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Cryogenic Heat Exchangers for Process Cooling and Renewable Energy Storage: A Review

Dimityr Popov^{a,*}, Kostadin Fikiin^a, Borislav Stankov^a, Graciela Alvarez^b,
Mohammed Youbi-Idrissi^c, Alain Damas^c, Judith Evans^d, Tim Brown^d

^a Technical University of Sofia, 8 Kliment Ohridski Blvd., BG-1756 Sofia, Bulgaria

* Corresponding author. E-mail address: dpopov@tu-sofia.bg (D. Popov)

^b IRSTEA (ex Cemagref), 1 rue Pierre Gilles de Gennes, CS 10030, F-2761 Antony Cedex, France

^c Air Liquide R&D, 1 Chemin de la Porte des Loges, BP 126, F-78354 Jouy en Josas Cedex, France

^d School of Engineering, London South Bank University, Langford, Bristol BS40 5DU, UK

Abstract

The cryogenic industry has experienced remarkable expansion in recent years. Cryogenic technologies are commonly used for industrial processes, such as air separation and natural gas liquefaction. Another recently proposed and tested cryogenic application is Liquid Air Energy Storage (LAES). This technology allows for large-scale long-duration storage of renewable energy in the power grid. One major advantage over alternative storage techniques is the possibility of efficient integration with important industrial processes, e.g., refrigerated warehousing of food and pharmaceuticals. Heat exchangers are among the most important components determining the energy efficiency of cryogenic systems. They also constitute the necessary interface between a LAES system and the industrial process utilizing the available cooling effect. The present review aims to familiarise energy professionals and stakeholders with the latest achievements, innovations, and trends in the field of cryogenic heat exchangers, with particular emphasis on their applications to LAES systems employing renewable energy resources. Important innovations in coil-wound and plate-fin heat exchanger design and simulation methods are reviewed among others, while special attention is given to regenerators as a prospective component of cryogenic energy storage systems. This review also reveals that the geographical spread of research and development activities has recently expanded from well-established centers of excellence to rather active emerging establishments around the globe.

Keywords: energy storage, cryogenics, heat exchanger, heat transfer, modeling, optimization

Abbreviations

BAHX	Brazed Aluminum Plate-Fin Heat Exchanger
CFD	Computational Fluid Dynamics
CWHE	Coil Wound Heat Exchanger
FLNG	Floating Liquefaction of Natural Gas
GA	Genetic Algorithm
HE	Heat Exchanger
HFDB	Hybrid Friction Diffusion Bonding
HTC	Heat Transfer Coefficient
IIR	International Institute of Refrigeration
LAES	Liquid Air Energy Storage
LNG	Liquefaction of Natural Gas
MHE	Multi-stream Heat Exchanger
MR	Mixed Refrigerant
MPFHE	Multi-stream PFHE
NG	Natural Gas
PCHE	Printed Circuit Heat Exchanger
PCM	Phase-Change Material
PFHE	Plate-Fin Heat Exchanger
PSO	Particle Swarm Optimization
SFM	Sliding Flow Method

1. Introduction

Cryogenics is the science of production and application of artificial cold at very low temperatures. For a long time, the temperature range of cryogenics was not strictly defined, until the 13th *IIR International Congress of Refrigeration* (held in Washington DC in 1971) adopted a universal definition of “cryogenics” and “cryogenic” by accepting a threshold of 120 K to distinguish these notions from conventional refrigeration [1]. This is a rather reasonable temperature limit because of the normal boiling points of the most important working substances in the cryogenic

industry (including helium, hydrogen, nitrogen, oxygen, and air) all lie below 120 K. For comparison, the refrigerants commonly used in conventional refrigeration applications, such as domestic refrigerators, air conditioners, and freezers boil at temperatures above 120 K at atmospheric pressure.

Heat exchangers are among the principal components of cryogenic systems. To achieve good overall energy performance of the system, which is mostly determined by the energy requirements of the compressors, cryogenic heat exchangers should be designed to operate at relatively small temperature differences. This requires large heat transfer areas and thus bulky equipment with high initial cost [2]. High effectiveness is necessary for such heat exchangers to be cost-efficient. It has been demonstrated that a 1% increase in the effectiveness of a cryogenic heat exchanger, used in an air separation plant, decreases the compressor power consumption by 5% [3].

Cryogenic heat transfer occurs in a variety of process applications [4]. High-performance heat exchangers are essential for air separation systems which are used to produce liquid nitrogen, liquid oxygen, and liquid argon. There, recuperative heat exchangers are employed to pre-cool the incoming warm air stream by the outgoing cold gas stream, reducing the need for external refrigeration [5]. The effectiveness of those heat exchangers must be approximately 95% or higher for the system to operate economically.

The industrial applications of cryogenic technologies can be summarised in three categories: (1) process cooling; (2) separation and distillation of gas mixtures; and (3) liquefaction for transportation and storage [6]. The cryogenic industry has experienced continuous growth in the last decades, which was mostly driven by the worldwide development of liquefied natural gas (LNG) projects. The LNG technology provides an economically feasible way of transporting natural gas over long distances and currently accounts for nearly 30% of the international trade of this resource. Natural gas liquefaction requires large cryogenic heat exchangers, where natural gas is cooled, liquefied and subcooled to 111 K. Some liquefaction cycles utilize separate heat exchangers to perform the precooling or subcooling duty, while others may require only a single multi-stream heat exchanger per production train, commonly referred to as main heat exchanger. The economic feasibility of many LNG projects, considering the associated capital and operating costs, is to a large extent controlled by the performance of the main cryogenic heat exchanger. The liquefaction process presented in Fig. 1 involves such an apparatus. It is incorporated in a mixed refrigerant closed-loop cycle, using a mixture of hydrocarbons and nitrogen [7]. The refrigerant is compressed and separated into liquid and gaseous streams. The lightest fractions of the refrigerant are directed to the cold end of the main heat exchanger, cooled and sent back to the compressors, after being vaporized through the main heat exchanger. The heaviest fractions are let down and vaporized at an intermediate level also in the main heat exchanger. The advantage of the mixed refrigerant cycle is the employment of the vaporization temperature difference between the generated refrigerant streams to optimize the NG liquefaction heat exchange profile.

Liquefaction shrinks the volume of the gas almost 600 times, making it easier and safer to store and ship. Once delivered, it is converted back into gas at re-gasification facilities, usually involving large heat exchangers again.

Another industrial application of cryogenics, called Liquid Air Energy Storage (LAES), has been recently proposed and tested by Morgan *et al.* [8]. LAES systems can be used for large-scale energy storage in the power grid, especially when an industrial facility with high refrigeration load is available on-site. LAES systems can help accommodate the increasing proportion of renewables in the energy supply mix by compensating for the mismatch between the periods when the availability of renewable energy is high and the periods of high energy demand in the power grid [6]. LAES involves three core processes: (1) system charging; (2) energy storage; and (3) power recovery (also referred to as discharging), as shown in Fig.2. Charging is done by a liquefier, which uses electricity, e.g., power generated from renewable sources integrated into the grid, to draw air from the surrounding environment, purify it and then cool it to cryogenic temperatures until it liquefies. The liquid air is then stored in an insulated tank at low pressure. When power is required, liquid air is drawn from the tank and pumped to high pressure. The air is evaporated and superheated to ambient temperature. This produces a high-pressure gas, which is used to drive an air turbine, thereby providing power to the grid. LAES is prone to easy integration with important industrial technologies, such as refrigerated warehousing of food and pharmaceuticals. The LAES technology has many similarities with air separation and natural gas liquefaction. The use of high-performance cryogenic heat exchangers is also an essential prerequisite for this promising energy storage technology.

Another recent application of cryogenics involves carbon (as CO₂) capture. It is a post-combustion technology that cools the flue gas of a fossil fuel power plant to de-sublimation temperatures (173 to 138 K), separates the generated solid CO₂ from the light gaseous components, uses the cold products to cool the incoming gases in a recuperative heat exchanger, compresses the solid/liquid CO₂ to final pressures (100-200 bar), and delivers the compressed CO₂ stream ready for storage (Fig. 3).

According to Baxter *et al.* [10], this CO₂ capture technology is very promising, as the overall costs seem to be at least 30% lower than most competing processes that involve air separation units (ASUs), solvents, or similar technologies. Cryogenic technologies are also gaining ground in transport refrigeration, particularly with the Dearman engine. This is a liquid-nitrogen-powered piston engine, which is used to both move and cool the vehicle [6].

The presented brief overview of the current large-scale cryogenic technologies clearly indicates the critical and expanding role of cryogenic heat exchangers. Such heat exchangers are operated under demanding conditions, which include: large temperature range, multiple streams, multi-component mixtures and flow injection and removal. Therefore, many design aspects, which are not considered in the design of conventional heat exchangers, need to be taken into account in the design of cryogenic heat exchangers. As revealed by Pacio and Dorao,

cryogenic heat exchangers have traditionally been designed and rated using three types of models: lumped parameter, distributed parameter and stream evolution [11]. Various studies have also employed computational fluid dynamics (CFD) modelling to optimize the design of cryogenic HEs or to obtain empirical coefficients for simpler models, as discussed in the sections below. In addition, the unique modelling challenges presented by multi-stream heat exchangers (MHE), especially when multiple phases are involved, have led to the development of new modelling methods based on pinch analysis. A general classification of the models is given in Fig. 4.

Lumped parameter models are the most common and represent the basic design theory for heat exchangers. The basic differential equations, describing the energy balance of the heat exchanger, are integrated over the whole heat exchange surface area by applying the following assumptions: (1) steady-state operating conditions; (2) no heat transfer with the surroundings; (3) negligible longitudinal heat conduction; (4) constant overall heat transfer coefficient and constant heat capacities of the fluids. Therefore, lumped parameter models neglect some effects which are irrelevant to the typically required engineering precision [3].

However, the high-effectiveness requirements for cryogenic HEs necessitate consideration of the changes in fluid properties, heat exchange with the surroundings, longitudinal thermal conduction in the wall, flow maldistribution and combined effects, as reported by Pacio and Dorao [11]. The distributed parameter and stream evolution models can take these effects into account. Distributed parameter models are based on dividing the heat exchanger into elements of variable size and applying a lumped parameter model to each of them. Stream evolution models are based on steady-state one-dimensional mass, momentum, and energy balance equations for each stream. These models are usually implemented in proprietary software. Among their key features are the correlations used for determining the fluid properties and heat transfer and pressure drop characteristics.

More detailed analyses of the fluid dynamics and heat transfer processes in the flow passages of the heat exchangers can be accomplished with CFD models. They are based on discretization of the space domain into a two or three-dimensional mesh of finite volumes or elements and a coupled numerical solution of the continuity, momentum (Navier-Stokes) and energy equations applied to each volume (element). Nevertheless, CFD models require large amounts of time and effort, restricting their use primarily to the area of research and development.

