REVIEW ARTICLE

INTERNATIONAL RESEARCH JOURNAL OF
MULTIDISCIPLINARY TECHNOVATION

Asian Research Association

DOI: 10.54392/irjmt2332

Printed Circuit Heat Exchangers (PCHEs): A Brief Review

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DOI: https://doi.org/10.54392/irjmt2332

Received: 18-10-2022; Revised: 01-02-2023; Accepted: 12-02-2023; Published: 20-03-2023

Abstract: Heat exchangers and other heat transfer devices/systems play vital roles of heat transfer in thermal fluid flow systems for industrial application. Sodium cooled fast reactors are normally designed to have two loops of sodium coolants and one loop of water coolant which generates steam for power production. The two loops of sodium coolants consist of primary cooling system of sodium which cools the fuel rods of the reactor core and secondary cooling system of sodium transferring heat from the sodium primary cooling system. The water-cooling system transfers heat from the secondary cooling system of sodium for steam generation. Lead cooled fast reactors on the other hand are designed to have primary cooling system of lead cooling the fuel rods in the reactor core and secondary cooling system of water transferring heat from the lead cooling primary system for steam generation. Water cooled Nuclear Power Plants used water to cool the reactor core in the primary system and the heat removed from the core is used for steam generation directly as in BWRs and SCWRs or in the secondary system of heat exchanger as in PWRs. Other reactor systems such as Gas-cooled fast reactor (GFR), Molten-salt reactor (MSR), High-temperature gas-cooled reactor (HTGR), and Small Modular Reactors (SMRs) also have various types of heat exchangers in their designs to support power/electricity generation. Appropriate heat exchangers are therefore needed for various stages of heat transfer in power generation systems. Thus, Heat exchangers and other heat transfer devices/systems play vital roles of heat transfer in thermal fluid flow systems for industrial applications. This study presents brief review of PCHEs which have comparable advantages over other types of heat exchangers. Recent studies on PCHEs and other heat exchanger types have been reviewed. Design and optimization of PCHEs, optimization of Brayton and Rankine circles, and fluid flow and heat transfer devices/systems have been discussed briefly. The review findings show that the design and optimization of PCHEs depends on the intended industrial application of the heat exchanger. The various channel types and channel cross-section types available for design and optimisation as well as the design and optimised system being able to withstand high pressure and temperature conditions in addition to its compact size for the intended industrial application make PCHEs unique among other types of heat exchangers.

Keywords: Printed Circuit Heat Exchangers, Heat Transfer Devices/Systems, Design and Optimization, Reactor Systems, Brayton and Rankine cycles

1. Introduction

Fluid flow and heat transfer is one of the industrial processes upon which heat exchangers are designed for many industrial applications. Heat transfer from thermal fluid flow systems is important for safe operation of many industrial equipment/systems. Inappropriate heat removal from heat transfer systems could lead to mechanical failures of these systems because of heat built-up and eventually affecting the systems' material integrity. Heat exchangers and other similar heat transfer devices/systems are mainly used for extraction of heat from one thermal fluid flow system to another system for industrial applications. There are several types of heat exchangers which form part of many industrial installations. These types of heat exchangers include Tubular Heat Exchangers, Plate Heat Exchangers, Tube Fin Heat Exchangers, Printed Circuit Heat Exchangers, and Regenerative Heat Exchangers among others. One of the important performance factors of all the heat exchangers is the compactness factor, the amount of heat transfer surface area within the volume of the heat exchanger and is measured in square meters per cubic meter. Other performance parameters are briefly discussed in Section 2.

The design and operation of new advanced systems for power generation and other applications require the use of heat exchangers that can withstand adverse high temperature and pressure conditions in order to prolong the lifespan of these new advanced systems. Printed Circuit Heat Exchangers (PCHEs) have been proven to have comparable advantages over other types of heat exchangers based on various studies highlighting their use for high pressure and temperature industrial heat transfer applications. Table 1 provides scope and main findings of recent reviews carried out on PCHEs and other heat exchangers. Table 2 provides information on study geometry, means of data collection, study type, purpose and main findings of recent studies carried out on PCHEs.

The main purpose/aim of research with respect to fluid flow and heat transfer is design and operational parameters optimization to get the optimum economical design and operational parameters. Experimental studies are first carried out to establish the feasibility of the research and also to provide experimental/validation data for validation of similar numerical studies. Numerical studies using modelling tools are then carried out to determine these optimum economical design and operational parameters as the experimental studies are mostly complex and expensive to be used for the optimization studies. Design and optimization of PCHEs have been briefly discussed in Section 2. Brief discussion on optimization of Brayton, Rankine and organic Rankine cycles have been provided in Section 3. Section 4 provides brief descriptions of different types of heat transfer devices such as boiler, steam generator, condenser, radiator, evaporator, cooling tower, regenerator, recuperator, heater, reheater, preheater, cooler, intercooler, aftercooler, precooler, economizer and superheater.

