot{m}c_p\right)_c (T_{c@in} - T_{c@out}) \tag{4}$$

In the given equation, the symbols represent the following: $\dot{m}$ is the mass flow rate in kg/s, c_p is the specific heat capacity in kJ/kg·K, and T is the temperature in Kelvin. Additionally, the subscripts h, c, in, and out correspond to hot streams, cold streams, inlet, and outlet, respectively. The calculation for $\dot{Q}_{max}$ is as follows [24]:

$$\dot{Q}_{max} = \left(\dot{m}c_p\right)_{min}\left(T_{h@in} - T_{c@in}\right) \tag{5}$$

In the aforementioned equation, the variables are defined as follows: $\dot{m}$ represents the mass flow rate in kg/s, c_p stands for the specific heat capacity in kJ/kg·K, and T denotes the temperature in Kelvin. Additionally, the subscripts h, c, in, and min are associated with the hot, cold, inlet, and minimum streams, respectively. The minimum stream, whether hot or cold, is determined by comparing the multiplying mass flow rate and specific heat capacity to find the minimum value. By substituting Equation (4) and Equation (5) into Equation (3), the effectiveness of the heat exchangers for the single-phase process can be adjusted as follows [25]:

$$\varepsilon = \frac{\left(\dot{m}c_p\right)_c\left(T_{c@out} - T_{c@in}\right)}{\left(\dot{m}c_p\right)_{min}\left(T_{h@in} - T_{c@in}\right)} = \frac{\left(\dot{m}c_p\right)_h\left(T_{h@in} - T_{h@out}\right)}{\left(\dot{m}c_p\right)_{min}\left(T_{h@in} - T_{c@in}\right)} \tag{6}$$

Where in the above equation, ε is the effectiveness of the heat exchanger, $\dot{m}$ is mass flow rate, kg/s, c_p is specific heat capacity, kJ/kg.K, T is the temperature, K. Also, the subscripts h, c, in, out, and min are related to hot streams, cold, inlet, outlet, and minimum streams, respectively. By using Equations (3) to (6), the amount of heat transfer can be obtained. Also, the Number of Transfer Units (NTU) that represent the dimensionless size of heat transfer is defined as follows:

$$NTU = \frac{UA}{C_{min}} \tag{7}$$

In equation (7), U is the total heat transfer coefficient, kW/m^2, A is the heat transfer area in the heat exchanger, m^2, and C_{min} is the heat capacity of the minimum stream. Depending on the type of heat exchanger, the relationship between effectiveness and NTU can be calculated using the following equation [26]:

(a) Counterflow heat exchanger:

$$\varepsilon = \frac{1 - e^{-Ntu\left(1 - \frac{C_{min}}{C_{min}}\right)}}{1 - \left(\frac{C_{min}}{C_{min}}\right)e^{-Ntu\left(1 - \frac{C_{min}}{C_{min}}\right)}} \tag{8}$$

(b) Parallel-flow heat exchanger:

$$\varepsilon = \frac{1 - e^{-Ntu\left(1 + \frac{C_{min}}{C_{min}}\right)}}{1 + \left(\frac{C_{min}}{C_{min}}\right)} \tag{9}$$

(c) Crossflow, one fluid mixed, other unmixed heat exchangers:

$(C_{max} = C_{mixed} \,.\, C_{min} = C_{unmixed})$

$$\varepsilon = \frac{C_{max}}{C_{min}}\left(1 - e^{-\left(1 - e^{-Ntu}\right)C_{min}/C_{max}}\right) \tag{10}$$

(d) Crossflow, both fluids mixed:

$$\varepsilon = \frac{Ntu}{\frac{Ntu}{1 - e^{-Ntu}} - \frac{(C_{min}/C_{max})Ntu}{1 - e^{-Ntu\left(\frac{C_{min}}{C_{min}}\right)}} - 1} \tag{11}$$

(e) For condenser and evaporator:

$$\varepsilon = 1 - e^{-Ntu} \tag{12}$$

The second law analysis

Based on [27, 28], entropy generation in heat exchangers is obtained from the following relation:

$$\dot{S}_{gen} = \dot{m}_c\left(s_{c@out} - s_{c@in}\right) + \dot{m}_h\left(s_{h@in} - s_{h@out}\right) \tag{13}$$

In the provided equation, $\dot{S}_{gen}$ represents the entropy generation during the process in kW/kg·K, $\dot{m}$ is the mass flow rate in kg/s, s denotes entropy in kJ/kg·K, and the subscripts h, c, in, and out are associated with the hot, cold, inlet, and outlet streams, respectively. Solving Equation (13) for ideal gases yields the following Equation (14) [29, 30]:

$$\dot{S}_{gen} = \left(\dot{m}c_p\right)_c ln\left(\frac{T_{c@out}}{T_{c@in}}\right) + \left(\dot{m}c_p\right)_h ln\left(\frac{T_{h@out}}{T_{h@in}}\right) - (\dot{m}R)_c\, ln\left(\frac{P_{c@out}}{P_{c@in}}\right) - (\dot{m}R)_h\, ln\left(\frac{P_{h@out}}{P_{h@in}}\right) \tag{14}$$

In the given equation, $\dot{S}_{gen}$ represents the entropy generation during the process in kW/kg·K, m ̇ is the mass flow rate in kg/s, c_p is the specific heat capacity in kJ/kg·K, T denotes the temperature in Kelvin, and R is the ideal gas constant (8.314 J/kmol·K). The subscripts h, c, in, and out are associated with the hot, cold, inlet, and outlet streams, respectively. In Equation (13), the first term on the right signifies entropy production due to heat transfer, while the second term represents entropy production due to pressure drop. Therefore, Equation (13) can be reformulated as follows [31]:

$$\dot{S}_{gen} = (\dot{S}_{gen})_{\Delta T} + (\dot{S}_{gen})_{\Delta P} \tag{15}$$

If equation (15) is solved for the incompressible fluid in the heat exchanger, Equation (16) is obtained [17, 32, 33]:

$$\dot{S}_{gen} = (\dot{m}c_p)_c \ln\left(\frac{T_{c@out}}{T_{c@in}}\right) + (\dot{m}c_p)_h \ln\left(\frac{T_{h@out}}{T_{h@in}}\right) + (\frac{\dot{m}}{\rho})_c(\Delta P)_c + (\frac{\dot{m}}{\rho})_h(\Delta P)_h \tag{16}$$

Where in the above equation, $\dot{S}_{gen}$ is entropy generation during the process, kW/kg.K, $\dot{m}$ is mass flow rate, kg/s, c_p is specific heat capacity, kJ/kg.K, T is the temperature, K, ρ is density, kg/m^3, and ΔP is a pressure drop in each stream, kPa. Subscripts h, c, in, and out represent the hot, cold, inlet, and outlet streams, respectively. The pressure drop can be obtained as follows [34]:

$$f = \frac{\Delta P}{\left(\frac{L}{d}\right)\rho U_m^2/2} \tag{17}$$

In this equation, f is the friction coefficient, ΔP is pressure drop, kPa, L is the pipe length, m, d is the pipe diameter, m, ρ is density, kg/m^3, and U_m is the average fluid velocity in the pipe, m/s. The entropy generation value for each heat exchanger can be determined as follows:

$$N_s = \frac{\dot{S}_{gen}}{C_{min}} \tag{18}$$

In the above equation, N_s is the entropy generation number, $\dot{S}_{gen}$ is the entropy generation rate, kJ/K, and C_{min} is the minimum heat capacity, kJ/K. Equation (18) is established for all heat exchangers with different flow arrangements.