MHEs are common in cryogenic applications, such as natural gas liquefaction systems, where the streams can be both multicomponent and multiphase [12]. These are very difficult to model which has motivated a number of recent studies to develop new approaches, borrowing techniques from pinch analysis. Kamath *et al.* [13] propose such a model with special emphasis on handling phase transitions. They demonstrate its capability in two case studies of cryogenic applications. Further work concerning the same model is done by Holene [14]. Other methods based on pinch analysis, which are applicable for cryogenic MHEs, are proposed by Saldida [15] and Watson and Barton [12].

The high-performance requirements and demanding operating conditions of cryogenic heat exchangers lead to high manufacturing and installation costs. In air separation and natural gas liquefaction plants, heat exchangers represent 20 to 30 % of the investment costs, as reported by Coyle *et al.* [16]. Furthermore, the design constraints lead to a very limited variety of the HE construction types and geometries. The main types of heat exchangers used for large-scale cryogenic applications are summarized in Fig. 5.

This contribution reviews the recent developments in cryogenic heat exchangers theory and design approaches for large-scale applications. Section 2 deals with the tubular spiral-wound heat exchangers which have been widely used since the very beginning of the cryogenic industry. The main features and the latest achievement in plate HE design and manufacturing are presented in section 3. Special attention is given to the plate-fin HE as the most promising type of cryogenic HE. Printed circuit HEs are also discussed as a highly compact though less common plate-type alternative with a significant potential. The design and future application of regenerators are discussed in section 4. The conclusions are summarized in section 5.

2. Coil-wound Heat Exchangers

The coil-wound heat exchanger (also known as Giauque–Hampson or spiral wound exchanger) has been used since the beginning of industrial-scale liquefaction of air, as described by Timmerhaus and Flynn [17]. The design of these heat exchangers involves winding many small-diameter tubes in a helical geometry around a central core tube, i.e., mandrel, as shown in Fig. 6. The tubes are arranged in multiple layers and enclosed in a cylindrical shell. The arrangement is flexible and in more complex configurations allows for multiple streams to be accommodated in a single heat exchanger. Regarding general flow distribution, the high-pressure stream flows inside the tubes, while the low-pressure stream flows across the tube layers on the shell side. The materials used for cryogenic applications are usually aluminum alloys, as specified by Thulukkanam [18].

A CWHE typically has a heat exchanger area per unit volume in the range of 20-300 m²/m³ [19], which lies between those of shell-and-tube and plate heat exchangers. Therefore, a single CWHE can accommodate a heat transfer surface of up to 40 000 m². Some designs feature a counter-flow arrangement with a very limited number of bypasses, which allows for very close temperature approaches – down to 1.5 K. The special mechanical design mentioned above, with the tube bundle acting as a spring, allows for quick start-up and efficient transient operation. Another advantage is the robust behavior of the heat exchanger during upset or plant trip conditions.

However, the CWHE is associated with high material costs [20] and because the small passages on both sides do not permit mechanical cleaning – it is only suitable for pure solid-free fluids. Another drawback is that this type of equipment is proprietary and the main design and production capabilities were until recently concentrated in only two original equipment manufacturers, as reported by Shukri [21]. Nevertheless, the growing needs of the LNG industry

by the end of the Twentieth Century facilitated the development of cryogenic technology outside of the traditional innovation and manufacturing centers in Europe and the USA.

In Norway, a development programme was started in 1985 to collect experimental data and develop simulation models for coil-wound heat exchangers, as reported by Fredheim *et al.* [22]. The programme involved the construction of two different test facilities – one for measurement of heat transfer rates and pressure drops in the condensing mixtures on the tube side and another one for similar measurements in the evaporating refrigerant on the shell side. The extensive test programme, lasting more than a decade, provided measurements for single-phase and two-phase flows, using different pure and mixed hydrocarbons. The experimental data was used to develop heat transfer and pressure drop correlations, which were implemented in a software tool called COIL. This tool can be used for rating calculations.

To perform both design and rating calculations, as well as static and dynamic process simulations of the main cryogenic heat exchanger in LNG plants, Neeraas *et al.* have developed more refined heat-transfer and pressure drop models for spiral-wound heat exchangers [23, 24]. The test above facility in Norway was used to obtain experimental data with further measurements for single-phase vapor flow, liquid film flow, and two-phase shear flow. Based on the measurements, using gaseous nitrogen and various hydrocarbons as test fluids, methods for estimation of the heat transfer coefficients and frictional pressure drops were tested and further improved.

Austegard *et al.* have experimentally investigated the falling film behavior around a tube array simulating typical spiral wound heat exchanger configurations using n-pentane as test fluid [24]. The measurements were compared with empirical models from other sources. The results indicated that those empirical models are not universal and not appropriate when applied to liquids with significantly different properties and closely spaced tubes.

Gupta *et al.* have published a series of papers on coil finned-tube heat exchangers. They investigated the design and optimization of coil finned-tube heat exchangers by varying the clearance between the shells and the fins [26]. They elaborated an experimental test rig, reproducing modified reverse Brayton cycle using helium as working fluid. The aim has been to acquire experimental data on the effectiveness of the heat exchanger in a real refrigeration cycle [27]. A sectional view of the Gupta's cross-counter-flow coiled finned-tube heat exchanger is shown in Fig. 7.

They aimed to obtain improved correlations for the heat transfer coefficients and pressure drops, suitable for designing finned-tube heat exchangers for cryogenic applications [28]. Later on Gupta *et al.* presented a comparison between a plate fin heat exchanger and their finned-tube heat exchanger and concluded that finned-tube heat exchangers are more suitable for application in small to medium capacity helium liquefiers, whereas plate-fin heat exchangers are a better alternative for larger capacities [29].

A three-fluid cryogenic tubular heat exchanger, involving thermal interaction between all the three fluids, was studied by Krishna *et al.* They investigated the effects of longitudinal wall conduction and ambient heat-in-leak on its performance [30, 31]. Their research provides valuable insights into the behavior of the hot fluid in this type of heat exchangers.

In several papers Zhang *et al.* present their work on research and development of cryogenic coil-wound heat exchangers [32, 33, 34]. They developed design and calculation methods, which were implemented in a software tool. However, the accuracy of these engineering tools is still questionable, and they need further development efforts.

Wang *et al.* have developed a distributed-parameter model for LNG coil-wound heat exchanger design based on the graph theory [35]. Their model is an alternative to the well-established stream evolution models [11]. In the study of Wang *et al.* the HE is partitioned into several tube bundles, each bundle is partitioned into several layers along the shell radial direction, and each layer is further divided into several multi-tube control volumes along the shell axial direction, as shown in Fig. 8.

The authors claim that their model has addressed numerous cryogenic heat transfer features. The list includes: (i) falling film evaporation, (ii) rapid changes of the thermodynamic properties of the working fluids along the tube and shell side, (iii) multiple streams with multi-component mixtures through the parallel tube layers and (iv) flow maldistribution on the shell and tube sides, whereas the alternative stream evolution models encompass only the first three factors. The model is validated against measurements obtained from existing large LNG SWHE. The results show that the predicted heat exchange capacity and an outlet temperature of LNG agree well with the experimental data. The simulation results also show that the accuracy of the distributed parameter model is comparable to the existing stream evolution model for the base-load LNG CWHE. Later on, a complete simulation tool based on the proposed methodology has been developed [36]. The cited research and especially the simulation tool appears to be a real breakthrough in the field of coil-wound heat transfer modeling, which until recently was occupied solely by software named GENIUS, developed and tested by Linde AG [37].

The same research team has decided to enhance their distributed-parameter model for LNG coil-wound heat exchanger design with capabilities for simulation of floating LNG CWHEs under rolling conditions [38]. In a tilted CWHE there are liquid-rich and gas-rich regions on the shell side due to the downward flow of the liquid refrigerant along the gravity direction, as shown in Fig. 9. During a rolling period, the boundary between the gas and liquid-rich regions dynamically changes at different tilt angles, leading to dynamically changing liquid maldistribution on the shell side and then the non-uniform heat transfer for the floating CWHEs applied in floating LNG plants.

The mathematical model of floating LNG CWHEs consists of three new sub-models, including the model for evaluating the heat transfer performance of floating CWHEs during a rolling period, the mass flow rate distribution model under rolling conditions, and the heat transfer models of the control volumes in gas and liquid-rich regions. The mass flow rate distribution of the shell-side fluid is simulated at arbitrary tilt angles. The three-dimensional mass transfer between adjacent control volumes is simulated to determine the outer limit between the gas and liquid-rich areas. The heat transfer models for the two regions are developed afterward. The simulations are used to determine the influence of the tilt angle on the performance of CWHEs. The validation of the model presented by the authors is rather artificial, as it is not based on experimental data for the performance of CWHE for FLNG. They validated their heat transfer model and distribution model separately because the accuracy of the model of floating LNG CWHEs under rolling conditions is determined by both the heat transfer model and the mass flow rate distribution model. The mean deviation of the predicted mass transfer rate distribution from the experimental data is within 16.3%, and the deviation of the predicted heat exchange capacity from the operation data of a practical application is 3.2%. It seems that the mass transfer rate distribution model needs further improvements.

A more comprehensive study on floating LNG CWHEs design is presented by Lex *et al.* [39]. Two test facilities have been built for this purpose: (1) a small scale facility set up to investigate the adiabatic fluid flow behavior; and (2) a facility of the semi-industrial size set up to investigate the thermal and hydraulic performance. The experimental outcomes have been used to develop a CFD model for design calculation of the shell-side fluid distribution.

Wu *et al.* studied the shell-side boiling heat transfer in spiral wound heat exchangers, using CFD methods [40]. For propane and ethane, thirty cases were simulated considering the variation of the most influential parameters: pressure, mass velocity, and inlet quality and heat flux. The computational models were validated through comparison with experimental data taken from [24]. The investigations indicate that the CFD method employed by the authors is applicable for the flow at low vapor quality conditions. The falling film heat transfer performance of CWHEs is primarily affected by the Reynolds number.

Ding *et al.* performed an experimental investigation of these heat transfer characteristics with propane as refrigerant [41]. The results indicate that the heat transfer coefficient initially increases and subsequently decreases with vapor quality, reaching a maximum at vapor qualities in the range of 0.8-0.9. A new correlation was developed to predict the local heat transfer coefficient of propane on the shell side of LNG CWHEs, which agrees with 98% of the experimental data within a deviation of $\pm 20\%$.

Coil-wound heat exchangers for cryogenic applications are often made of aluminum alloys and the tubes are joined to the tube sheet as shown in Fig. 10 [42]. For this specific task, conventional joining technologies such as laser beam welding or tungsten inert gas welding cannot be easily performed in fully-mechanized mode or are not economical. A joint undertaking between the Helmholtz-Zentrum Geesthacht (HZG) and LINDE Engineering has

resulted to the elaboration of a new solid-state joining process named Hybrid Friction Diffusion Bonding (HFDB) [43]. The bonding tool consists of a horizontal friction area (1) and a conical friction area (2), as shown in Fig. 11a. During the joining phase, the tool rotates at a speed of n_T and an axial force F_p is applied. During the insertion phase (Fig. 11b) only the conical friction area of the rotating tool comes into contact with the inner surface of the tube, where it exerts pressure on the contact zone between tube and tube sheet. Due to the introduced frictional heat, the tube material in the contact zone between the conical area and the tube is plasticized, which causes elastic and plastic deformations of the workpiece in the joining area. During the welding phase (Fig. 11c) the conical friction area, as well as the horizontal friction area of the tool, is in contact with the workpiece. This full-contact generates high process temperatures, which together with the pressure in the contact zone between tube and tube sheet activates diffusion processes. The rotating tool is withdrawn after a predetermined welding time.