Table 1. Scope and findings of recent reviews on Printed Circuit Heat Exchangers and other heat exchangers

Review reference	Scope of the review and the main findings from the review
Sadeghianjahromi and Wang [1]	Sadeghianjahromi and Wang carried out review on experimental and numerical investigations on fin-and-tube heat exchangers. Different mechanisms of heat transfer enhancement were compared, effects of different geometrical parameters on heat transfer and pressure drop in each mechanism, and fouling and coatings on surfaces of fin-and-tube heat exchangers were discussed, proposed correlations regarding fin-and-tube heat exchangers were also compared, and operating conditions and limitations for the different heat transfer mechanisms and correlations were specified. The main findings include: (i) Heat transfer augmentation in fin-and-tube heat exchangers can be achieved by the three different heat transfer mechanisms such as increase in heat transfer area, rise of heat transfer coefficient, and improvement of temperature difference. (ii) Using plain and wavy fins can increase heat transfer area. Louver and slit fins cause increase in heat transfer coefficient by boundary layer restarting, while using vortex generators augments mixing of hot and cold fluids, which leads to enhancement of heat transfer coefficient. Partial bypass is one of the ways for increasing temperature difference. (iii) Geometrical parameters play pivotal roles in heat transfer and pressure drop characteristics of fin-and-tube heat exchangers with different fin types. *N* (number of tube rows), *D* (tube diameter), *Fp* (fin pitch) have important effects in plain fins, while *Pl* (longitudinal tube picth), *Pt* (transversal tube pitch), *t* (fin thickness) have insignificant influence. *αw* (wavy angle), *αl* (louver angle), and *αvg* (vortex generator angle of attack) cause considerable effects in wavy fin, louver fin, and vortex generators, respectively. Louver and slit fins are highly affected by *Lw* (louver width), *Sw* (slit width), respectively. (iv) Compound enhancement mechanisms are helpful in further improvement in the performance of fin-and-tube heat exchangers so that it can be more applicable in a wider operating range. (v) Surface treatments such as using hydrophilic, hydrophobic, and desiccant coatings on fin- and-tube surfaces can affect the performance of fin-and-tube heat exchangers under dehumidifying, frosting, and defrosting conditions. (vi) Fouling, particle deposition, and thermal contact between fins and tubes are important factors influencing the heat transfer and pressure drop characteristics of fin-and-tube heat exchangers, and these influences should be taken into consideration in design, fabrication, and assembly of heat exchangers.

	(vii) Developed correlations for determining heat transfer and pressure drop characteristics of fin-and-tube heat exchangers with different fin types and circular and flat tubes under dry and wet conditions are tabulated, including comparisons between them.
Liu et al. [2]	Liu et al. carried out review on the thermal-hydraulic performance and optimization of printed circuit heat exchangers PCHEs for supercritical CO_2 in advanced nuclear power systems. Background information about supercritical fluids, the supercritical CO_2 Brayton cycle and PCHEs; and heat transfer and pressure drop in PCHEs with different channel types in the supercritical CO_2 Brayton cycle, have been discussed. The industrial feasibility and maturity level of PCHEs with different channel types, which have important implication for the large-scale application of PCHEs have also been discussed. The findings from the review are: (i) Several studies carried out on the effects of channel structure and operating parameters of heat transfer and pressure drop in PCHEs. (ii) Few theoretical and experimental studies carried out analyzing thermal-hydraulic performance of PCHEs (iii) The zigzag channel with a semicircular cross-section is found to be the preferred channel type for the supercritical CO_2 side of PCHEs based on the analysis of thermal-hydraulic performance, pressure resistance, maturity level and manufacturing cost. (iv) The literature review indicated that the PCHE is an efficient and compact heat exchanger, which is a reasonable candidate for heat exchanger in the supercritical CO_2 Brayton cycle. (v) Several channel types have been developed, and the thermal-hydraulic optimizations of the geometrical and operating parameters were carried out mainly by numerical simulation. (vi) A collaborative application of the chemical-etching plate and the formed plate after compression, which could remarkably reduce the cost of PCHE, has a good prospect in the IHXs (Intermediate Heat Exchangers) and the coolers in LFRs, MSRs and SFRs.
Ali et al. [3]	Ali et al. carried out review on common failures in heat exchangers. Common modes and causes of failure of heat exchanger tubes have been discussed. Heat exchanger failure investigations based on the mechanisms of failure have also been discussed. The main findings are: (i) A large majority of heat exchanger components fail by way of fatigue, creep, corrosion, oxidation, and hydrogen attack. (ii) Most common causes of failure include fouling, scaling, salt deposition, weld defects, and vibration. (iii) The service life of heat exchangers can be prolonged by factors such as suitable materials selection, appropriate tubes design (heat exchanger design), use of skilled workforce, and strict control of operating conditions such as fluid chemistry, pH, velocity, temperature, pressure and level of vibrations.
Chai and Tassou [4]	Chao and Tassou discussed the performance of PCHEs and also provided survey of heat exchangers (HEs) available for commercial activities. Heat transfer and pressure drop characteristics of PCHEs with different flow passages; geometric design optimisation of PCHEs taking into consideration the complex relationships between heat transfer enhancement and pressure drop penalty, compactness, fluid inventory and capital cost; optimisation of flow passage configurations in terms of thermohydraulic performance, complexity and manufacturing costs; material selection, manufacturing and assembly to further increase the range of high temperature and pressure operation; and development of more generalized correlations for performance prediction and design optimization, were discussed. The main findings include: (i) PCHEs relate to high-strength, high-temperature materials. At operating temperatures lower than 650 °C, 316/316L/347 stainless steel can be employed for the manufacture of PCHEs whereas for operating temperatures higher than 650 °C, nickel-based alloys, such as Alloy 625 or 617, can safely be employed but at much higher capital cost. (ii) Creep and corrosion are the two most important factors influencing material selection, but very little experimental data is available for helium or supercritical CO_2 in high-temperature environments.