The exergy concept, which was defined earlier, will be formulated in this section. The exergy of a stream is defined as the sum of chemical and physical exergy [13, 35]:

$$e = e^{ph} + e^{ch} \tag{19}$$

In the equation provided, e represents the specific exergy of a stream, measured in kJ/kg, and the superscripts ph and ch show the physical and chemical exergy. e^{ph} can be obtained as follows [36, 37]:

$$e^{ph} = (h - h_0) - T_0(s - s_0) \tag{20}$$

e^{ph} illustrates the physical specific exergy, kJ/kg, T is the temperature, K, s represents specific entropy, kJ/kg.K, and h represents the specific enthalpy, kJ/kg. Also, the subscript 0 refers to the reference conditions. Equation (21) illustrates the specific chemical exergy, kJ/kg [38, 39]:

$$e^{ch} = \sum x_i e_{0.i}^{ch} + RT_0 \sum x_i \ln x_i \tag{21}$$

Here, x_i represents the mole fraction of each substance, $e_{0.i}^{ch}$ denotes the standard-specific chemical exergy in kJ/kg, R signifies the specific air constant in kJ/kmol·K, and T_0 represents the temperature at standard conditions in Kelvin. Exergy analysis for heat exchangers can be conducted using Equation (22) [40]:

$$\dot{I} = \sum \dot{E}_{in} - \sum \dot{E}_{out} \tag{22}$$

Where in the above equation, $\dot{I}$ is exergy destruction, kW, $\dot{E}_{in}$ and $\dot{E}_{out}$ are inlet and outlet exergy to the control volume, kW, respectively.

If phase-change happens in a heat exchanger, Equation (24) can be used to calculate the entropy generation [7]:

$$\dot{S}_{gen@tp} = \frac{\dot{Q}_{tp}\Delta T}{T_{sat}T_0} + \frac{\dot{m}_{tp}\Delta P_{tp}}{T_{sat}\rho v} \tag{23}$$

$\dot{Q}_{tp}$ is heat exchange in two-phase conditions, kW, ΔT is the temperature changes, K, T_{sat} is the saturation temperature, K, T_0 is the reference ambient temperature, K, $\dot{m}_{tp}$ represents the mass flow rate of two-phase flow, kg/s, ΔP_{tp} shows the pressure drop in a two-phase fluid, ρ represents the density, kg/m$_3$, and v shows the kinematic viscosity, m^2/s.

Exergy efficiency in the context of heat exchangers is a measure of how effectively these devices utilize available energy for heat transfer processes. Unlike traditional thermal efficiency, which focuses on the quantity of energy transferred, exergy efficiency considers the quality of the energy involved, accounting for irreversibilities and losses during the heat exchange. Exergy efficiency (η_{ex}) is calculated as the ratio of the actual exergy transferred ($\dot{E}_{actual}$) to the maximum possible exergy transfer ($\dot{E}_{max}$):

$$\eta_{ex} = \frac{\dot{E}_{actual}}{\dot{E}_{max}} \quad (24)$$

The maximum possible exergy transfer represents the idealized scenario where the heat exchange process is reversible and free of irreversibilities. In real-world heat exchangers, irreversibilities such as temperature differences, pressure drops, and friction lead to exergy losses. High exergy efficiency indicates effective utilization of available energy, while lower efficiency values imply increased irreversibilities and exergy losses. Improving exergy efficiency in heat exchangers is crucial for enhancing overall system performance, reducing energy consumption, and minimizing environmental impact. It involves optimizing design parameters, improving materials, and minimizing sources of irreversibility to achieve more efficient heat transfer processes.

Cumulative exergetic consumption

Irreversibilities in heat exchangers can be attributed to factors associated with their construction and manufacturing. To evaluate exergetic losses and exergy destruction throughout the entire lifecycle, cumulative exergy analysis has been developed. This analysis considers the exergy associated with raw materials or energy carriers from their initial extraction to final utilization, encompassing materials, electricity consumption, and various equipment used in the construction of heat exchangers. In essence, cumulative exergy encompasses processes such as extraction, preparation, transportation, pretreatment, and manufacturing [41]. Hence, when examining the second law in heat exchangers, it is possible to calculate irreversibility not only stemming from temperature difference and pressure drop but also associated with the manufacturing of heat exchangers [42]. So, Eq. (23) is proposed to calculate the irreversibility from cumulative exergy [9]:

$$\dot{I}_m = \frac{ME_d}{t} \quad (25)$$

Where in the above equation, $\dot{I}_m$ is exergy destruction due to the construction, kW, M represents the heat exchanger mass, kg, E_d is cumulative exergy destruction, kJ/kg, t is the heat exchanger operating time.

Exergoeconomic

One of the topics that many researchers have recently considered is the exergoeconomic investigation of the process equipment. Exergoeconomic analysis is a powerful tool that combines exergetic and economic concepts to design cost-effective thermodynamic processes and equipment. The final cost of a heat exchanger based on the entropy generation can be obtained as follows [43]:

$$C = C_e R_f \varphi + C_s n_h T_0 (\dot{S}_{gen@H} + \dot{S}_{gen@P}) \quad (26)$$

In the above equation, C represents the cost of producing a heat exchanger, C_e is the cost of equipment, R_f is the total recovery factor, C_s is the cost of irreversibility, and n_h shows the operating time of the heat exchanger in a year. $\dot{S}_{gen@H}$ and $\dot{S}_{gen@P}$ show the entropy generation due to heat transfer and pressure drop. R_f is also obtained from the following relation [44, 45]:

$$R_f = \frac{i_e(1 + i_e)^{TL}}{(1 + i_e)^{TL} - 1} \quad (27)$$

Where in the above equation, i_e represents the effective rate of return, and TL shows the heat exchanger life cycle.

Heat exchangers types

In this section, the various studies on the exergetic conditions of various heat exchangers are reviewed. In each subsection, first, the main characteristics of the proposed heat exchangers are briefly described, and in the end, the main studies are itemized in a table. Then a summary of the most significant remarks will be presented. But before starting to review the literature, a glance at the number of papers about exergy/exergoeconomics will be very useful. Fig. 1 illustrates the trend in the number of papers focusing on the topic of exergy/exergoeconomic assessments of diverse heat exchangers from 2010 to the conclusion of August 2022. The graph demonstrates a noticeable increase in the number of articles, particularly since 2016, with a peak observed in 2019. This surge indicates a heightened interest among researchers in enhancing the performance of heat exchangers in recent years. This development is particularly promising as heat exchangers play a crucial role in various applications across

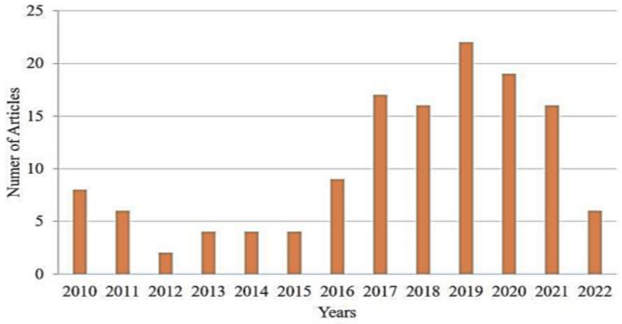

Fig. 1: The number of papers with the subject of exergy/exergoeconomic assessments of various heat exchangers in the last 12 years (August 2022)

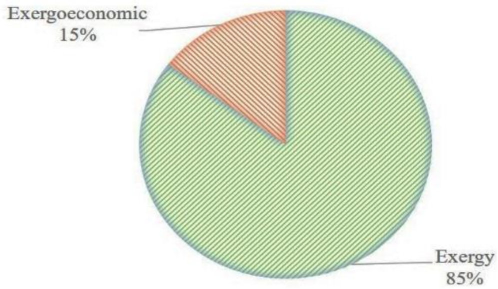

Fig. 2A: The quota of exergy- and exergoeconomic-based papers from all papers presented in this research

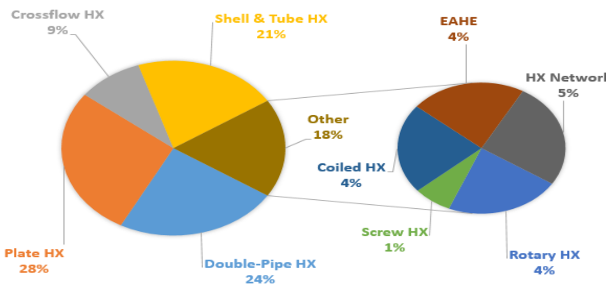

Fig. 2B: The quota of papers related to each heat exchanger from all papers presented in this research

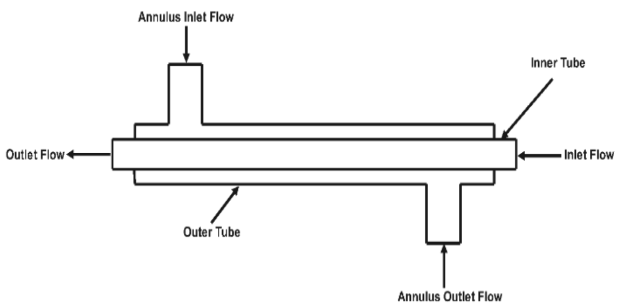

Fig. 3: Schematic of a double-pipe heat exchanger

residential and industrial sectors. Improving their performance is noteworthy, especially in the context of the ongoing global energy challenges.