The HFDB process has been matured to industrial readiness, and the quality of the joints has been validated by gas leak tightness tests and tensile pull-out tests. The joints cope with the requirements for industrial application.

3. Plate heat exchangers

Plate-fin heat exchangers (PFHE) are compact heat exchangers, consisting of a stack of alternate flat plates, known as “parting sheets,” and corrugated fin sheets brazed together as a block in a vacuum furnace [44, 45]. Aluminum is the preferred material for cryogenic duties, because of its relatively high thermal conductivity, strength at low temperatures, and low cost. The flow passages are formed by placing a corrugated fin sheet (with the corrugations aligned with the flow) between two parting sheets. Sidebars are placed around the outline of the flow passages to seal them. The fluid streams are connected to headers, which are welded onto the block, with distribution devices which are designed to ensure that the fluid stream flows evenly over the full width of the flow path. Fig. 12 illustrates part of a typical PFHE. The brazed aluminum plate-fin heat exchangers (BAHXs) can accommodate multiple streams in a single unit. BAHXs can be designed to operate at pressures up to 100 bar and temperatures ranging from -269 to 204 °C.

A major issue concerning the modeling and design of PFHE is the layer pattern efficiency. Extensive research is focused on the specification of the fin geometry. Fin types include plain, triangular, perforated, and serrated, among others. Løvseth *et al.* have recapitulated the outcomes from a long-term Norwegian project on the creation of expertise and tools with the goal to make possible environment-friendly, economical, and energy-efficient natural gas liquefaction [46]. The work has made use of a multilevel approach, where fundamental studies, heat exchangers modeling, and process optimization has been performed simultaneously. The project work has produced vigorous and precise Plate-Fin Heat Exchanger models that can include the description of instabilities and also include geometrical considerations.

In a series of papers, Skaugen *et al.* have presented the outcome of this project in more detail. The simulation software for PFHE and brazed plate heat exchangers have been created at

SINTEF Energy Research [47]. These tools can be made part of process simulation environments and in this way can be used as an integrated part of larger models when doing process simulations and optimization studies. Static flow fluctuations that can arise in cryogenic PFHEs are discussed. Examples of how to execute and explain a Ledinegg instability analysis are presented using the created programs. The examples illustrated that there is a tradeoff between process efficiency and static stability and that static instabilities may be avoided by moving to less efficient operating conditions or reshaping the heat exchanger geometry.

A robust and flexible modeling environment, suitable for multi-stream heat exchanger rating, was further elaborated and presented by Skaugen *et al.* [48]. In their modeling environment, the heat exchanger is partitioned into elements, each consisting of a fluid node and an array of surrounding surface. Each surface is coupled to a solid wall temperature node. Due to its flexibility, this technique can be used to simulate the performance of almost any type of heat exchanger. The authors have described this modeling environment as a lumped layer-by-layer model, but it can be classified as distributed parameter model according to the categorization given by Pacio and Dorao in [11].

Later on, Skaugen *et al.* applied this modeling technique to study the performance of plate-fin heat exchangers in LNG systems [49]. The model has been validated against the Aspen MUSE software, which is treated as an industry reference model. The discrepancy observed between the models is very small. The cited research and elaborated model appear to be a real breakthrough in PFHE simulation, which until recently was dominated by commercial design software (ASPEN HYSIS, Honeywell UniSim, etc.) [50].

A particle swarm optimization (PSO) algorithm has been developed by Peng *et al.* for sizing plate-fin heat exchangers [51]. The optimization algorithm aims to minimize the weight and annual cost of PFHEs. The objective function that gave an account to the weight of the heat exchanger provides the minimization of the size and the initial cost of the heat exchanger under the defined constraints, while the annual cost target yields the optimum pressure drops, considering the trade-off between energy consumption and heat exchanger weight. The optimization variables include the dimensions of the PFHE core, the fin geometry, the number of hot-side layers and others. The logarithmic mean temperature difference (LMTD) method is applied for the design calculations. According to the authors, compared with the results of the traditional genetic algorithm, the PSO algorithm requires shorter computational time and provides better results under the same design constraints.

Later on, Peng *et al.* [52] extended the capabilities of the model to incorporate multi-stream PFHE layers pattern design for multiple operating conditions and then obtained the heat exchange efficiency, layer pattern and fin structure. The design of the layer pattern under multiple operating conditions is optimized using a hybrid PSO. The effectiveness of these models is verified by the optimization design of a 24-stream main heat exchanger in an air separation unit.

Peng *et al.* have further supplemented their design algorithm [53], by including the effect of inlet flow mal-distribution on the thermal performance of multi-stream PFHE. With this modification, the passage arrangement has been optimized using hybrid particle swarm algorithm to raise the overall heat transfer rate. The estimated heat transfer rates were in good agreement with those obtained from Aspen calculations. The passage arrangement, i.e., layer pattern, has a significant influence on the performance of multi-stream PFHEs. One method for passage arrangement is presented by Tian *et al.* [54]. Their method has been compared with the results of previous work, indicating 2.9% reduction of the mean square error of the cumulative heat load.

Wang *et al.* presented a study on the optimal design of layer patterns in a multiple-stream PFHE [55]. The design was developed through an optimization algorithm, and its performance was evaluated by conducting an experimental study. The flow diagram of the test facility is shown in Fig. 13.

Three streams, i.e., cold air, hot air and carbon dioxide, can be used. The layer pattern experiments have been conducted for these three streams, and a twenty-two-layer heat exchanger resulted in different layer patterns, as obtained through the GA. The experimental results showed that the model performs well in optimizing the layer pattern of an MPFHE.

Wang and Li published a paper on the improved multi-objective cuckoo search algorithm, used for the design of plate-fin heat exchangers [56]. Their algorithm includes two initial objective functions, representing the irreversibilities due to heat transfer and fluid friction, which are minimized simultaneously with the aim of narrowing the scope of the design parameters. The maximization of efficiency and the minimization of pumping power and total annual cost have been considered as final objective functions. Three previous studies have been selected to evaluate the proposed algorithm's accuracy and performance. The computed examples have demonstrated the ability of the presented method to reach optimum solutions with a low number of iterations, while providing higher accuracy and lower irreversibility than alternative methods, such as single-objective design approaches.

Wang and Li also presented a review of recent studies on the theory and design of cryogenic plate-fin heat exchangers [57]. They start with a review of traditional approaches for the design of multi-stream PFHEs and then narrow the scope to layer pattern design methods. New methods for optimization of the layer pattern design, including heuristic algorithms, are discussed. The authors also present two original optimization methods developed by their research team, i.e., the "layer pattern ring model" and the "dual objective function" optimization methods.

In the design of multi-stream PFHEs surface selection and layer pattern optimization have been traditionally investigated as two independent problems. To consider these two problems simultaneously, Wang *et al.* have presented a combined method of the equipartition of entropy production and the balance of local heat load [58]. The specific entropy generation rates, resulting from heat transfer and fluid friction, are defined as a dual-objective function. The

optimal design, regarding the number of layers, pattern arrangements, and fin structures, is thus obtained through multi-objective optimization.

However, as described by Chen *et al.* [59], the optimization principle based on minimum entropy generation has been originally developed for heat engines and is more suitable for heat-to-work conversion processes. The authors recommend an optimization principle based on the entransy theory, as more suited to purely heat transfer processes and thus more appropriate for the design of heat exchangers. Accordingly, Chen *et al.* [60] define the minimum thermal resistance principle which is used to optimize the thermal performance of a heat exchanger couple.

An optimization design method for MPHEs, founded on the field synergy principle and incorporating a distributed parameters model, has been developed by Wang *et al.* [61]. The heat exchanger is partitioned into elements, i.e., sub-exchangers, along with the flow path of the mainstream, each composed of a number of passages along the height of the heat exchanger, as depicted in Fig. 14.

This method uses coupled models for calculation of the temperature and pressure fields, i.e., temperatures and pressures of both streams in each sub-exchanger, considering the variation of their physical properties and the axial heat losses. Optimum results are obtained when the temperature field along the cross-section of the heat exchanger is relatively uniform, and the temperature differences of each stream are small.

Wang *et al.* [62] also address various issues associated with the current design methodology for PFHE and elaborate an optimization framework for designing multi-stream PFHEs and the associated heat exchanger network. Their framework uses the principles of Pinch analysis for optimization of the heat exchanger network, a multi-objective algorithm for the selection of fins and layer patterns and a field synergy optimization analysis for the temperature fields in the heat exchanger. The various algorithms are compatible with each other because of the mutual restriction of design parameters and the process requirements. The temperature fields of all streams and layers are obtained with a distributed parameter model. The optimization scheme has been tested on two multi-stream heat exchangers used for cryogenic air separation, and the results have been paralleled with those of traditional design methods. The outcomes of the comparison demonstrate improvement in both the thermal and the hydraulic performance of the heat exchangers by using the proposed optimization framework.

One issue pertinent to single-phase cryogenic heat exchangers is flow maldistribution which leads to deterioration of their performance [11]. It is highly dependent on the header configuration. Picard *et al.* [63] have used CFD modelling to study this effect and observed that the deterioration of HE performance is even more pronounced when only one of the flows is disturbed. Furthermore, the authors demonstrate that the HE performance can be improved with appropriate design modifications. In another study based on CFD modelling, Yang *et al.* [64] have proposed an improved header configuration with a quasi-S shape, designed to reduce

flow maldistribution and thus enhance the thermal performance. A schematic diagram of the conventional header (denoted as configuration (a) and the improved quasi-S type header (b) are shown in [Fig. 15](#).

During the transient states plate-fin structures, as the key component of PFHE, may cause severe stress concentration and thermal shock if the operation is not appropriate. In two papers Ma *et al.* have presented some recommendations on how this situation can be controlled. Firstly, they have developed a mathematical model, based on the finite element method and the thermal elastic theory, to simulate the material stress occurring in the plate-fin structures of LNG heat exchangers during the cool-down process [65]. A cool-down experiment, in which the liquid nitrogen has been used as the cryogenic medium, was implemented to validate the simulation results. The simulation results show a rapid change in the equivalent stress, which reaches a peak value in the brazed joint, as a consequence of which a crack is initiated in this area during the cool-down process. In their next study, the authors have investigated the stress characteristics of plate-fin structures during the heat-up process, using the same computational model [66]. The results demonstrate the complexity of the stresses occurring in plate-fin structures, which again tend to reach peak values in the brazed joint.