	(iii) Photochemical machining and diffusion bonding are the two main processes of PCHE manufacture. Photochemical machining provides flexibility in thermohydraulic design, and diffusion bonding forms a compact, strong, all-metal heat exchanger core. (iv) Four main types of PCHE flow passages that include: straight channel; zigzag (or wavy) channel; channel with S-shaped fins; and channel with airfoil fins, have been developed and tested at test facilities in the USA, Japan, Korea and China using helium and supercritical CO_2 as the working fluids. Their thermohydraulic performance has been investigated both experimentally and numerically. The test conditions for helium were 900 °C and 3.0 MPa, while for supercritical CO_2 test conditions were 500 °C and 20 MPa. Generally, PCHEs with airfoil fins showed best performance, followed by S-shaped fins and zigzag (or wavy) channel PCHEs. (v) Several authors have developed empirical correlations for the prediction of average heat transfer and pressure drop characteristics. However, most of the correlations were developed for a specific flow passage and using thermophysical properties corresponding to the average temperature and pressure at inlet and outlet conditions. (vi) Several optimization techniques for PCHE design have also been developed, usually based on the relationship between the Euler and Nusselt numbers and the cost-based objective function consisting of the capital cost of the PCHE and operating costs by assigning relative weighting factors to heat transfer and pressure drop for different operating conditions.
Huang et al. [5]	Huang et al. carried out a review on experiments and numerical simulations in relation to characteristics of flow and heat transfer in PCHEs with single-phase flow supercritical carbon dioxide or helium working fluid. Correlations related to flow and heat transfer in PCHEs were also discussed. The main findings obtained from this review include: (i) There is the need to carry out further studies on single-phase flows with S-CO_2, or helium as working fluid at high Reynolds number in experiments relating to widely accepted PCHEs with semi-circular Zigzag channels. Boundary conditions and turbulent models should be considered in detail in numerical simulations in order to obtain high accurate results compared with experimental ones. (ii) The geometric structures and operating conditions should be taken into account to deeply investigate the characteristics of flow and heat transfer in PCHEs, especially for S-CO_2 Brayton cycle used in the fourth-generation nuclear system. And more universal correlations should be proposed for the design of PCHEs. (iii) Recently, much attention has been paid to spiral wound heat exchangers (SWHEs) used in Liquefied Natural Gas (LNG) fields. However, PCHEs also have reasonable applications in LNG fields, especially in floating production storage & offloading system (FPSO). So the following studies should be investigated: (a) the characteristics and correlations of complex flow and heat transfer, considering multi-components of natural gas and mixture cryogen; (b) the design and optimization of PCHEs for LNG fields; (c) the sloshing effect on flow and heat transfer performances when PCHEs are used in FPSO; (d) the scaling problem of channels flow with soft water.
Javadi et al. [6]	Javadi et al. carried out review on performance of ground heat exchangers. The effects of ground heat exchanger geometry, pipe material, working fluid, and depth of ground heat exchanger on heat flux, heat transfer coefficient, outlet temperature, thermal resistance, and pressure drop have been discussed. The main findings include: (i) Among the studies investigating ground heat exchangers, numerical investigations are the most common type of study, followed by experimental and analytical studies. Optimization studies are the least common. (ii) When applying short-term and long-term thermal loads, helical BHEs (Borehole Heat Exchangers) exhibit better thermal performance than W-tube and U-tube BHEs. However, in helical BHEs, the pressure drop is much higher than for triple U-tube BHEs. Interestingly, by increasing the helix diameter and helical coil length of a helical BHE, the BHE performance improves, while the effect of helix pitch is negligible. (iii) A variety of working fluids have been used in ground heat exchangers (GHEs) and, among them, pure water is preferred because of its availability and low cost, as well as its great potential in storing and recovering thermal energy.