Fig. 2 provides insights into the distribution of papers addressing exergy/exergoeconomic aspects in heat exchangers, as utilized in this study. According to Fig. 2A, a mere 15% of the papers examined in this research pertained to exergoeconomic assessments, while the majority focused on exergy assessments. This underscores the need for more research in the realm of exergoeconomic evaluations for diverse heat exchangers, especially in light of energy crises and climate change challenges. Fig. 2B illustrates the proportion of each heat exchanger type to the total papers cited in this research. The findings suggest that further research on the exergetic assessments of cross-flow heat exchangers is recommended for future investigations.

Double-pipe heat exchangers

Double-pipe heat exchangers are the simplest type of heat exchangers. The structure of these heat exchangers is made of two coaxial different size pipes. The streams in a double-pipe heat exchanger can be co-current and counter-current flow. The advantage of this heat exchanger over other types of heat exchangers is its simplicity in design and construction [46]. Fig. 3 shows a schematic of double-pipe heat exchangers.

Review of exergy and exergoeconomic literature

A double-pipe heat exchanger was designed using Entropy Generation Minimization (EGM). For this purpose, the heat exchanger was first designed by coding in Matlab and then optimized using the EGM approach. Its results showed that the entropy generation reaches its maximum when the heat exchanger efficiency approaches C/C+1, where C is the ratio of the heat capacity of the minimum fluid to the maximum fluid. It was also shown that there is an optimal Reynolds number for the heat exchanger, so if the Reynolds number of the hot stream is less than this optimum number, the difference between the entropy generation of parallel and opposite stream heat exchangers is so small. But if the Reynolds number is higher than the optimum, The difference in entropy generation in the two cases is not negligible [47]. The thermal coupling during laminar flow inside a double-pipe heat exchanger was studied. A parameter called the dimensionless conductive heat transfer number (KS) was defined for the solid wall of the heat exchanger. It was represented that for KS = 1000, the local entropy generation at the input of the heat exchanger has the highest value and decreases during flow. However, for KS = 1, the entropy generation is the minimum in the middle region of the heat exchanger, and it is maximum at the outputs of the heat exchanger. Also, results show that for KS = 100, the amount of entropy generation due to radial and longitudinal thermal conductivity has a uniform distribution [48]. *Sieniutycz et al.* [49] investigated the minimum entropy generated, the distribution related to the heat exchanger faces, and the contact temperature for the heat exchangers under the constant total heat load and constant heat transfer coefficient conditions. Their results represented that the optimal conditions (minimum entropy production) are obtained only for a certain distribution. Also, the structural parameters affect the optimal conditions; for example, the optimum point cannot be found for heat exchangers with a cold stream and several hot streams. Mohamed [50] divided double-pipe heat exchangers into two types, balanced and unbalanced, and defined the former as if the heat capacities of two streams are the same and the latter as if they are not equal. Also, the relationship between efficiency and entropy generation number of the double-pipe heat exchanger was described, and it was proved that this relationship is an almost symmetric parabolic function for all temperature ratios and the maximum amount of entropy generation occurs at 50% of efficiency. Also, a new function for entropy generation of unbalanced heat exchangers was introduced has a relatively good approximation to the relationships obtained by Bejan.

Using helical gear inside the inner tube is one of the ways to increase heat transfer in heat exchangers. However, this method can increase entropy generation and, consequently, the system's irreversibility. So far, various studies have been conducted on exergetic assessments of heat exchangers with swirl designs.

The impact of using helical wire inside the inner tube on heat transfer and the dimensionless entropy generation number was investigated. It was shown that the dimensionless entropy generation number increases with increasing flow rate and NTU. Another important result was that the dimensionless entropy generation number in the presence of helical wire and a laminar flow regime is much less than the turbulent flow [51].

The effect of the snail turbulator at different angles on the heat exchanger inlet was investigated. The angles of 15 to 75 degrees with an interval of 15 degrees were examined, and it was shown that the friction coefficient for an angle of 75 degrees is the highest, which increases the pressure drop and the entropy generation in the system [52]. *Durmuş* [53] conducted another study on the effect of conical turbulators on double-pipe heat exchangers, which increased the angle between 5 and 20 degrees and reached similar results. Khaled [54] investigated the improvement of heat and exergy transfers in double-pipe exchangers with conical tubes. A comprehensive parametric study was carried out, considering the impact of thermal conductivity, thermal capacity, and the ratios of the tube-to-pipe diameters. Remarkable enhancement factors were observed when the fluid with lower thermal conductivity was positioned in the annulus. In such cases, the enhancement factors for heat and exergy transfers could increase by up to 2.0 and 2.38, respectively. When thermal entry-regions were negligible, the convergent tube led to increased heat and exergy transfers compared to conventional exchangers under the same input power. Conversely, the divergent tube achieved this when entry-regions were significant. Ultimately, the utilization of conical tubes for improving heat and exergy transfers within double-pipe exchangers was recommended. The study [55] examines how gradual changes in the inner tube's geometry affect a double-pipe heat exchanger's thermal performance using CFD. Results are validated against prior research, showing strong agreement. Six inner tube designs, flat and nozzle-like, are analyzed, and their performance is compared to the base model. The model featuring a flat inner tube, particularly the variant with a flat inner tube possessing an aspect ratio of 0.3, exhibits remarkably superior performance compared to the base case at Reynolds numbers below 6000. However, at higher Reynolds numbers, the baseline model is expected to demonstrate more favorable overall conditions.

The helical effect in double-pipe heat exchangers has been thoroughly investigated by [56]. Swirl wire was placed inside the inner tube while hot air flowed in it and cold water flowed in the annulus, and it was shown that as the helical pitch decreases and the number of the helical gears increases, the heat transfer increases. It was also shown that using these swirl wires increases the pressure drop, which increases the entropy generation dimensionless number. Fig. 4 represents the proposed equipment.

The types of tube materials on the entropy production of a double-pipe heat exchanger were studied, and it was shown that maximum entropy production and effectiveness are for copper tubes, and minimum effectiveness and exergy destruction for steel tubes [57]. *Allouache* and *Chikh* [58] investigated the presence of a porous medium in double-pipe heat exchangers. A porous surface layer was assumed to be inside the annulus space and attached to the inner tube surface. Also, the boundary layer flow was considered slow and assumed the fluid conditions to be incompressible. It was shown that if the porous medium's conductive coefficient is small, the entropy generation enhances with the enhancement of the medium porosity. However, entropy generation reaches its extremum value by increasing the porosity when the porous medium's conductive heat transfer coefficient is large. *Cornelissen* and *Hirs* [59] investigated the exergetic optimization of these heat exchangers. The exergetic optimization of the system was performed based on the pressure drop and temperature differences. Also, it was shown that as the tube length and diameter decrease, the irreversibility rate of the life cycle increases. Again, *Cornelissen* and *Hirs* [60] examined the life cycle in concentric heat exchangers with three inner pipes inside and an outer pipe and reached results equivalent to their previous research [59, 60].

Nanofluids, colloidal suspensions of nanoparticles in a base fluid, offer notable advantages in energy-related equipment. Their enhanced thermal conductivity and heat transfer properties make them ideal for applications requiring efficient heat dissipation or absorption. Nanofluids exhibit increased heat capacity, reduced thermal resistance, and improved lubricating properties, contributing to enhanced overall thermal performance and reliability in systems with moving parts [61, 62]. Their tunable properties allow customization based on specific energy application requirements. Nanofluids also demonstrate thermal stability and can potentially reduce fouling and corrosion, prolonging the lifespan of energy equipment [63, 64]. Additionally, they show promise for nanogenerator applications, enabling efficient energy harvesting at the nanoscale. Ongoing research aims to optimize these advantages and address practical considerations for widespread implementation in energy systems [65].