Ma *et al.* have also studied the influence of operation parameters on the stress occurring in the plate-fin structures of LNG heat exchangers, where natural gas is cooled by a mixed refrigerant [67]. The results witnessed that material fatigue and cracks would develop initially in the area of the brazed joint neighboring to both the plate side and the fin side. The peak value of the equivalent stress grows steadily when the temperature difference between the fluid flows increases and also when the pressure of the mixed refrigerant or the natural gas is increased (the impact of the natural gas pressure is more pronounced). The equivalent stress is slightly impacted by the temperature gradient along the direction of the natural gas flow.

The compactness of plate heat exchangers is a major advantage as they can incorporate larger heat transfer surfaces per unit volume than the bulkier coil-wound heat exchangers. Although the PFHE is the most common plate heat exchanger, printed circuit heat exchangers (PCHE) are even more compact due to their manufacturing technology. The heat exchanger area per unit volume of a PCHE exceeds $2500 \text{ m}^2/\text{m}^3$ [68], as compared with $20\text{-}300 \text{ m}^2/\text{m}^3$ in a CWHE. The metal plates of a PCHE (typically 1.6 or 2 mm thick) are subjected to photochemical etching to form fine grooves on the surface, which serve as flow passages. The latter are typically semi-circular channels with a hydraulic diameter in the range of 0.5 to 2 mm. The plates are then joined together by diffusion bonding to form a very strong compact metal block. The complete solid-state HE core comprises a number of such blocks welded together. This manufacturing method allows for elaborate design featuring different flow geometries and high integrity [68, 69]. Apart from PCHEs, there are two other types of diffusion bonded heat exchangers – formed plate heat exchangers (which are manufactured from corrugated fins) and hybrid HEs (which are a kind of amalgamation of printed circuit and formed plate HEs) [70]. The three types of diffusion bonded heat exchangers are shown in [Fig. 16](#).

The robust diffusion-bonded non-gasket construction of PCHEs ensures safe operation at high pressures (up to 600 bar) and wide temperature ranges (from -200 to 900 °C) [71], while the risk of leakage is much lower than in any other HE type [68]. Moreover, the thermal effectiveness of a PCHE can exceed 98% [68]. The PCHE drawbacks include high initial cost and the possibility of flow blockage because of the very small channel diameters. To avoid contamination and blockages, the heat transfer fluids should be well-purified, while the filters should be regularly cleaned [71]. However, fouling does not constitute a substantial problem for gas-gas applications [68]. A significant limitation of PCHEs, operating at low or moderate pressures, is the large pressure drop due to the small channel cross sections with high hydraulic resistance. Nevertheless, this is not a constraint for high-pressure applications [68].

The first PCHE has originally been developed in 1980 at the University of Sydney, while a major PCHE manufacturer and pioneer is the UK-based Heatric company, which is highly specialized in the area since 1985 [70]. PCHEs are also currently produced by other manufacturers (such as Alfa Laval) and have been widely used in offshore applications where the small size and weight are a major advantage [69]. Cryogenic applications of PCHEs include natural gas liquefaction systems [69] and a unique LAES pilot plant in the UK [72].

The number of publications, presenting research on PCHE performance at cryogenic temperatures, is still scarce. For example, Baek *et al.* [73] present an experimental investigation of the thermal and hydraulic performance of PCHEs in the cryogenic temperature range, with an emphasis on the effect of axial conduction. The authors tested two PCHEs with different number of plates and observed that the thermal performance is significantly affected by axial conduction at low Reynolds numbers, while this effect diminishes at high Reynolds numbers. However, due to the lower thermal conductivity at cryogenic temperatures, the effect of axial conduction is not as large as the analogous effect at higher temperatures [73]. Another study by Biswal, based on ANSYS Fluent CFD modelling, reports on similar findings [71]. Baek *et al.* [73] also modified the PCHE design by reducing the cross-sectional area with wire-cutting. This modification has further been tested to prove that the PCHE performance is thus enhanced [73].

While PCHEs are not as widespread in the cryogenic industry as CWHEs and PFHEs, their potential to further penetrate the market is rather substantial, especially if the HE size, weight and endurance are major concerns.

4. Regenerative heat exchangers

Barron and Nellis [4] have specified regenerators as a class of cryogenic heat exchangers, in which heat exchange between the two fluids occurs through intermediate heat exchange with a matrix of high-heat-capacity material (Fig. 17). Energy is first transferred from the hot stream to the matrix, i.e., the regenerator packing material. When the cold fluid flows through the matrix during the next stage of the cycle, the energy is transferred from the solid material to the cold stream. The periodic nature of the operation of regenerators contributes to two important design characteristics: (1) the use of a matrix material produces a high-performance,

compact geometry; and (2) the transient nature of the device leads to complex mathematical analyses based on time-dependent temperature and heat transfer parameters.

There are several drawbacks associated with regenerative heat exchangers. The material stress due to the cyclic heating and cooling can damage their components and thus reduce their operational lifetimes. Another weak point is the mixing of the warm and cold streams during the switching process, which cannot be eliminated. These disadvantages have resulted in the very limited application of regenerative heat exchanges for continuous cryogenic processes, such as air separation and natural gas liquefaction. On the other hand, every regenerative heat exchanger can be thought of as a thermal energy storage device [74]. Thermal energy is stored in a porous matrix of high-heat-capacity material and used to heat or cool fluid flowing through the matrix. This unique feature of regenerators has renewed the interest in their research and development, especially for application in different energy storage technologies.

Regenerators can be found in various small cryogenic refrigerators (known as cryocoolers), e.g., magnetic, pulse tube, Stirling, Gifford-McMahon and Vuilleumier refrigerators [74]. Concerning large-scale applications, regenerators are particularly suited for LAES systems. This is because the round-trip efficiency (i.e., the ratio of the energy recovered by the system during the discharge stage to the total energy input) of a LAES system can be substantially improved when cold energy released by liquefied air during the discharge stage is stored and reused to reduce the work required for liquefaction [75,76].

According to Ackermann, the list of materials suitable for the matrix includes stainless steel, bronze, copper, lead antimony spheres and magnetic intermetallic compounds [74]. The author also lists four commonly used regenerator packing geometries, i.e., annular gap, wire mesh, dimpled ribbon and randomly packed sphere matrix (also referred to as packed bed). The annular gap geometry has been employed in early Stirling engines but has major limitations, i.e., the restricted heat transfer area and the small thermal penetration depth of the walls. Woven wire mesh screens are common in cryocoolers where they provide major advantages, among which low cost, availability in many different sizes and materials, and high utilization of the thermal capacity of the matrix, attributed to the small diameter and high thermal conductivity of the wire [74]. The dimpled ribbon regenerator is constructed of a ribbon, which has multiple flow channels formed by dimpling or embossing, wound on a mandrel. The flow is distributed to the channels through a flow header. This configuration can achieve very high ratio of heat transfer area to pressure drop, but a major concern is flow maldistribution which impedes the thermal performance. One solution to reduce this effect is the etched foil matrix where the pattern involves slits to provide cross flow between the channels [74]. The packed bed regenerator normally comprises spheres (or approximately spherical particles) of random size with diameters deviating from the average by ± 0.04 mm. The porosity is usually in the range of 32 to 37% [74].

As described by Morgan *et al.* [76], the regenerative heat exchanger in a LAES system is a very large structure, whereby its design and material selection play a crucial role in the tradeoff

between cost and thermal performance. They note that materials such as ferritic steels and many ceramics are unsuitable for cryogenic regenerators, as they are susceptible to damage due to thermal fatigue. From the viewpoint of heat capacity per unit volume and thermal conductivity, metals, such as stainless steel, copper or aluminium, are advantageous but can be prohibitively expensive for larger systems [76]. Therefore, cheaper materials, such as glass beads, gravel, pebbles or concrete, can be more attractive, considering the price per heat capacity and productivity of a large-scale cold storage unit. Araki *et al.* [77] proposed a cold storage unit with steel pipes inserted in the concrete medium. A thermal hydraulic model for simulation of cold storage performance has been developed and validated against experiments in a special test facility. However, Morgan *et al.* [76] note that in this case conduction losses normal to the flow direction are high which is a significant drawback. Hence, they preferred a packed bed regenerator for implementation in a large-scale LAES pilot plant [8]. The cryogenic regenerator of the plant consists of modular packed beds with quartzite rocks with a total volume of 9200 m³ [9].

An important consideration for grid support energy storage applications, e.g., LAES, is the fluctuation of power supply and demand. This may result in incomplete charge and discharge cycles. The major implications are variations of the fluid flow rates and some degree of self-discharging of the regenerator, i.e., heat transfer from the hot to the cold regions of matrix, which in turn reduces the quality of the stored energy. To address this issue, Morgan *et al.* [76] propose a multi-cell solution where individual cells of packed beds are arranged in parallel or series depending on the flow rate. This configuration is equivalent to a variable geometry of the packed bed, which can be adjusted according to the load. Another major benefit is that individual cells can be isolated from the rest, thereby limiting the self-discharging effect. Based on modelling and simulation with the Flowmaster software, the authors observe a 4.8 % improvement of the efficiency over a single-cell packed bed, without accounting for the additional benefit of reduced self-discharge losses [76].

Various other studies have investigated the heat transfer processes in packed bed regenerators. White *et al.* presented a numerical and theoretical analysis of thermal wave propagation in a packed bed thermal reservoir [78]. This item is a part of pumped thermal energy storage and works in the cryogenic temperature range. As the solid storage medium exhibits significant changes in its specific heat capacity, the resulting non-linear wave propagation may lead to the formation of shock-like thermal fronts. Such effects lead to exergy losses due to irreversible heat transfer and should be considered during the design and optimization of the storage.

Anderson *et al.* presented experimental results and simulation model of sensible heat storage in a packed bed of aluminum oxide (alumina) particles [79]. The simulation model analyzes the heat transfer between compressed air and solid packing consisting of alumina spheres with a nominal diameter of 3 or 6 mm. The validation of the model is done via comparison with experiments performed in a laboratory-scale pilot plant unit. Finally, a two-equation model is presented that can predict experimental results in a thermal energy storage vessel comprising a packed bed with a compressed gas as heat transfer fluid. Two equations determine the

transient fluid and solid temperatures in the axial direction of the bed. Later on, Anderson *et al.* presented a simplified model of the behavior of a packed-bed regenerator, using aluminum spheres as sensible heat storage medium and air as heat transfer material [80]. A Navier–Stokes solution of the fluid flow has been coupled to the model to calculate the transient temperature field in the packed bed. The model has been successfully validated against experimental data obtained from a test facility, a schematic of which is shown in Fig. 18.

However, the simplified thermal model approach is beneficial exclusively if the heat capacity and the thermal conductivity of the solid material are much higher than those of the heat transfer fluid.