Table 2. Overview of recent studies carried out on Printed Circuit Heat Exchangers

Reference/Study (Study type, e.g., Experimental, Numerical)	Geometry, shape of the channel and channel cross section adopted. Measuring device or Computational tool adopted for data collection.	Purpose of study and findings from the study
Zhang et al. [7] (Experimental and Numerical study)	Figures 1, 2, 3 and 4 show geometry, channel shape, and working fluids used in the study. Numerical tool: Commercial software ANSYS Fluent 19.2. Measuring devices: K-type thermocouple (Exhaust) and thermal resistance PT100 (CO_2); Temperature and pressure sensors; a Coriolis mass flowmeter (CO_2), Air flow meter, Fuel mass flowmeter	**Purpose:** To perform thermal hydraulic analysis of PCHE in exhaust waste heat recovery by considering the differences in physical properties and working conditions between the exhaust gas side and the supercritical CO_2 side. **Findings**: (1) The experimental test results show that the PCHE with a novel structure is stable and feasible (2) The numerical results show that the comprehensive performance of the PCHE with an optimised structure was improved by 11.92% (3) An optimal PCHE structure (rectangular channels + hollow channels) obtained via simulation performed better compared with the PCHE prototype (expanded channels + traditional channels) by reducing the power loss caused by the pressure drop and weight by approximately 40.43% and 16.98%, respectively.
Lee et al. [8] (Numerical study)	The geometry and channel shapes adopted for the study are shown in Figures 5 and 6. Numerical tool: PCHE code was developed theoretically to obtain the study results.	**Purpose:** To carry out thermal-hydraulic performance evaluation and economics assessment of PCHE using CO_2 and N_2 as working fluids based on the channel type (Straight, Zigzag, Wavy and Airfoil), shape variables (zigzag: angle; wavy: pitch; airfoil: vertical and horizontal numbers), tube diameter, and Reynolds number with the aim of using PCHE for recuperators of Sodium cooled Fast Reactors (SFRs). **Findings**: (1) The zigzag type with high heat transfer performance is the most economical channel type in the CO_2 Brayton cycle, and (2) The airfoil type with a low pressure drop is superior in the N_2 Brayton cycle.
Yang et al. [9] (Experimental study)	Figures 7 and 8 show the rhombic fin geometry adopted in the experimental study. A Coriolis mass flowmeter was mounted upstream of the PCHE. The temperatures of sCO_2 at the inlets and outlets of the PCHE were measured by PT100 sensors, whose maximum uncertainty is ± 1.5℃. The pressures at the inlets of the PCHE were measured by pressure transducers with a full-scale	**Purpose:** To investigate flow and heat transfer performance of a PCHE with rhombic fin channels experimentally using sCO_2 as a working fluid (the effects of turbulent intensity and physical property on the PCHE performance were investigated) **Findings**: (1) The overall heat transfer coefficient of the PCHE is roughly linear dependent on the Geometric Mean of Peclet numbers of the cold and hot fluids. (2) The friction coefficient of the sCO_2 in PCHE is closely pertinent to the channel structure and the friction coefficient correlations of the cold and hot fluid are often different considering their different channel structure at the inlet and outlet regions. (3) The iterative method combined with GA (genetic algorithm) is an effective way to obtain the heat transfer correlation of sCO_2 in PCHEs with

	uncertainty of ± 0.1%. The pressure drops of the two sides of the PCHE are measured by the pressure difference transducer with a full-scale uncertainty of ± 0.1%. All the above measured parameters were collected and recorded every two seconds by a programmable logic controller system.	tremendous property variation and limited boundary parameters. (4) The rhombic fin channel can realize the heat transfer rate per unit volume equivalent to the zigzag channel at the expense of relatively low pressure drops.
Wang et al. [10] (Numerical study)	The geometry and channel shapes adopted for the study are shown in Figures 9 and 10, and Table 3. Numerical tool: The commercial software ANSYS FLUENT 16.0 was used to obtain the study results.	**Purpose:** To carry out numerical study on heat transfer and flow characteristics of straight-type PCHEs with rectangular channels of different widths under different mass fluxes, and compare and analyse the straight-type PCHE results with that of semi-circular-type PCHE using supercritical CO_2 and water as working fluids. **Findings:** (1) The heat transfer performances of rectangular channels in PCHEs are inferior to semicircular channel. At the same mass flux, with the increase of rectangular channel width, the average heat transfer coefficients in the cold side decrease first and then increase, and the average heat transfer coefficients in the hot side always increase. The sharp corners of all PCHE channels are the heat transfer deterioration zones. (2) The increase of rectangular channel width can effectively increase the heat transfer rate per unit volume and reduce the pressure drop at the same mass flux. However, the change of heat transfer rate per unit volume and pressure drop tends to be gentle with the width increase. (3) The overall performance evaluations of rectangular channels in the hot side of PCHEs are enhanced compared with the semicircular channel, and the enhancement becomes more and more significant with the increase of mass flux. While, the overall performance evaluations of the rectangular channels in the cold side are not as good as that of the semicircular channel. (4) A new PCHE design recommendation is proposed that the channel width should be appropriately increased if the strength requirements are met in the hot channel design of straight-type PCHEs to reduce flow resistance and improve overall performance evaluation. At the same time, the sharp corners of the PCHE channel cross-sections should be eliminated to reduce the heat transfer deterioration zone.
Wang et al. [11] (Numerical study)	The geometry and channel shapes adopted for the study are shown in Figures 11, 12 and 13. Numerical tool: A genetic algorithm developed was used to obtain solution for	**Purpose:** To carry out a multi-objective optimal design of the PCHE proposed using three targets (heat transfer rate, pressure drop, and compactness) and six factors (arc height *hl*, maximum arc height position *ll*, airfoil thickness *tl*, airfoil fin horizontal spacing *Lh* and airfoil fin vertical spacing *Lv*, and velocity inlet). The proposed airfoil channel PCHE is