Exergetic assessments of the crimped spiral rib within a triple-pipe heat exchanger worked with an Al_2O_3/H_2O nanofluid were numerically investigated. Its results showed that the total entropy generation and exergy destruction decrease with the volume fraction increasing, rib pitch ratio decreasing, and an increase in either crimped intensity or rib height ratio [66]. An additional triple-pipe heat exchanger was examined, featuring different rib shapes. Five distinct rib configurations, including trapezoidal, semi-circular, semi-elliptical, triangular, rectangular, and square shapes, were analyzed. The findings indicate that employing a semi-circular rib shape increases the dimensionless exergy loss rate by 5.4% to 21.6% more than other configurations [67]. Gas or air bubbles have been injected to create turbulence in the heat exchanger fluid. Bubble injection can improve the hydrodynamic performance and increase heat transfer [68]. *Heyhat et al*. In a study conducted by [69] the introduction of bubble injection in a double-pipe heat exchanger resulted in a notable increase in the heat transfer coefficient, ranging from 10.3% to 149.5%. Additionally, the dimensionless number of exergy destruction exhibited a significant rise, ranging from 2% to 226%.

The study investigated the impact of employing a hybrid nanofluid (HN) composed of Cu-GO/Therminol VP-1 (copper-graphene oxide) on both the energy and exergy performance of a double-pipe heat exchanger. Carried out under steady-state conditions using the pressure-based finite volume method, the research utilized a mixture model to simulate the Two-phase Cu-GO/Therminol VP-1 and incorporated the k-ε turbulence model. Various combinations of Reynolds number (Re) and volume fraction (φ) for graphene oxide (GO) and copper (Cu) nanoparticles were examined, spanning a range from 17000 to 41000 for Re and 0 to 3% for φ. The investigation also explored diverse geometric shapes of a novel turbulator with curvature angles (β) set at 0º, 45º, 90º, and 135º. The numerical results indicated that an increase in Re, signifying higher HN velocity, resulted in improved Nusselt number and convective heat transfer coefficient. The optimum thermal performance in the double-pipe exchanger occurred when the turbulator had β = 135º and φ = 3%. Moreover, for all scenarios, the values of the performance evaluation criteria surpassed 1, justifying the adoption of the innovative turbulator and the adjustment of its curvature angle in the heat exchanger based on PEC. Exergy efficiency exhibited enhancement with φ and Re for all β values but decreased with β [70].

Another issue that researchers have considered today is the application of nanofluids in various heat exchangers. The effect of nanofluids on small-size double-pipe heat exchangers was investigated. The graphene-silver nanocomposite was used as a nanofluid. It was shown that the exergy destruction due to temperature gradient in the presence of nanofluid is more than pure water fluid. Also, exergy destruction due to pressure drop in the presence of nanofluid is low in low Reynolds number. However, in Reynolds number above 1000 (tube side), exergy destruction due to pressure drop in the presence of nanofluid exceeds from pure water. In addition, it was found that increasing the concentration of nanofluid increases exergy destruction [71]. *Mmohammadiun et al*. [72] estimated the degree of exergy destruction of twisted tape double-pipe heat exchangers in the presence of nanofluids using a neural network approach. The results of their numerical simulation showed that with increasing Reynolds number, the exergy efficiency increases. They also showed that increasing the nanofluid concentration increases the exergy efficiency at a constant twist ratio. *Sahin et al*. [73] optimized the heat transfer of concentric double-pipe heat exchangers by considering exergy destruction, equipment cost, and heat transfer rate. They introduced a function called the thermo-economic objective function. The higher value of this function indicates a better heat exchanger performance. They also showed that a maximum value for this objective function exists for different heat exchanger efficiencies and NTUs.

Ghazizade-Ahsaee and *Ameri* [74] investigated the effect of internal heat exchangers on CO_2 direct-expansion geothermal heat pumps. Their research used a double-pipe heat exchanger as an internal heat exchanger. It was shown that when an internal heat exchanger is used, the exergy generation decreases, and therefore the exergy efficiency increases. *Jain* and *Sachdeva* [75] studied multi-objective optimization using the NSGA-II method in vapor absorption heat exchangers. In the cycle considered in this research, a double-pipe heat exchanger was used as a solution heat exchanger. Considering the economic cost, exergy destruction, and energy efficiency as variables of the NSGA method, they examined this type of heat exchanger. Their results also show that with increasing overall cost, exergy destruction in the system decreases.

Tavangar et al. [76] asserted that during the computational analysis of double-pipe heat exchangers, the first law analysis leads to some answers, which can't justify the second law. In other words, there are some thermodynamic states calculated from the first law that lead to negative entropy generation. For this purpose, they used Heat Balance Error (HBE) and virtual energy generation approaches to find the appropriate answers. Their research shows that almost 8% of results for double-pipe heat exchangers are unacceptable. Also, they showed that the number of unacceptable results is lower with smaller inlet temperature and heat exchanger efficiency.

A dynamic approach based on the Genetic algorithm was used to optimize the cascaded latent heat storage system, which used the concentric double-pipe heat exchanger to exchange heat between PCM and heat transfer fluid. It was shown that as the operating fluid mass flow rate became greater than 0.2 kg/s, the exergy efficiency decreased; however, the charged exergy efficiency remained constant. Also, for a steady-state condition, charged exergy efficiency increases linearly as inlet temperature increases. However, for an unsteady state, charged exergy efficiency decreases linearly as temperature fluctuations increase. The charged exergy efficiency is related to the exergy efficiency during charging mode [77].

Keshavarz and *Zamani* [78] explored the utilization of Phase Change Materials (PCMs) in a dual-pipe latent heat exchanger system designed for thermal energy storage. In this configuration, the outer pipe contained the salt hydrate TH29 PCM, while the inner pipe conveyed water. The operational cycle involved charging the system with hot water at 70°C and discharging it with cold water at 8°C. To enhance the efficiency of the charging and discharging processes, an adiabatic time interval was introduced, during which the pump power was deactivated, and water flow ceased. The study focused on assessing stored energy and exergy destruction during an hour of system operation, incorporating variable adiabatic time intervals. The Genetic Algorithm was employed to either maximize stored energy or minimize exergy destruction in the heat exchanger, starting with 20 initial samples and adding 50 samples per iteration. The results indicated that stored energy was dependent on the adiabatic time and reached a maximum of 25.3 kJ with an interval of 2150 s. Furthermore, exergy destruction was minimized with an adiabatic time interval of 2249 s.

Dentice d'Accadia et al. [79] studied the presence of two-phase refrigerants in double-pipe heat exchangers. First, the irreversibility of the system was calculated based

Table 1: Various references' brief conclusions and remarks about the double-pipe heat exchangers

Ref.	Type of study	Brief title	Remarks
[79]	Numerical simulation	Optimal configuration of a heat exchanger using exergoeconomics	The total cost of the heat exchanger increases with increasing saturation temperature.
[80]	Numerical simulation	exergoeconomic evaluation of heat pump with internal heat exchanger for space heating	Using IHX decreases total irreversibility and optimal cost per unit of exergy production by 2.74 % and 3.77%, respectively.
[82]	Theoretical and experimental	Second law in the concentric tube heat exchanger	By increasing the streams' mass flow rate and hot stream's inlet temperature, entropy generation increases.
[83]	Numerical simulation	Second law in the corrugated double-pipe heat exchanger	The corrugated pipe causes higher heat transfer and exergy efficiency in the heat exchanger.
[84]	Theoretical and experimental	Second law in the double-pipe heat exchanger with swirl generator	The swirl generator increases entropy generation and heat transfer in the heat exchanger.
[85]	Numerical simulation	Second law in the concentric tube heat exchanger with various tube materials	Results show that the use of copper causes more exergy destruction than aluminum.
[86]	Numerical simulation	Effect of porous environment positions on entropy generation	Placement of the porous medium on the outer wall can increase entropy generation compared to when the porous medium is inside the heat exchanger
[87]	Numerical simulation	Effect of pressure drop and longitudinal conduction on exergy destruction	With increasing NTU in the heat exchanger, irreversibility due to temperature difference and pressure drop decreases and increases, respectively.
[88]	Theoretical and experimental	Exergy analysis of air -nanofluid bubbly flow in a double-pipe heat exchanger	Using air bubble injection and nanofluid decreases and increases the exergy efficiency, respectively.
[89]	Theoretical and experimental	Second law in the double-pipe heat exchanger with nanofluid and twisted tape	Increasing the nanoparticles' volume concentration, uplifting the Reynolds number, and reducing the twist ratio can significantly raise the exergy efficiency.
[90]	Theoretical and experimental	Second law in the double-pipe heat exchanger with nanofluid and twisted tape	Exergy efficiency increases with increasing Reynolds number and nanofluid volume ratio and decreases with increasing twisted and cavity diameter ratio.
[91]	Numerical simulation	geothermal Rankine cycle utilizing a coaxial heat exchanger	By increasing the tube diameter ratio, the exergy destruction decreases.