Bindra *et al.* [81] underlined that two major factors lead to exergy destruction in packed-bed sensible heat storage systems: pressure drop and temperature dispersion. To reduce the pressure drop, they proposed a new sliding flow concept, whereby the inlet and outlet ports for the fluid flow are changed according to the propagation of the temperature front in the bed. Bindra *et al.* described their innovative approach as follows [81]. It is well known that the conventional method of storing sensible heat in packed beds has the flow entering from one end of the bed and leaving at the other end. Correspondingly, the fractional length of the dispersed front, i.e., the ratio of dispersed temperature front to the total length, determines the destroyed exergy from the bed. The longer the length of the bed, the smaller is the fractional thermal exergy destruction, but a longer bed also causes higher pressure drop, which results in exergy destruction as well. The sliding flow method (SFM) has been developed, with the aim to reduce fractional axial dispersion and pressure drop simultaneously. This method is named ‘sliding flow’ as the flow path, inlet and outlet ports to the storage vessel/system change during the storage operation as the temperature front moves in the bed, unlike the conventional storage method. In the SFM, instead of a bed with one inlet and outlet, the bed is divided into many segments such that the length of each segment is roughly equal to the length of the dispersed front. The arrangement is such that fluid inlet and outlet can be connected to any of these segments. Illustration of this design is shown in Fig. 19, where a full length packed bed is divided for instance into four segments.

During the storage stage of a cycle, hot fluid is introduced to the first segment, and after transferring its heat into the first two segments, it exits through the second bed outlet. With this arrangement, the pressure drop, across the portion of the packed bed in which the flow takes place, is only a fraction of the pressure drop incurred in the original method. After the first segment, it is fully saturated, or the fluid exiting from it reaches its steady peak temperature, there is no need to feed more hot fluid to the first segment. At that point, a switch is activated to introduce hot fluid at the entry of the second segment and cold fluid exits from the end of the third segment. This changeover is done again after saturating the second segment. Then the fluid is fed into the third segment and leaves out of the fourth segment. Therefore, with this method, the instantaneous flow path is always equal to the length of two segments, or in other words, for this example, the pressure drop is always half as compared to the pressure drop in the original method with full packed bed always in-line. This method of

separating pressure drop effects from heat transfer effects ensures that axial dispersion remains the same. The authors have conceded that due to division into segments, the system requires additional valves and piping which can lead to higher capital costs. The numerical example used in the paper for the SFM with four segments shows the considerable improvement in the overall exergy of the system by decreasing the pressure drop. If the pressure drop can be reduced in these process systems, it can reduce the operating costs. Further optimization based on this increment in capital costs and reduction in operating costs is required to make final investment decisions.

Sciacovelli *et al.* [9] developed a dynamic energy model of a packed bed regenerator which was validated against experimental data from a LAES pilot plant, located at the University of Birmingham. Parts of the model are developed in EES and COMSOL, and the coupled simulations are performed through MATLAB. The simulations show a strong temperature gradient between the hot and cold end of the packed bed, but the stratification degenerates rapidly over the charge/discharge cycles. A steady temperature profile is established after 15-20 cycles [9]. The degradation of the thermal front in the packed bed leads to 25 % increase of the air outlet temperature at the end of the discharge cycle, which in turn reduces the recovered cold by the same amount. Similarly, the degradation of the thermal front at the end of each charge cycle reduces the efficiency of the liquefaction process [9].

Using liquids, PCMs or PCM slurries for hot or cold thermal energy storage is a contemporary trend when developing LAES systems and associated heat exchangers [82, 83]. A number of cryogenic mixed refrigerants [82] can be employed as heat transfer fluids and cold accumulators. For instance, Li *et al.* [83] proposed a mixture of propane and methanol, which is operational at a wide low-temperature range and possesses a high heat capacity. The mass flow rate and storage volume needed for such fluids are close to those for liquid air, while cold storage by solid media and gaseous heat transfer fluids requires a storage volume approximately 10 times larger than the liquid air storage volume [77]. Thus, involving liquid storage media makes the heat exchangers and the integrated LAES systems much more compact, with a huge potential for reduction of capital costs [83].

In any case, the choice of type and material for a cryogenic regenerator is a matter of cost-benefit analysis, involving various factors, such as capital costs, system complexity and long-term performance, so that there is no one-size-fits-all solution.

5. Concluding remarks

This review highlights the recent advancements in the design and operation of cryogenic heat exchangers for large-scale applications. After being intensively used for air separation for many years, cryogenic HEs have found another prominent role in natural gas liquefaction. Liquid Air Energy Storage (LAES) is another industrial application where cryogenic heat exchangers are likely to be employed to a much greater extent in the future.

Contemporary thermodynamic and economic considerations pose stringent efficiency requirements, which result in the need for sufficiently accurate simulation models. Until recently, these models were usually implemented in proprietary software. Their key features were related to the experimental correlations used to determine the fluid properties, heat transfer, and pressure drop characteristics, obtained in the manufacturer's laboratories. However, recent developments of simulation and design models have shifted state of the art to more advanced numerical methods.

The above-analyzed recent progress in cryogenic HE research and development is expected to enlarge the manufacturing capacities and to foster competition, which in turn can reduce the production and selling prices of this critical equipment.

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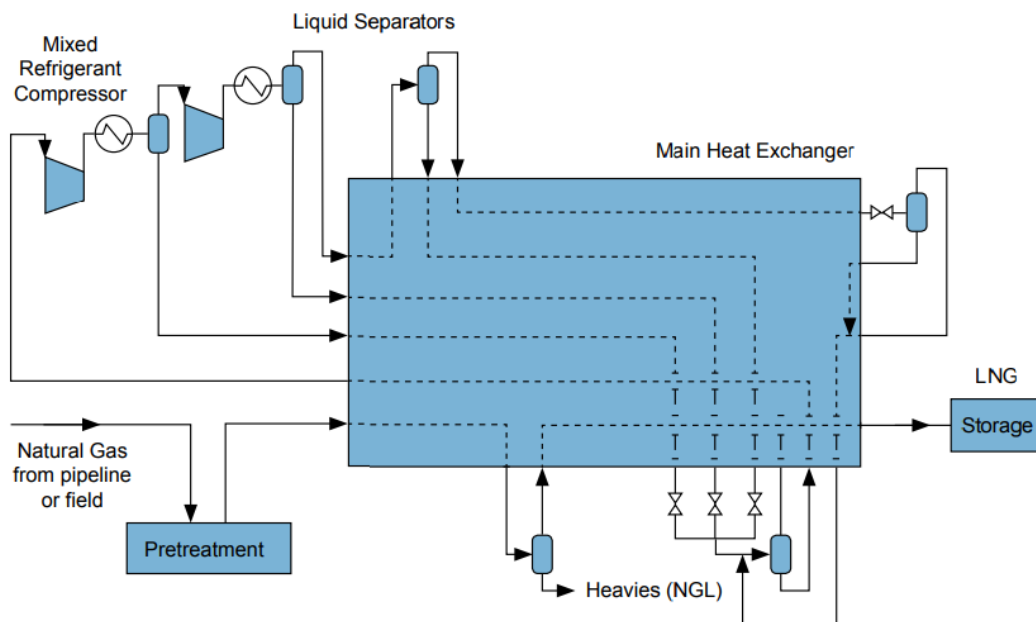


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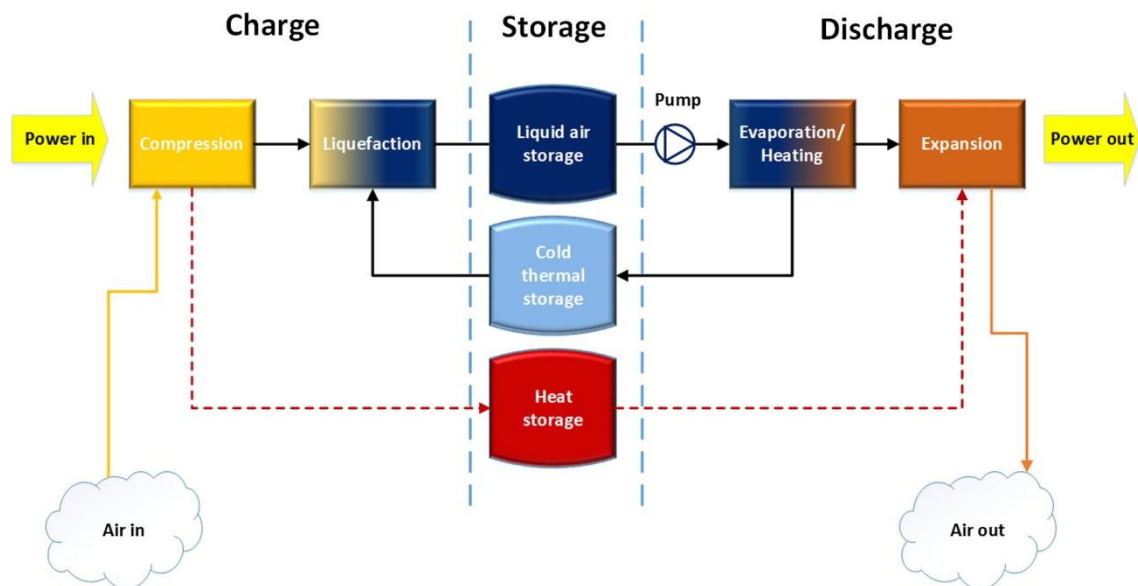


Fig. 2 Process flow diagram of liquid air energy storage plant (Sciacovelli *et al.* [9])

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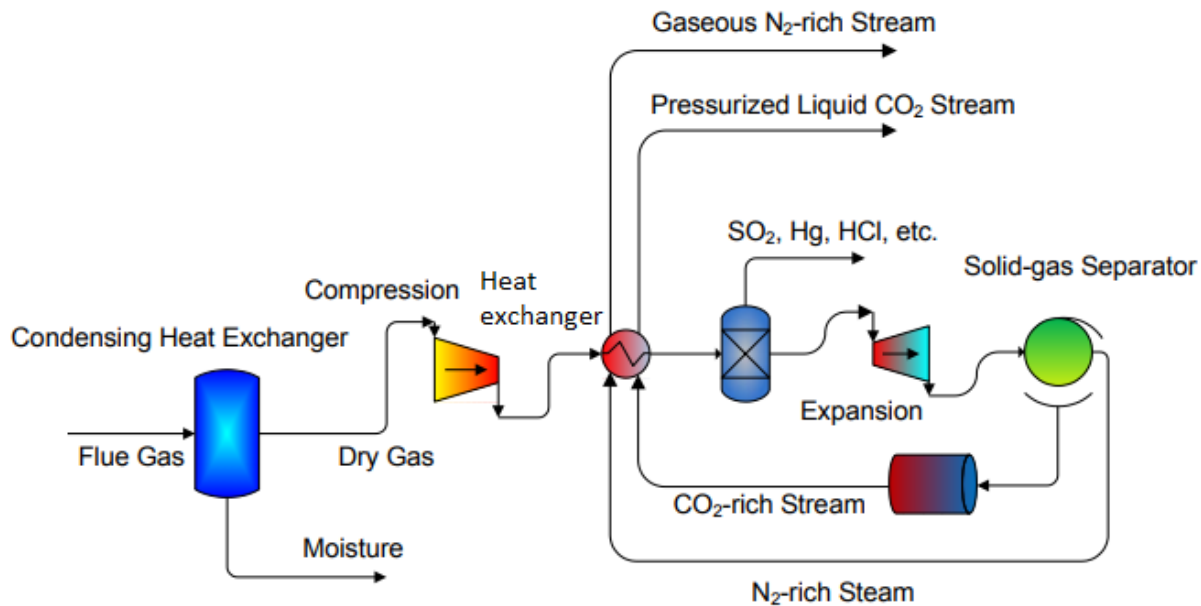


Fig. 3 Process flow diagram of the cryogenic carbon capture (Baxter *et al.* [10]).