	multi-objective design problem.	to be used as a recuperator. The working fluid for both sides is supercritical CO_2. **Findings:** A comprehensive optimal solution was recommended for the recuperator PCHE with an overall heat transfer rate of 181.29 $W/m^2.K$, a total pressure drop of 26.63 kPa/m, and compactness of 8.5 kW/m^2. And the correspond geometry and arrangement parameters are hl = 2, tl = 12.5, Lh = 3.5 mm, Lv = 1 mm, and $u$cold-hot = 15–30 m/s, for both cold and hot sides.
Li et al. [12] (Numerical study)	The geometry and channel shape (channel cross-section type and channel length type) adopted for the study are shown in Figure 14. Numerical tool: ANSYS FLUENT CFD tool was used to obtain results in the study.	**Purpose:** To study flow and heat transfer characteristics in a semi-circular cross-section Zigzag channel by numerical simulation with supercritical methane as flow media in both cold and hot channels. **Findings:** (1) Under small channel diameter, it has larger turbulent energy, turbulent dissipation rate, poor flow performance, and better heat transfer performance. (2) Turbulent kinetic energy, turbulent dissipation rate and secondary flow velocity increase with the decreasing channel pitch, and fluid temperature decreases with the decreasing channel pitch. There are more times of flow direction reversal, larger heat transfer rate, more significant flow separation, and more energy dissipation under small pitch. (3) As bending angle increases, the velocity gradient between boundary layer and turbulent core area, the turbulence intensity, and the axial component of vorticity increase. (4) By coupling flow and heat transfer characteristics, the optimization analysis of structural parameters was carried out based on the Response Surface Methodology (RSM) and using thermal performance factor (TPF). For supercritical methane flow in a Zigzag-channel PCHE, the optimum channel diameter, pitch and turning angle are obtained, namely, 1.427 mm, 24.6 mm and 15°, with respect to conditions adopted in this study.
Zhu et al. [13] (Numerical study)	The semi-circular cross-section straight channel PCHE geometry designed is shown in Figure 15. Numerical tool: Pressure Vessel code	**Purpose:** To design a 2 MW ZrC/W-based molten-salt-to-sCO_2 PCHE for concentrated solar power. The geometry and dimensions, power density, and pressure drop were obtained through thermal and mechanical design analyses. **Findings:** (1) The PCHE plate was 0.5 m wide, 1.7 m long, and 1.62 mm thick. The PCHE consisted of 308 plates with 65,912 semi-circular channels in total. The volumetric power density was 8 MW/m^3 and the effectiveness was 0.90. (2) Processing cost contributed to a large part of the total cost for nickel alloy 316SS- and IN740H- based PCHEs whereas material cost is the dominating factor for ZrC/W-based PCHEs. (3) The total cost of the ZrC/W-based PCHE was 59% and 37% lower than the nickel alloy IN740H- and 3116SS- based PCHEs.

Xu et al. [14] (Numerical study)	The semi-circular cross-section straight channel PCHE geometry adopted is shown in Figure 16. Numerical tool: Modelica is used for numerical evaluation of the PCHE.	**Purpose:** To optimize the performance of straight-channel PCHE for supercritical CO_2 Brayton cycle. The Pareto front was obtained by the Multi-objective optimization procedure. **Findings:** (1) The variation of heat transfer coefficients along flow direction is different between hot fluid and cold fluid. (2) The pumping power drops by nearly 39% when compared to the reference case with the heat transfer rate and the PCHE volume kept constant. (3) The heat transfer rate of the optimized proposed PCHE size was found up to 4% when the pumping power and the PCHE volume were kept constant. (4) In high temperature recuperator, compared with the reference case, the heat transfer rate increased by 0.2%, the pumping power reduced by 33.1%, and the comprehensive performance can be increased by 5.3% while the volume remains unchanged for the optimal size of the PCHE.
Cong et al. [15] (Numerical study)	Semi-circular cross-section straight channel PCHE geometry was adopted for the study. Numerical code: Commercial CFD code Fluent was used to obtain the results in the study.	**Purpose:** To carry out thermal-hydraulic performance analysis of a PCHE with sodium and sCO_2 as working fluids. **Findings:** (1) The overall and localized Nusselt number increase with the Reynolds number and agree well with Nikitin correlation. (2) The overall friction factor agrees well with the Blasius and Colebrook correlations. (3) The local friction factor decreases along the flow direction in the developing regime in the channel due to the flow development and variations in physical properties. (4) The inlet temperature has significant influences on the local heat transfer characteristics in the entrance region, but no effect after the flow is fully developed.
Wen et al. [16] (Numerical study)	The geometry and channel shape (channel cross-section type and channel length type) adopted for the study are shown in Figure 17. Numerical tool: ANSYS FLUENT 15.0 CFD tool was used to obtain results in the study.	**Purpose:** To study heat transfer performance and flow characteristics of a sinusoidal wavy channel PCHE, which is applicable to thermal efficiency improvement of supercritical CO_2 Brayton cycle. In this study, effects of amplitude and wavelength of sinusoidal wavy channel PCHE were investigated. Discussions were made based on the obtained heat transfer rate, wall heat flux and temperature curves. Optimization study was also performed and optimum wavy parameters were obtained based on best heat transfer performance and overall performance criteria. **Findings:** (1) Increase of amplitude and decrease of wavelength result in larger flow length and heat transfer area of the wavy channel, as well as higher heat transfer rate. (2) The use of wavy channel instead of straight channel enhances heat transfer. (3) Because of the complicated flow fields in the wavy channels affected by thermophysical property changes and centrifugal forces, the shapes and