on two variables; inner diameter and refrigerant saturation temperature; then, a cost function was introduced. The operating costs were considered due to the exergy destruction, and it was shown that the total cost of the heat exchanger increases with increasing saturation temperature. *Wang et al*. [80] performed the exergoeconomic approach on the air source transcritical CO_2 heat pump that used a Double-pipe heat exchanger as an Internal Heat Exchanger (IHX), which was used to preheat working fluid before entering the compressor. Based on their results, using IHX leads to a decrease in total irreversibility and optimal cost per unit of exergy production by 2.74 % and 3.77%, respectively. Also, they showed by using IHX, the total exergy cost rates corresponding exergoeconomic factor gradually decreased from 33.43% to 25.59%.

Saberimoghaddam and *Bahri Rasht Abadi* [81] comprised two sections. The initial part addresses often-overlooked aspects when modeling low-temperature heat exchangers, aiming to assess their genuine thermal behavior accurately. The subsequent section focuses on modeling a counter-current helically coiled tube-in-tube heat exchanger within a hydrogen liquefier, integrating insights gained from the section 'Review on exergy and exergoeconomic literature'.

The model's validation relies on literature data. The results highlight the minor positive influence of longitudinal heat conduction on hydrogen liquefaction. The introduction of heat in-leak into the cold fluid leads to elevated outlet temperatures for both the cold and hot fluids. Simulations illustrate how heat in-leakage constrains overdesign in cryogenic heat exchangers. A comparison of models, considering different assumptions, underscores the considerable variability in results stemming from these assumptions.

Summary of literature

Table 1 presents other references' brief conclusions and remarks about the double-pipe heat exchangers [79, 80, 82-91].

Based on Table 1 and the literature in the section, the following points can be itemized as the most important parameters that affect the exergetic conditions of double-pipe heat exchangers. Nevertheless, it is crucial to note the insufficient availability of exergoeconomic studies pertaining to this specific type of heat exchanger. Given its extensive range of applications, it is strongly advisable to conduct a more comprehensive exergoeconomic

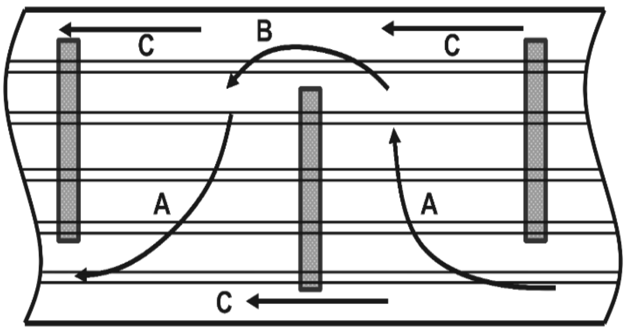

Fig. 4a: Schematic flow distribution for baffled shell-side flow without sealers (Redrawn from [95])

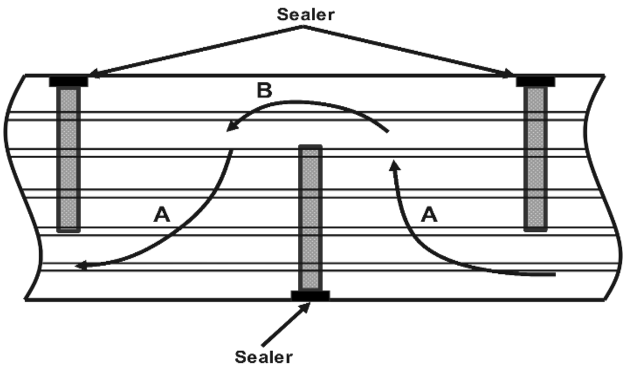

Fig. 4b: Schematic flow distribution for baffled shell-side flow with sealers (Redrawn from [95])

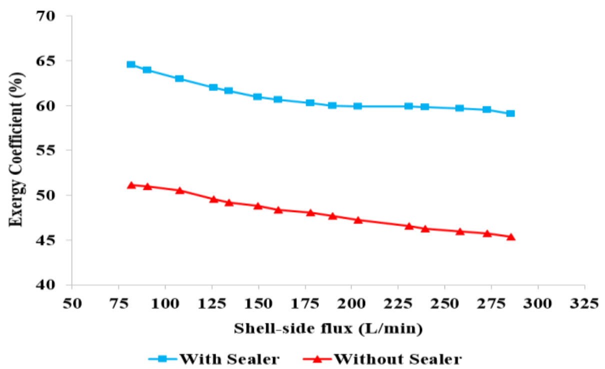

Fig. 4c: Effect of shell-side flow rate and using a sealer on heat exchanger exergy efficiency

investigations focused on the exergoeconomic assessments of double-pipe heat exchangers. The summery can be categorized as below:

- Exergy destruction increases by increasing the streams' mass flow rate and the hot stream's inlet temperature.
- Air bubble injection and swirl generator increase the exergy destruction in the heat exchanger. On the other hand, increasing tube diameter and using corrugated pipes and nanofluid will decrease the exergy destruction.
- Higher NTU number in heat exchangers leads to an increase in exergy destruction due to pressure drop and a decrease in exergy destruction due to temperature differences.
- The material used in the construction of the heat exchanger has a great impact on its exergetic condition.
- In a double-pipe heat exchanger works based on the two-phase refrigerants, the total cost of the heat exchanger increases with increasing saturation temperature.

Shell & tube heat exchanger

Shell and tube heat exchangers are the most common ones applied in industries and other major chemical processes and are also suitable for higher-pressure applications [92]. The structure of this heat exchanger consists of a shell and tubes inside it, and this structure significantly affects its irreversibility. In addition to the configuration of this heat exchanger, pressure drop and temperature change on the side of the tube-side or shell-side also affect the irreversibility[93].

Review of exergy and exergoeconomic literature

One of the industrial applications of shell and tube heat exchangers is their usage as a condenser. *Haseli et al*. [94] optimized and investigated the effect of condensate temperature on the exergetic conditions of shell and tube heat exchangers (Model TEMA type "E") used as condensers. The exergy efficiency of the shell and tube heat exchanger used as a condenser is obtained below:

$$\eta_{ex}=\frac{\dot{m}_c\, C_{p@c}\left[(T_{c2}-T_{c1})-T_0\ln\left(\frac{T_{c2}}{T_{c1}}\right)\right]}{\dot{m}_v\left\{C_{p@c}\left[(T_{v1}-T_{cond})-T_0\ln\left(\frac{T_{v1}}{T_{cond}}\right)\right]+h_{fg|T=T_{cond}}-T_{0fg|T=T_{cond}}\right\}} \quad (28)$$

In the above equation, η_{ex} is condenser exergy efficiency, $\dot{m}$ is flow rate, kg/s, T is temperature, °C, h_{fg} is saturation enthalpy, kJ/kg, C_p is specific heat capacity, kJ/kg.K, and indices v, c, $cond$, 0 represent vapor stream, cold stream, condensation temperature, and standard conditions, respectively. Also, they used Sequential Quadratic Programming (SQP) to optimize their system and obtained optimization results at two different condensate temperatures of 46 °C and 54 °C. They represented that by increasing the mass flow rate of vapor, exergy efficiency decreases while exergy destruction increases. However, the results of optimal values at the condensate temperature of 54 °C are higher than 46 °C.

They also showed that increasing the hot stream flow rate decreases the exergy efficiency of the heat exchanger. *Wang et al*. [95] investigated the effect of using sealers on heat transfer and the exergy efficiency of shell and tube heat exchangers. They used seals between the baffle plates and the shell to prevent short circuits on the shell side. Their experiments showed that the second law efficiency raised by 12.9-14.1%. Other results include a decrease in exergy efficiency due to a rise in the shell side's flow rate.