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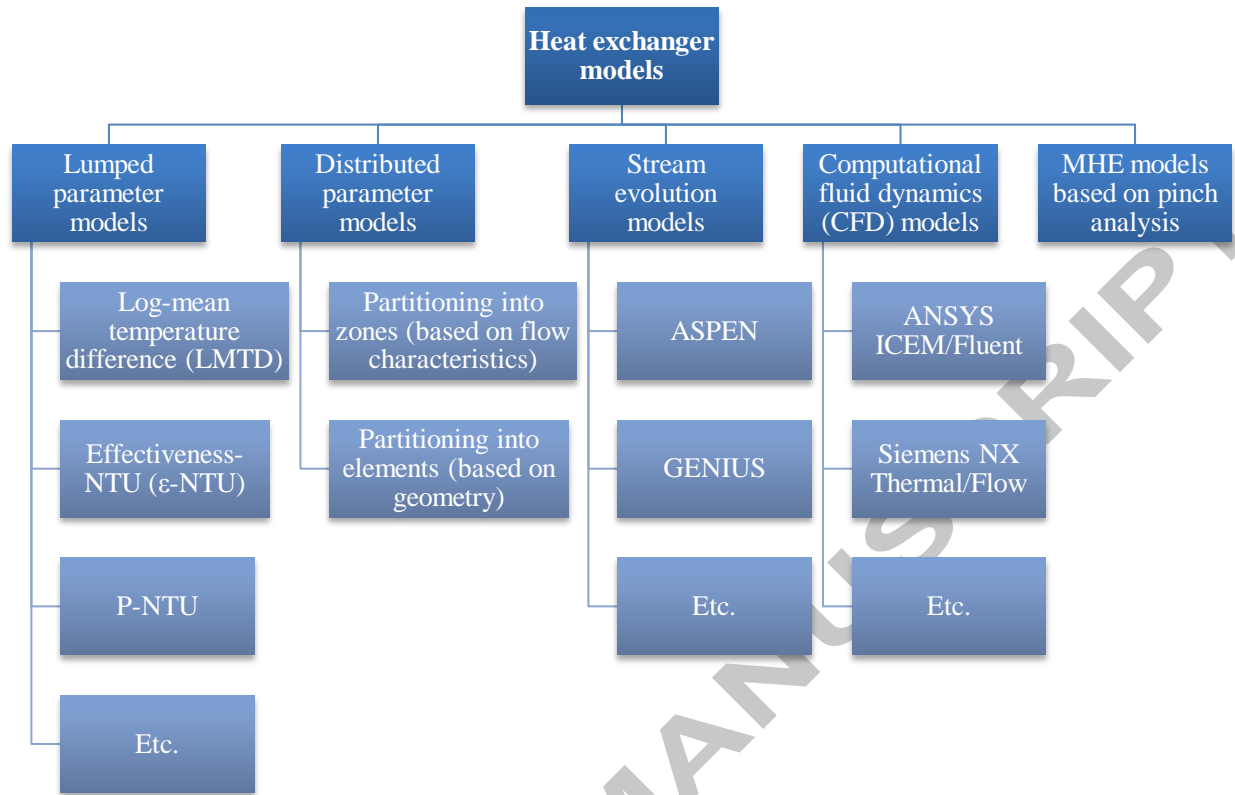


Fig. 4 Types of heat exchanger models (Pacio [2]).

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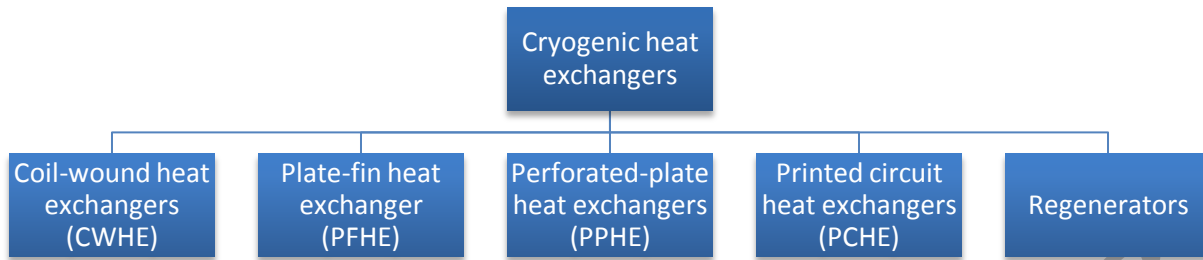


Fig. 5 Types of heat exchanger used in cryogenic systems.

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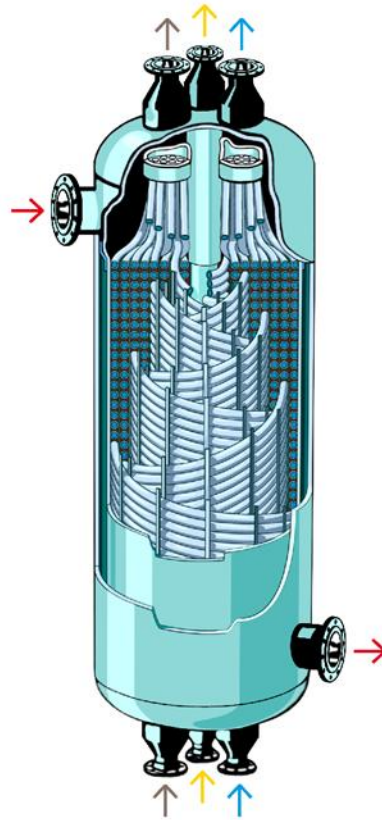


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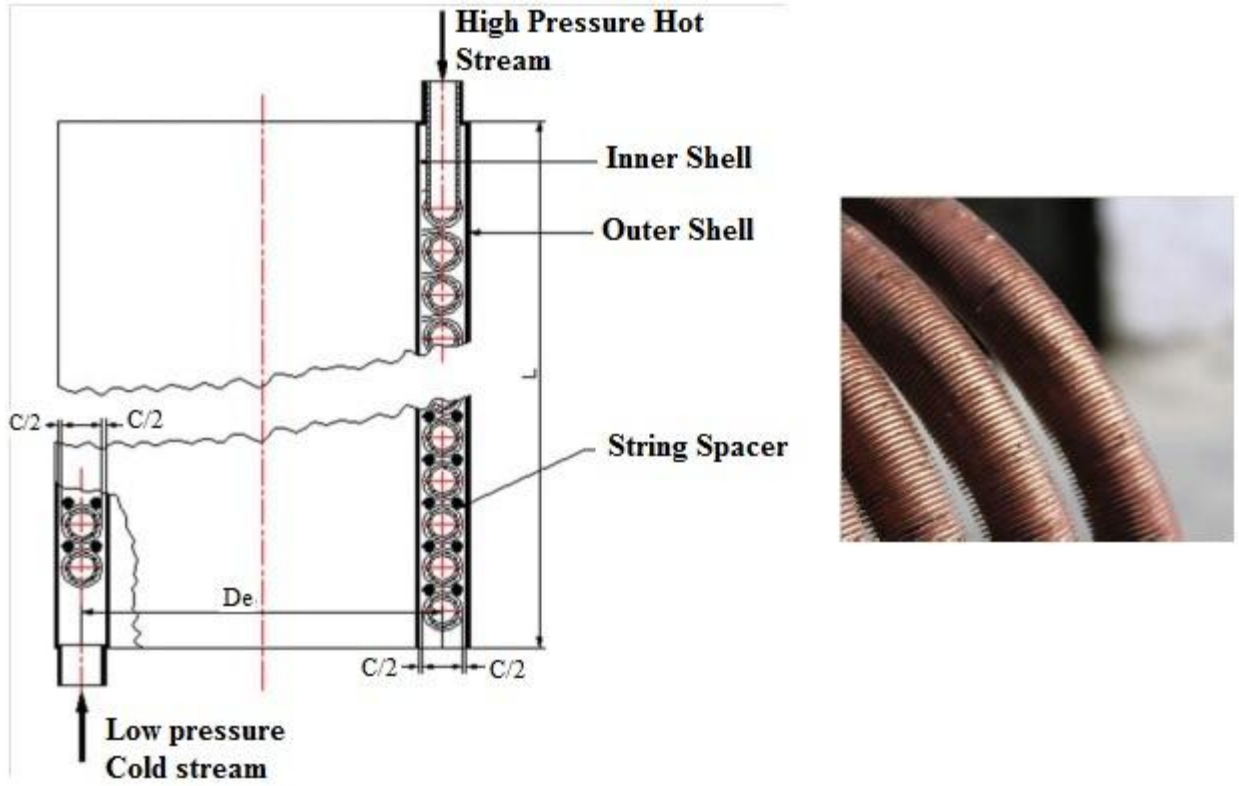


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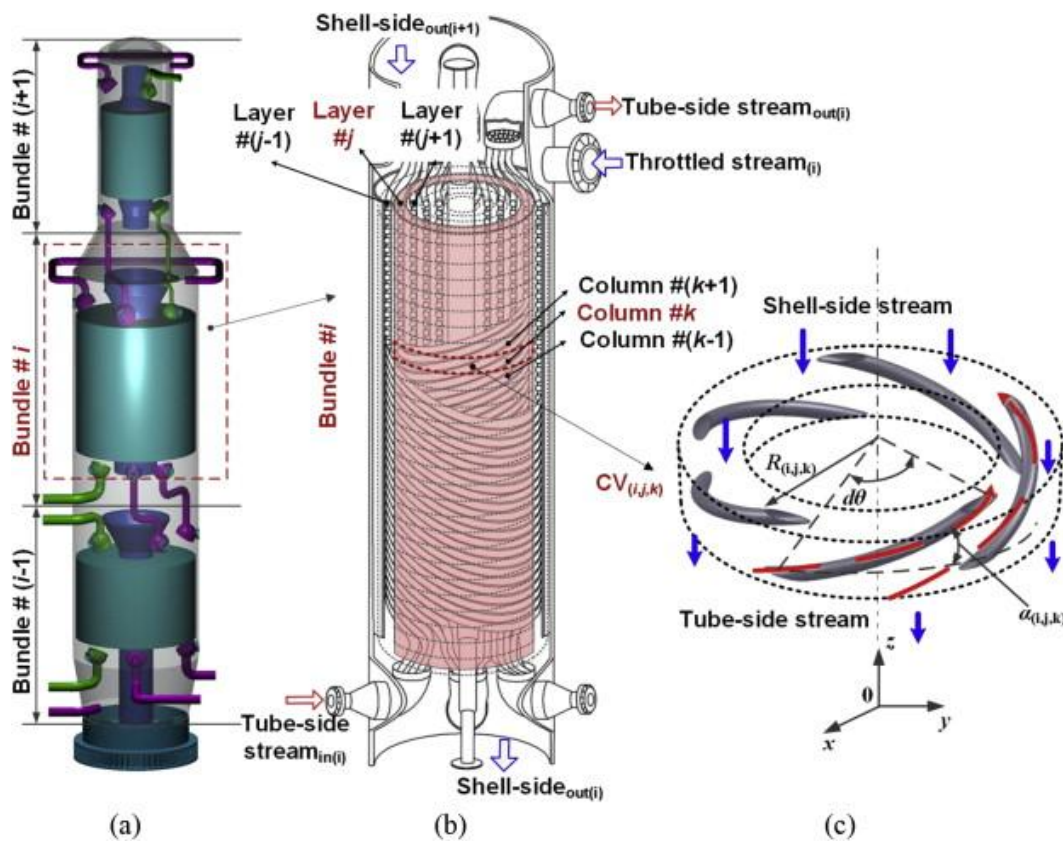