		locations of high heat flux zones shift with different configurations. (4) The variation in the secondary flow and heat transfer area results in the variation of heat transfer characteristics. (5) The best heat transfer performance is obtained with an amplitude of 3 mm and a wavelength of 50 mm or 75 mm.
Tang et al. [17] (Numerical study)	The PCHE rolling motion geometry with airfoil fins adopted is shown in Figures 18 and 19. Numerical tool: ANSYS FLUENT 19.1 used to obtain the results in the study.	**Purpose:** To investigate thermal-hydraulic performance in a PCHE with airfoil fins for supercritical liquefied natural gas (LNG) near the pseudo-critical temperature under the rolling condition. Comparisons of effects of operating pressures and rolling parameters were made in the analysis. **Findings:** (1) The PCHE has better thermal performance (effect of Nusselt number *Nu*) but worse hydraulic performance (effect of friction factor *f*) at a lower pressure condition. (2) The instantaneous Nusselt numbers and the instantaneous Darcy friction factors present quasi-sine patterns against time, with a period same as the rolling period. (3) The rolling condition gives larger time-averaged Nusselt numbers and Darcy friction factors, compared with the static condition, and the thermal and hydraulic performance increases with increasing rolling amplitude and rolling frequency. (4) The rolling amplitude has a prevailing effect on the thermal and hydraulic performance, in comparison to the rolling frequency. (5) The rolling motion can enhance the heat transfer, while the hydraulic performance under the rolling condition is worse than that under the static condition. (6) With increasing the rolling amplitude and the rolling frequency, the amplitudes of the fluctuations of instantaneous Nusselt number and instantaneous *f*-factor increase. Thus, the crest point increases and the trough point decreases.
Bennett and Chen [18] (Numerical study)	The semi-circular cross-section zigzag channel PCHE geometry adopted is shown in Figure 20. Numerical tool: ANSYS® CFX 17.2	**Purpose:** To develop and evaluate thermal-hydraulic correlations for zigzag-channel PCHEs. **Findings:** Thermal-hydraulic correlations for zigzag-channel PCHEs covering a broad range of design parameters have been developed and evaluated for the estimation of thermal and hydraulic performance.
Kwon et al. [19] (Numerical study)	The semi-circular cross-section straight channel PCHE geometry is adopted for the study. Numerical tool: Existing LMTD method code improved/modified and used to obtained the results in the study.	**Purpose:** To develop a PCHE off-design performance model. The developed model is to be used for both recuperator and pre-cooler in a supercritical CO_2 (s-CO_2) Brayton cycle to optimize power system operation strategies under off-design quasi-steady state conditions (a quasi-steady state cycle analysis was performed). **Findings:** (1) A quasi-steady state cycle analysis was performed to compare the developed off-design performance model with the reference method under two different off-design condition.

		(2) The results show that the obtained optimum operation strategies are still the same for the reference code and the developed model.
Ma et al. [20] (Numerical study)	The semi-circular cross-section straight channel PCHE geometry adopted is shown in Figure 21. Numerical tool: CFD software FLUENT 17 used to obtain the study results.	**Purpose**: To study the effects of dynamic response characteristics of thermodynamic parameters (fluid outlet temperature, total surface heat flux and surface heat transfer coefficient of fluid channels) on the PCHE of 1000 MW sCO_2 coal-fired power plants. **Findings**: (1) The fluid outlet temperature is positively correlated with the fluid inlet temperature and the hot fluid mass flow rate. But it is negatively correlated with the cold fluid mass flow rate. (2) The total surface heat flux and the surface heat transfer coefficient of the fluid channel are positively correlated with the hot fluid inlet temperature and the fluid mass flow rate. (3) The total surface heat flux and the surface heat transfer coefficient of the fluid channel are negatively correlated with the cold fluid inlet temperature. (4) When the fluid inlet temperature is changed, the equilibration time is reduced with the increase of the fluid mass flow rate. Compared to the hot fluid inlet temperature changed, it takes less time to reach the equilibration when the cold fluid inlet temperature is changed. (5) When the hot fluid mass flow rate is changed, higher fluid temperature leads to a longer equilibration time. The time to reach the equilibrium is longer near the design condition when the cold fluid mass flow rate is changed.
Yang et al. [21] (Numerical study)	The semi-circular cross-section wavy channel PCHE geometry adopted is shown in Figure 22. Numerical tool: Commercial software ANSYS Fluent.	**Purpose:** (1) To investigate the heat transfer performance of a wavy channel PCHE. (2) To investigate the effects of narrowing the channel cross section on the thermal-hydraulic characteristics of the wavy channel. **Findings:** (1) The heat flux distribution around the channel surface varies periodically and there exist three banded regions with low heat flux for each period, one on the arched surface and two on bottom surface. (2) Narrowing the channel cross section will increase the heat transfer rate per unit volume, making the PCHEs more compact and economical. (3) The results of performance evaluation for different channel cross sections reveal that PCHEs with narrowed cross sections provide better performance than the prototype PCHE under the objectives of increasing heat-transfer capability and reducing volume, but suffer from the penalty of increasing pumping power. (4) The Nusselt number and the frictional factor decrease when the channel cross-sectional area approaches the minimum value while the hydraulic diameter remains unchanged.