A shell and tube heat exchanger applied as a preheater was optimized using the firefly algorithm. The algorithm was performed on different exchangers, showing that exergy destruction decreases with increasing mass flow in the heat exchanger [96].

One of the main components of shell and tube heat exchangers is the arrangement and geometry of tubes. *Tahery et al*. [97] examined the effect of the number of tubes, tube arrangement, and geometry on the exergy destruction of shell and tube heat exchangers. They concluded that the No Tube in Window (NTW) design has lower exergy destruction than the original design.

The configuration of segmental baffles plays a crucial role in influencing the heat transfer and exergy destruction in shell and tube heat exchangers. To explore this aspect, *El-Said* and *Al-Sood* [98] investigated four distinct configurations for segmental baffles, namely: Conventional Single Segmental Baffle (CSSB), staggered Single Segmental Baffle (SSSB), Flower Segmental Baffle (FSB), and Hybrid Segmental Baffle (HSB).

Their results show that the HSB configuration has higher exergy, NTU, and pressure drop efficiency than the other configurations. The CSSB configuration has the worst performance in terms of thermodynamics's first and second laws. They also showed that the exergy efficiency increases with the increasing shell side volumetric flow. *Li et al*. [99] conducted an investigation on shell and tube heat exchangers with longitudinal flow. They examined three types of heat exchangers: Segmental Baffle Shell And Tube Heat Exchanger (SG-STHX), Rod Baffle Shell and Tube Heat Exchanger (RB-STHX), and Large-and-Small Hole Baffle Shell and Tube Heat Exchanger (LSHB-STHX). The study revealed that among these structures, SG-STHX exhibited the highest entropy generation, while RB-STHX had the lowest exergy destruction. The second law assessment presented in [98] indicated that the entropy generation number for SG-STHX is between 3.8% and 34.9% higher than that of LSHB-STHX. In contrast, the average entropy generation number for RB-STHX is 5.18% lower than LSHB-STHX. Additionally, RB-STHX's exergy destruction is 9.31% to 31.2% less than SG-STHX and 2.79% to 6.48% less than LSHB-STHX.

The thermodynamic performance of shell and tube heat exchanger in the presence of a wired-nails circular rod on the tube side was investigated. It was shown that using this design can increase the heat transfer coefficient and exergy efficiency by 280% and 210%, respectively [100]. *Marzouk et al*. [101] carried out an experiment involving the injection of air into the tube sides, utilizing Wired Nails-Circular Rod Inserts (WNCR). The Water Tube Side Flow Rate (TSFR) varied from 14 LPM to 18 LPM, while the water shell side flow rate remained constant at 18 LPM. Two configurations of inserts, characterized by different distances between the nails (10 cm (A) and 5 cm (B)), were employed, along with three different Air Volumetric Flow Rates (AVFR) of 1, 2, and 3 LPM. The results indicated that the second type of insert (A) had a more significant impact than the first one (B). Percentage improvements in performance parameters ranged from 31% to 131% for insert A, 43% to 177% for insert B, 33% to 143% for injection with insert A, 44% to 184% for injection with insert B, 2% to 19% for air injection only with insert A, and 1% to 23% for insert B. Furthermore, the study demonstrated that model (B) outperforms model (A) from both the first and second laws of thermodynamics' perspectives. The findings also highlighted an increase in exergy efficiency with a rising airflow rate.

Feng et al. [102] studied the cumulative exergy analysis in the shell and tube heat exchanger. They obtained the heat exchanger's cumulative exergy consumption by examining and calculating the energy consumption in producing heat exchangers (including consumables, machining, etc.). They then optimized heat transfer processes to minimize the heat cumulative exergy consumption. Their simulation results showed that the total cumulative exergy first decreases and increases with increasing LMTD in the heat exchanger. They also argued that cost optimization is related to the economic environment, while cumulative exergy optimization reflects natural resources.

One way to increase heat transfer in shell and tube heat exchangers is to use corrugated tubes. The effect of using corrugated tubes with cosine patterns on heat transfer and

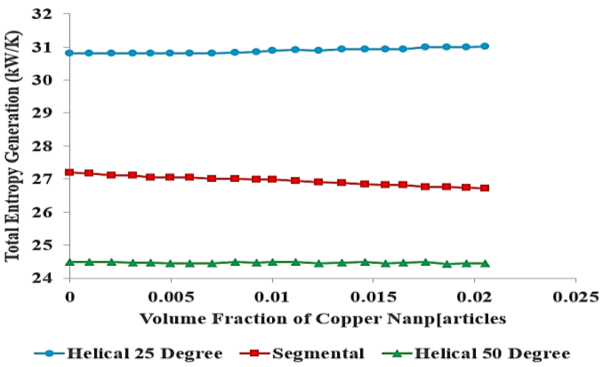

Fig. 5: Entropy generation in the shell and tube heat exchanger (Redrawn from [107])

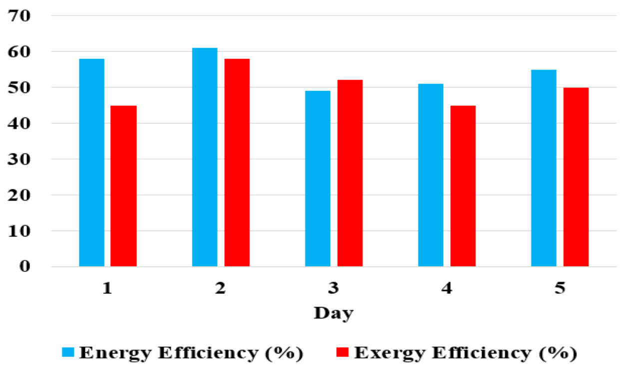

Fig. 6: The heat exchanger's average energy and exergy efficiency during the five days of evaluation (Redrawn from [110])

economic conditions of shell and tube heat exchangers and their governing equations is described by *Shirvan et al.* [103].

Various industrial devices can adopt diverse configurations, with the undulating geometry playing a crucial role. Over the past few decades, researchers and engineers have undertaken numerous studies, drawing inspiration from the heat transfer applications of sinusoidal geometry [104]. *Shirvan et al.* [105] studied multi-objective optimization by defining corrugated length, hot and cold mass flow rate, and wavelengths as the independent variables. They also defined a parameter called the exergetic sustainability index (ESI) as follows:

$$\mathrm{ESI} = \frac{1}{1-\eta_{II}} \tag{29}$$

Where in the above equation η_{II} is the exergy efficiency, and as can be seen, the value of ESI rises with the increase in exergy efficiency. Their simulation represented that enhancing the streams' mass rate and decreasing the wavy length increases the FSI value. *Deslauriers et al.* [106] studied the exergoeconomic conditions of shell and tube heat exchangers. They considered exergy and cost as independent target functions, and by optimizing both factors simultaneously, they were able to obtain some suitable solutions from the optimal Pareto front. They developed several scenarios to improve heat recovery and then optimized them using NSGAII. Their results showed that structural optimization based on exergy and cost reduces exergy destruction by 33% and increases the rate of return on costs and equipment by 22%.

Nanofluids are used as coolants due to their high heat transfer coefficient. *Leong et al*. [107] analyzed the heat transfer and entropy generation of shell and tube exchangers with three structures of segmental, 25-degree, and 50-degree helical baffle in the presence of nanoparticles. They used a combination of copper nanoparticles in ethylene glycol as nanofluids. Their results represented that increasing the volume ratio of nanoparticles has little effect on entropy generation. Also, they showed that using a 50° helical baffle has the highest exergy destruction because it produces the lowest pressure drop. Fig. 5 shows the results of entropy generation in the mentioned reference.

As can be seen in this Figure, with increasing the volume ratio of nanoparticles, the entropy generation remains almost constant, and only the segmental entropy generation decreases by about 0.5 kW/k. Also, the 25-degree helical had the best efficiency in the first law but generated the most entropy in the system.

Wang et al. [108] explored the influence of employing an Al_2O_3-CuO-water hybrid nanofluid on the energy and exergy performance of shell and tube heat exchangers. The findings indicated that the hybrid nanofluid, in combination with a turbulator and a high Reynolds number, significantly enhanced thermal performance. The results showed that a 6% increase in the volume fraction of hybrid nanoparticles, coupled with an escalation in Reynolds number from minimum to maximum, resulted in a notable 126% improvement in thermal performance in the presence of the turbulator. Ultimately, the use of nanofluids contributed to a reduction in environmental damage, energy consumption, and emissions.