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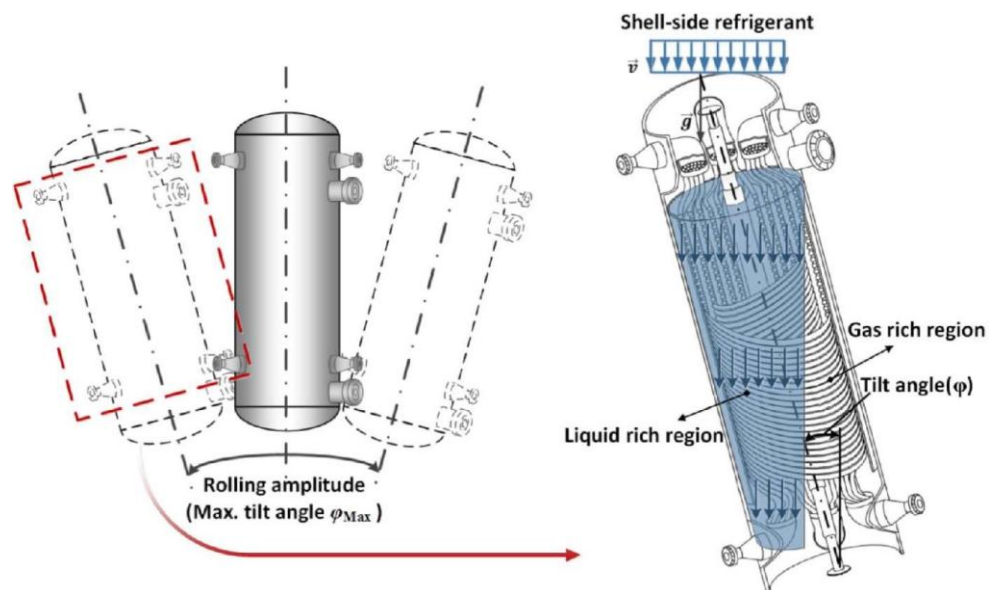


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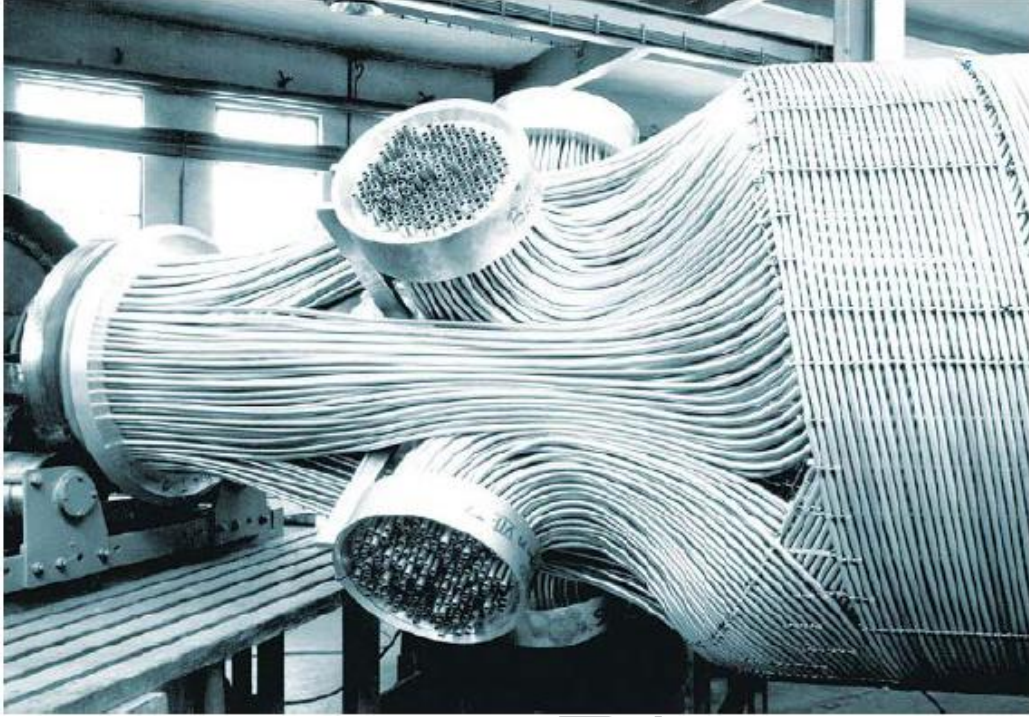


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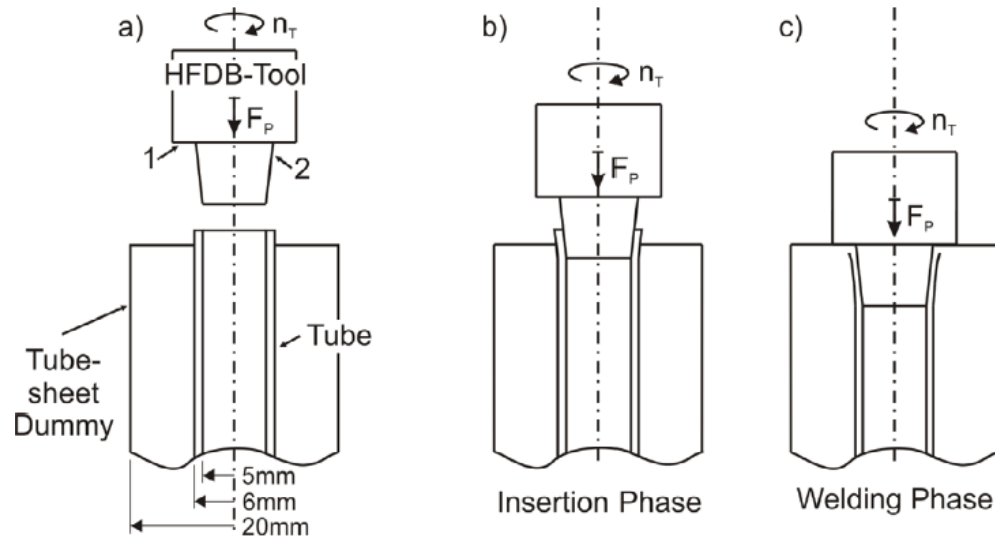


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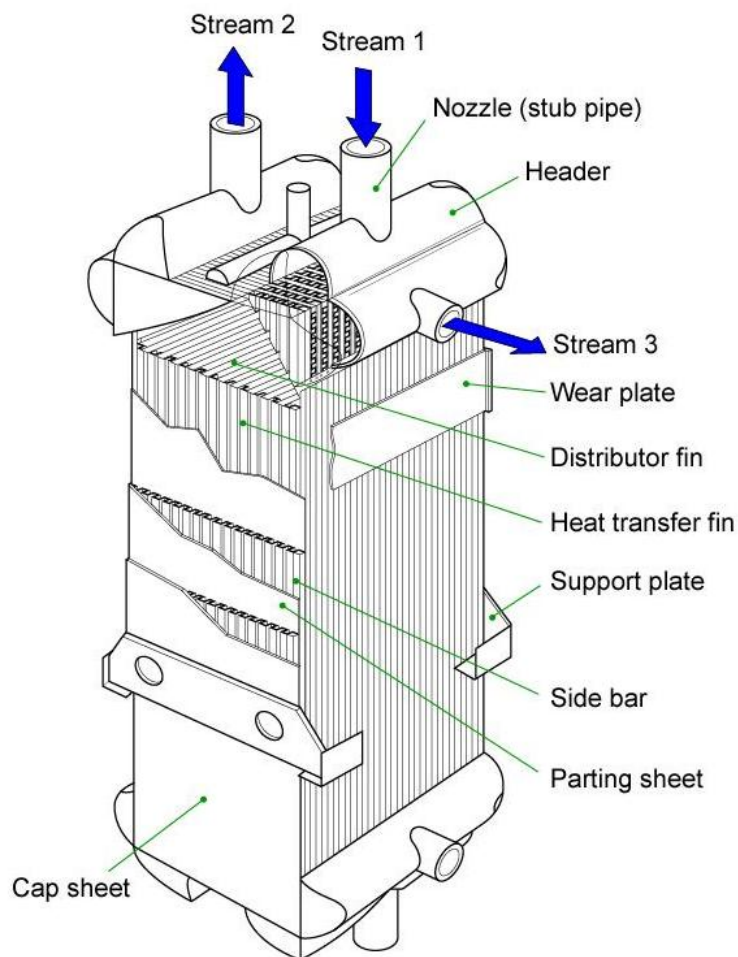


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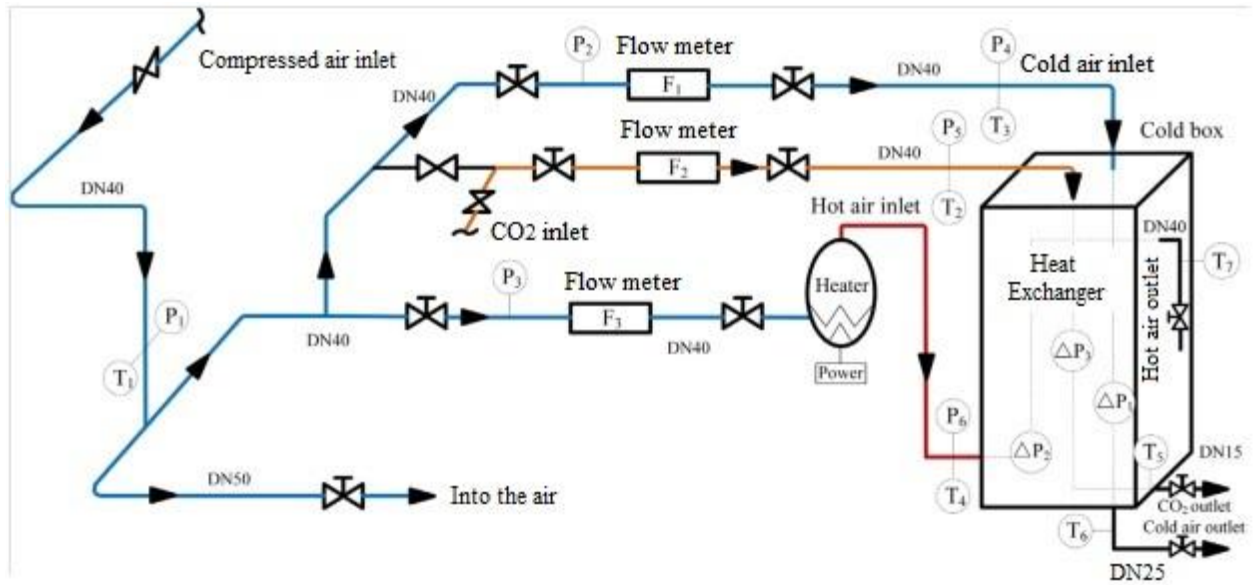