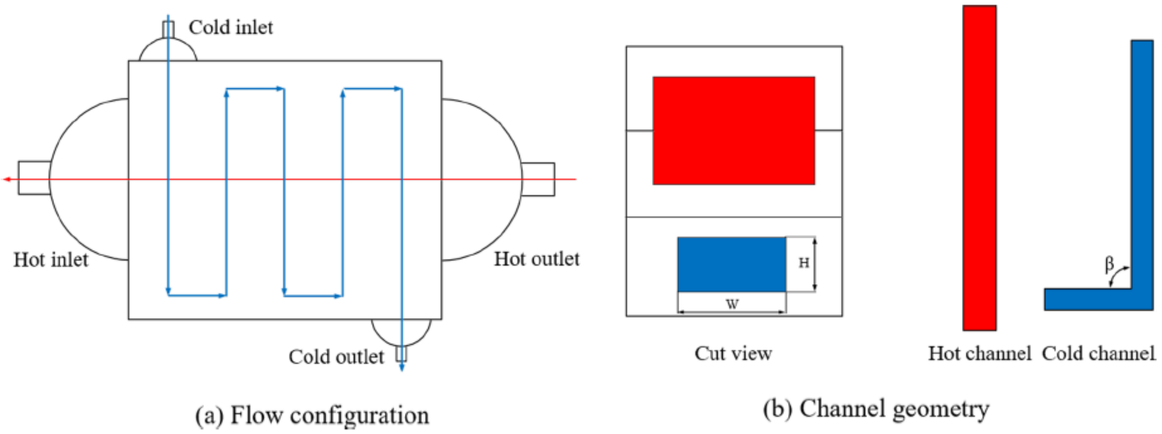


Figure 1. Flow configuration and channel geometry of designed PCHE prototype [7].

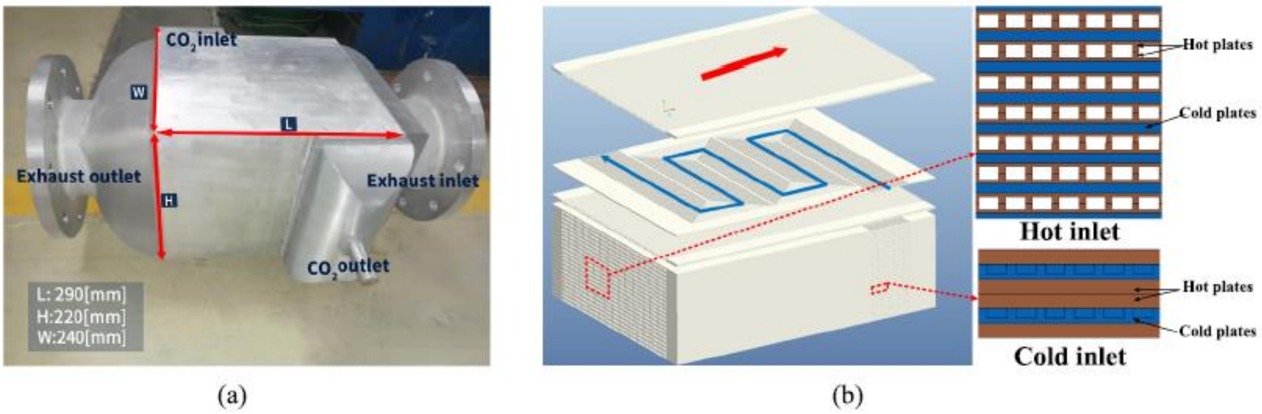


Figure 2. The photograph and geometric structure of PCHE prototype [7].