Another study examined the impact of bubble injection on the exergetic performance of a helical coil shell & tube heat exchanger employed in a thermal storage system. Its results showed that using the bubble injection technique increases the Nusselt number of the heat exchanger; however, exergy destruction increases up to 2.51 times [109].

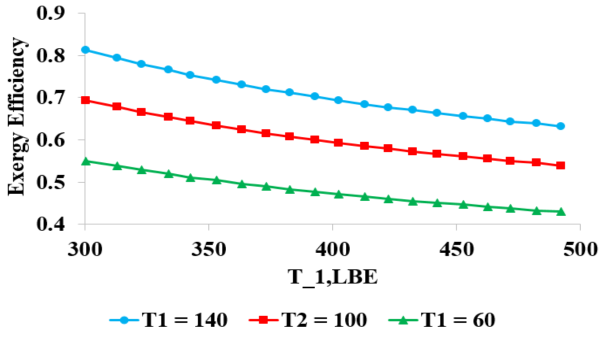

Fig. 7a: Effects of inlet conditions on exergy efficiency. (a) Inlet temperature; (Redrawn from [112])

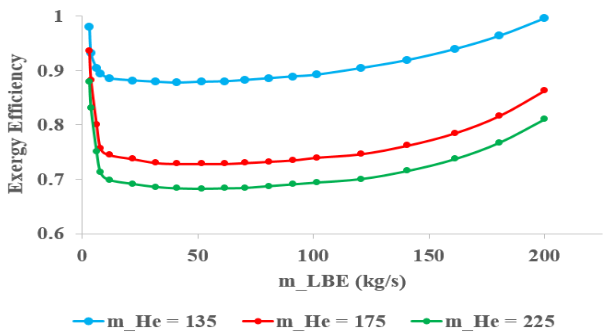

Fig. 7b: Effects of inlet conditions on exergy efficiency. (b) inlet flowrate; (Redrawn from [112])

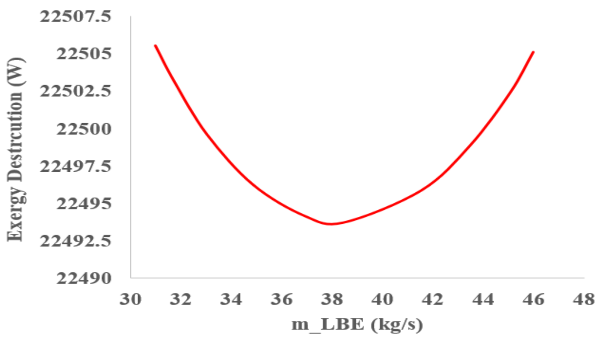

Fig. 7c: Unavoidable exergy destruction of LBE and helium gas. (c) LBE; (Redrawn from [112])

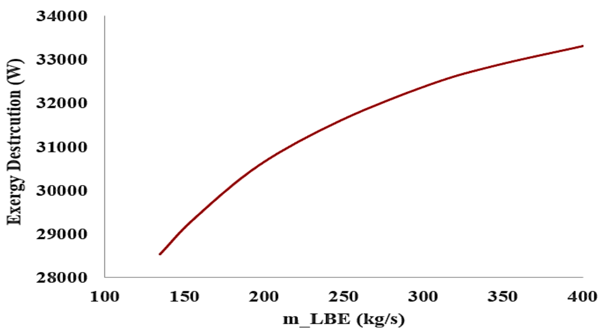

Fig. 7d: Unavoidable exergy destruction of LBE and helium gas. (d) helium gas. (Redrawn from [112])

Hosseini et al. [110] investigated an active solar distillation system integrated with a shell and tube heat exchanger and stated that unsuitable insulating would reduce the heat exchanger's second-law efficiency. They obtained the heat exchanger's maximum energy and exergy efficiencies values at 60.98% and 56.8% on the second day of the experiment. The reason for the presence of a range for the efficiencies is because exergy destruction are affected by environmental variables such as the intensity of sunlight. Fig. 6 shows the results of the energy and exergy analyses in the proposed system.

The sustainability assessment of the shell and tube exchangers for the spray dryer unit was evaluated. For this purpose, five various temperatures (0-5-10-15-20 °C) were considered as the dead state conditions. It was found that the second law efficiency was inversely proportional to the dead state temperature and reached a maximum of 65.49% at 0 °C [111]. Using low-temperature helium as a coolant, a shell and tube exchanger was investigated for cooling lead-bismuth eutectic (LBE). It was shown that the exergy efficiency decreases with increasing the inlet temperature of the LBE stream. Also, it was concluded that with the rising LBE mass flow rate, the inevitable exergy destruction decreases initially and increases after reaching the minimum point. Also, by raising the mass flow rate of the helium stream, the inevitable destruction of exergy was reduced. The simulation results are shown in Fig. 7 [112].

Jamil et al. [113] optimized the exergoeconomic condition of shell and tube exchangers using Kern, Bell-Delaware, and Wills-Johnston methods. They used exergoeconomic analysis to calculate the stream exergy-and-monetary costs and develop the cost flow diagram. Also, they used the genetic algorithm to determine the effective parameters. Their parametric investigation showed that with the optimization, the level of heat transfer had been reduced by 26.4%, capital cost by 20.5%, operational cost by 50.7%, total expenditure by 22%, and stream cost by 21%. Guo and Xu [114] used the Genetic algorithm to perform the multi-objective optimization based on the EGM and thermoeconomic situation of a shell and tube heat exchanger. Their calculation after optimization showed that the additional annual benefit from recycling is much higher than the increased annual operating cost, which means the optimization process can significantly benefit by enhancing the heat transfer between the cold and hot water. *Khan* and *El-Ghalban* [115] studied the exergoeconomic life cycle optimization

Table 2: Various references' brief conclusions and remarks about the shell & tube heat exchangers

Ref.	Type of study	Brief tit le	Remarks
[113]	Numerical simulation	Exergoeconomic optimization of a shell-and-tube heat exchanger	By optimization, the level of heat transfer had been reduced by 26.4%, capital cost by 20.5%, operational cost by 50.7%, total expenditure by 22%, and stream cost by 21%.
[114]	Numerical simulation	Optimization design of shell-and-tube heat exchanger by entropy generation minimization and genetic algorithm	After the additional annual benefit from recycling become much higher than the increased annual operating cost.
[115]	Numerical simulation	Heat Exchanger Exergoeconomic Lifecycle Cost Optimization	Operating cost has the highest impact on the exergoeconomic factor. Also, the tube arrangement has no significant effect on the total cost.
[117]	Numerical simulation	Second law in the shell & tube heat exchanger	With increasing NTU number, entropy generation first increases with a steep slope, and after reaching the maximum value, it remains constant for increasing NTU.
[118]	Numerical simulation	Exergy analysis of a shell and tube heat exchanger using DETHE software	With increasing tube pitch, decreasing tube layout, decreasing baffle cut, and decreasing shell diameter, exergy destruction increases.
[119]	Numerical simulation	Cost minimization and energy management of the shell & tube heat exchanger	NTW design could be an optimal structure for this heat exchanger type; it also has the least exergy destruction.
[120]	Theoretical and experimental	Second law in the corrugated shell & tube heat exchanger	If both the tubes and the shell are corrugated, the exergy destruction increases by 17-18%; also, the highest amount of exergy destruction is related to both the tube and shells being convexly wavy.
[121]	Numerical simulation	Exergetic Optimization in the shell & tube heat exchanger using NSGA-II	Increasing the heat exchanger's exergy efficiency increases its final cost.
[122]	Theoretical and experimental	Second law in the shell & tube heat exchanger in the presence of nanofluid (GO)	Increasing the concentration of graphene oxide leads to a reduction in exergy destruction.
[123]	Numerical simulation	Irreversibility Analysis for 1-2 TEMA G Heat Exchangers	• Irreversibility caused by the temperature difference for the configuration of the parallel stream reaches its maximum value by increasing the value of NTU, and its value becomes zero by inclining the NTU to infinity. • In contrast, this irreversibility for the configuration of the opposite stream in the initial values of NTU tends to zero and then reaches its maximum value with increasing NTU.
[124]	Numerical simulation	Usage of shell & tube heat exchanger as a heat recovery unit at solar combined cycle	The exergetic efficiency of 82% was achieved for high-pressure heat exchangers, and 92% was achieved for low-pressure heat exchangers.
[125]	Numerical simulation	Second law in the shell & tube heat exchanger in the presence of trapezoidal oblique baffles and different shaps nanofluids	The nanofluid with oblate spheroid-shaped particles shows the cold fluid's highest thermal and frictional entropy productions. The platelet-shaped particles are the best ones to be utilized in the heat exchanger, resulting in the lowest irreversibility.

based on the Genetic algorithm. They showed that for a shell and tube heat exchanger, the operating cost has the highest impact on the cost factor, and exergy destruction cost has the highest impact on the operating cost. Also, they showed that the tube arrangement has no significant effect on the total cost. *Khan* and *El-Ghalban* [116] conducted another research based on exergoeconomic life cycle optimization, but this time they used the Evolutionary algorithm for the optimization and reached similar results.