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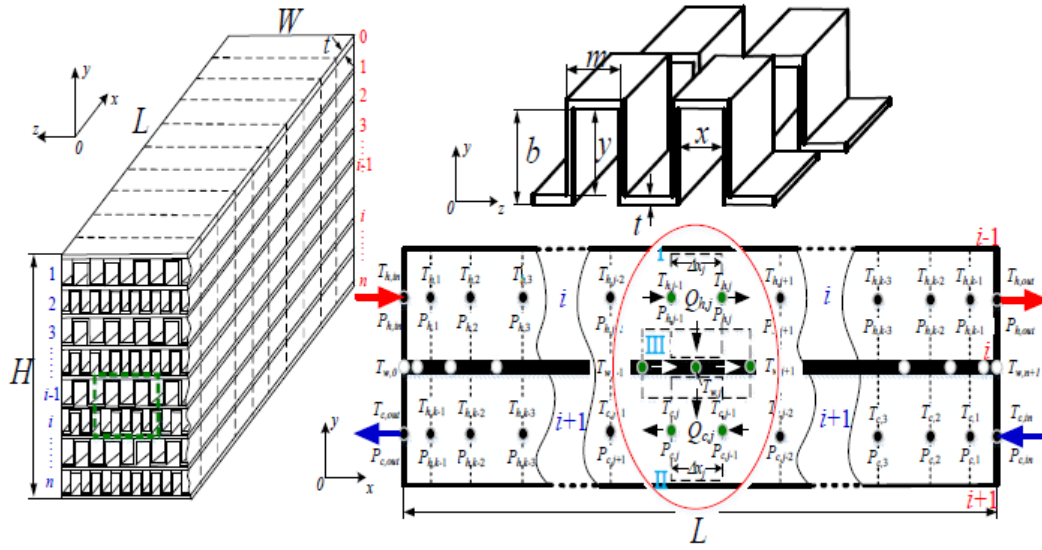


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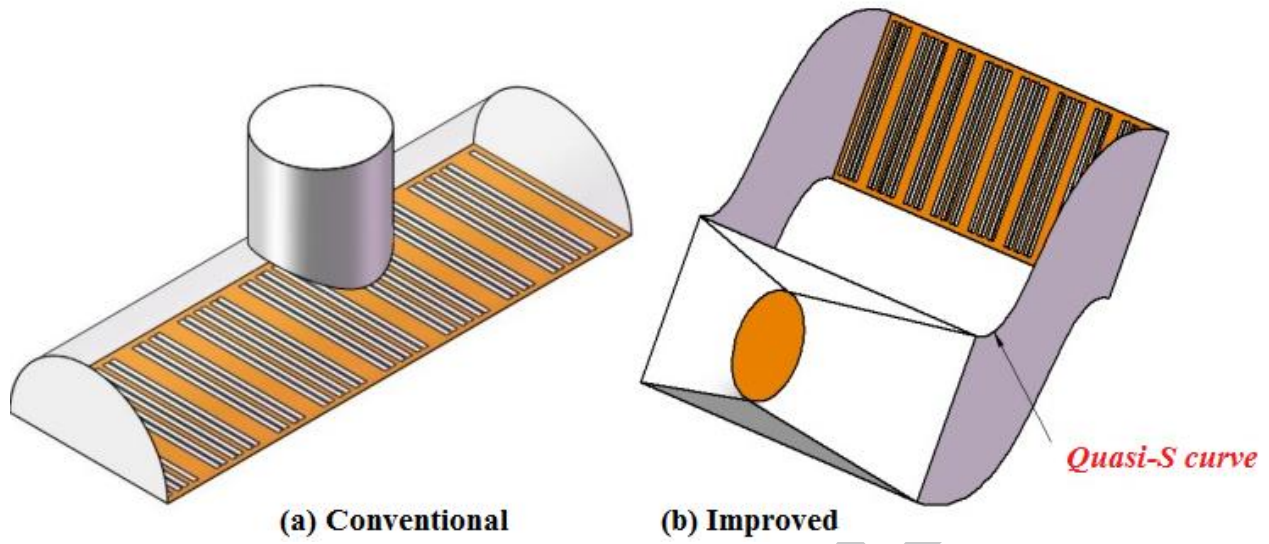


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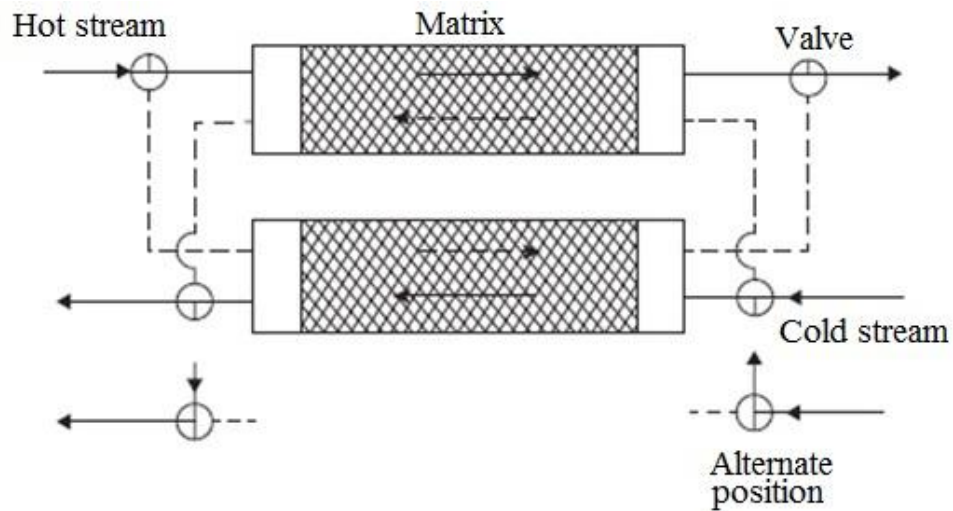


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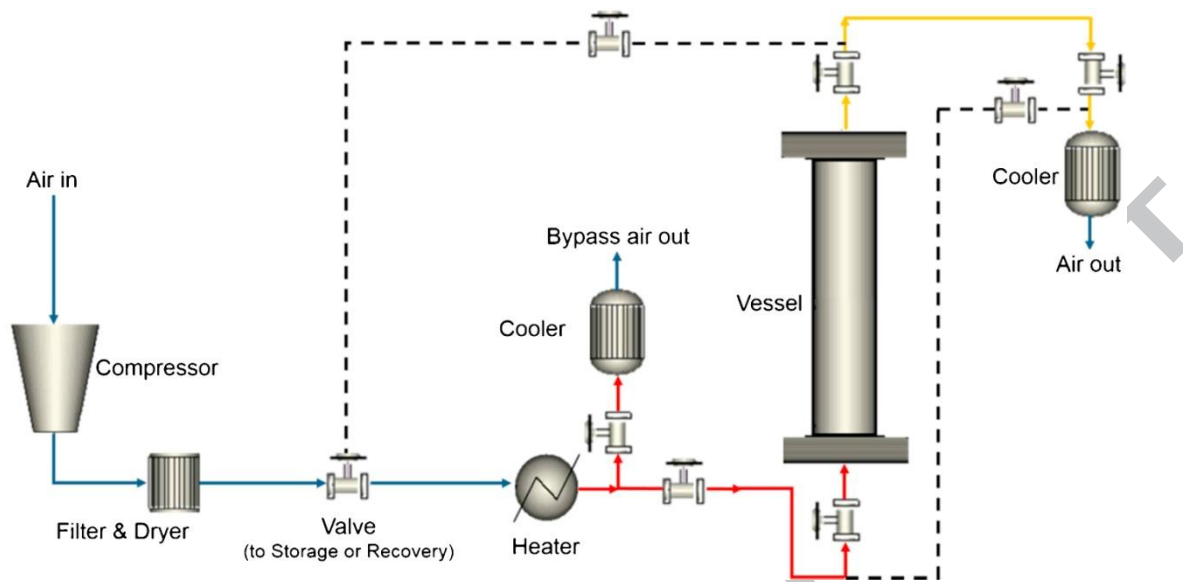


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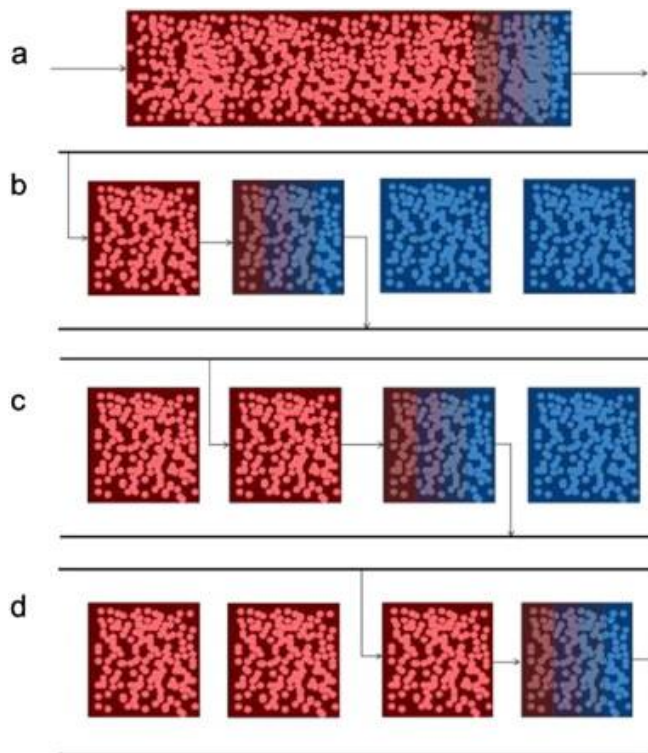


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Highlights:

- Pioneering synopsis of present cryogenic heat exchangers in energy storage systems
- First-of-its-kind review of trendy heat exchangers in a cryogenic technology context
- Spotlight on cryogenic energy storage as a novel technology to integrate renewables
- Deliberation upon the impact of heat exchangers' design on energy storage performance
- Outline of innovative modelling and design methods, alongside recent research trends

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