Summary of literature

Table 2 presents other references' brief conclusions and remarks about the shell & tube heat exchangers [117-125]. Based on the literature in Table 2 and section 'Review of exergy and …, the following points can be itemized:

- With increasing NTU number, exergy destruction first increases with a steep slope, and after reaching the maximum value, it remains constant for increasing NTU.
- Exergy destruction increases with increasing tube pitch, decreasing tube layout, decreasing baffle cut, and decreasing shell diameter.
- Corrugated geometry and bubble injection process lead to an increase in exergy destruction.
- By increasing LMTD in the heat exchanger, the total cumulative exergy first decreases and then tends to increase.
- Operating cost has the highest impact on the exergoeconomic factor. Also, the tube arrangement has no significant effect on the total cost.

CONCLUSIONS

One of the most practical devices among all process equipment is a heat exchanger used for energy exchanger between streams. This paper reviews and discusses the studies about heat exchangers with an exergetic focus for the first time. Applying thermodynamics's second law to the various types of heat exchangers can improve their operating conditions in the processes. Optimizing the heat exchangers is a critical issue that needs to be discussed. A comprehensive analysis of the various heat exchangers' performance based on thermodynamics's second law has been carried out by classifying their various configurations. The exergetic performance of various heat exchangers in different situations is briefly summarized below:

- Double-pipe heat exchangers:

Double-pipe heat exchangers, characterized as the simplest form in their category, feature a structure comprising two concentrically located internal pipes. The flow streams within these exchangers can either be parallel or opposite. According to existing references, entropy generation experiences an increase with higher mass flow rates of the streams and elevated inlet temperatures of the hot stream. Additionally, as the Number of Transfer Units (NTU) in the heat exchanger rises, irreversibility attributed to temperature difference decreases while that linked to pressure drop increases. An examination of the references and structural aspects of double-pipe heat exchangers reveals noteworthy insights. Corrugated pipes contribute to enhanced exergy efficiency, whereas swirl generators have the opposite effect. Furthermore, the literature indicates that a larger tube diameter results in reduced exergy destruction. Researchers in the field have explored the impact of external factors such as porous mediums, bubble injection, and nanofluids on heat transfer in these exchangers. It is noteworthy that placing a porous medium on the outer wall can elevate entropy generation compared to an arrangement where the porous medium is inside the heat exchanger. Additionally, using air bubble injection is associated with a decrease in exergy efficiency, while the use of nanofluids leads to an increase. Furthermore, elevating the volume concentration of nanoparticles has a significant positive impact on exergy efficiency. In the context of double-pipe heat exchangers operating with two-phase refrigerants, the total cost of the heat exchanger tends to rise with an increase in saturation temperature.

- Shell & tube heat exchanger:

The shell and tube heat exchanger stands as the most prevalent choice in numerous industrial plants and major chemical processes, particularly suitable for applications involving higher pressures. The configuration and design of the shell and tube play a pivotal role in influencing the irreversibility of the heat exchange process. According to the referenced literature, the entropy generation experiences an initial steep increase with a rising Number of Transfer Units (NTU), plateauing after reaching a maximum value. Irreversibility stemming from temperature differences in parallel stream configurations attains its peak with an escalating NTU, ultimately approaching zero as NTU tends towards infinity. A comprehensive review of references and structural characteristics of shell and tube heat exchangers reveals crucial insights. Exergy destruction is observed to rise proportionally with an increasing tube pitch, decreasing tube layout, decreasing baffle cut, and decreasing shell diameter. The incorporation of corrugation in both tubes and the shell corresponds to a notable 17-18% increase in exergy destruction. In contrast, the application of nanofluids in shell and tube heat exchangers is relatively rare. Notably, it has been demonstrated that augmenting the concentration of nanofluids leads to a reduction in exergy destruction. From an economic standpoint, the operating cost emerges as the most influential factor in the exergoeconomic analysis. Surprisingly, the tube arrangement is found to have an insignificant impact on the total cost, emphasizing the economic considerations in the overall performance of the shell and tube heat exchanger.

The heat exchangers examined in the other section are not mentioned in this section because there are limited references for them. To avoid the long conclusion section, it is advised to study Part of this paper.

Further research is needed to address the problems associated with heat exchangers closer to commercialization (higher efficiencies and lower cost) to make the various processes cost-effective on a large scale soon. So, more exergoeconomic-based research, especially for cross-flow heat exchangers, is highly recommended. Also, Pinch Analysis can be examined for future studies.

Nomenclature

$\dot{Q}$	Heat transfer rate (kW)
C_p	Specific heat (kJ/kg.K)
P	Pressure (kPa)
T	Temperature (K)
h	Specific enthalpy (kJ/kg)
s	Specific entropy (kJ/kg.K)
$\dot{E}$	Rate of exergy (kW)
d	Diameter (m)
e	Specific rate of exergy (kJ/kg)
$\dot{I}, \dot{E}_x$	Rate of exergy destruction (kW)
$\dot{m}$	Mass flow rate (kg/s)
x	Mole fraction (-)
R	Universal gas constant (kJ/kmol.K)
A	Area (m^2)
v	Specific volume (m^3/kg)
U	Total heat transfer coefficient (kW/m^2.°C)
U_m	Velocity (m/s)
N_s	Entropy dimensionless number (-)
L	Length (m)
$\dot{I}_m$	Exergy destruction due to the construction (kW)
M	Heat exchanger mass (kg)
E_d	Cumulative exergy destruction (kW)
t	Heat exchanger operating time (s)
n_h	Operating time in a year (s)
i_e	Effective rate of return (years)
TL	Heat exchanger life cycle (years)
Greek symbols	
η_{ex}, η_{II}	Exergy efficiency (%)
ε	Effectiveness (-)
ρ	Density (kg/m^3)
Subscripts	
in	Inlet
out	Outlet
1	State 1
2	State 2
0	Dead state
gen	generation
ph	Physical
ch	Chemical
max	Maximum
c	Cold stream
h	Hot stream
sat	Saturation state
tp	two-phase fluid
$cond$	Condension state
v	vapor
i	substance i
Abbreviations	
COP	Coefficient of performance
NTU	Number of Transfer Units
ORC	Organic rankine cycle
HBE	heat balance error
LPC	liquid-separation plate heat exchanger
CAR	corrugation amplitude ratio
PLR	path length ratio
LK	length of k path
BPHE	brazed plate heat exchangers
PFHE	plate-fin heat exchangers
LSHX	liquid to suction heat exchanger
SCHE	shell & coil heat exchanger
BPHE	brazed plate heat exchangers
CDLVG	convergent-divergent longitudinal vortex generator
EGM	Entropy Generation Minimization
SQP	Sequential Quadratic Programming
NTW	No tube at window
CSSB	conventional single segmental baffle
SSSB	staggered single segmental baffle
FSB	flower segmental baffle
LMTD	Logarithmic mean temperature difference
ESI	exergetic sustainability index
LBE	lead-bismuth eutectic
EAHE	earth-air heat exchanger
BIPVT	Building-Integrated Photovoltaic/Thermal
PEC	performance evaluation criterion
HEN	Heat exchanger network
HSB	hybrid segmental baffle

Received : Jun.23, 2023 ; Accepted : Dec.11, 2023

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