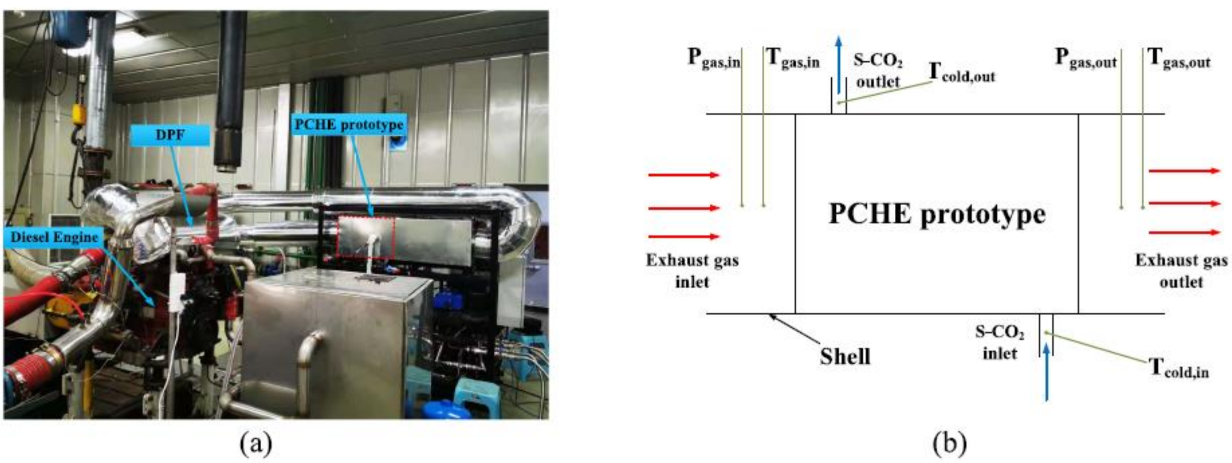


Figure 3. The experimental bench of PCHE prototype (a) and distribution of measurement points of PCHE prototype (b) [7]

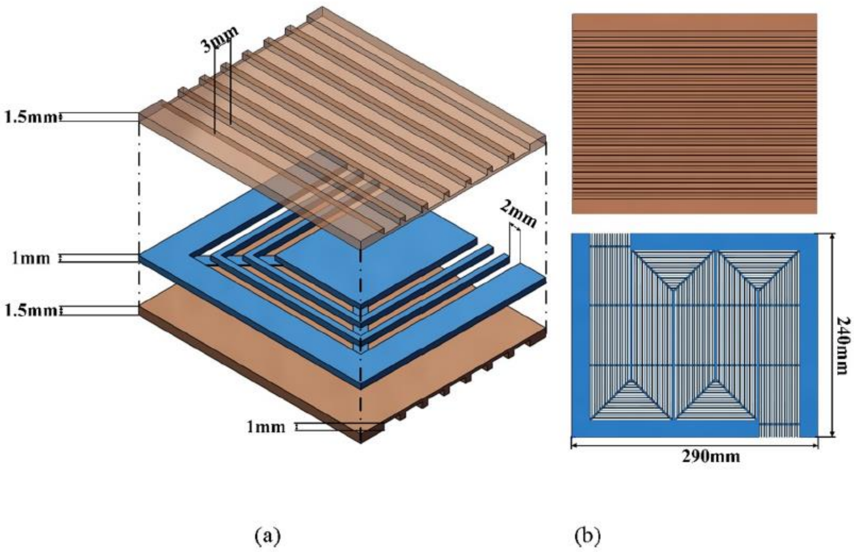


Figure 4. Improved structure: rectangular and hollow channel plate structure [7]

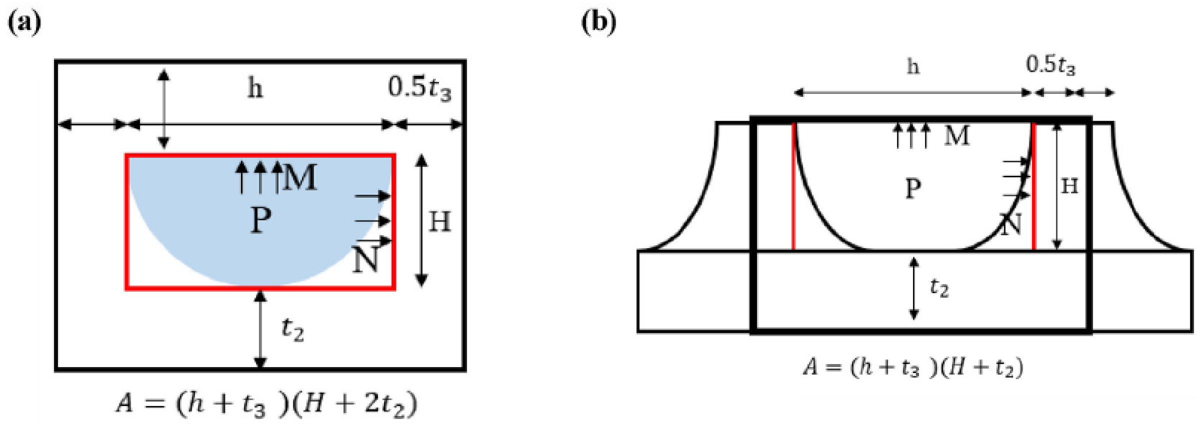


Figure 5. Geometry of PCHE: (a) Straight, zigzag, and wavy types; (b) airfoil type [8]

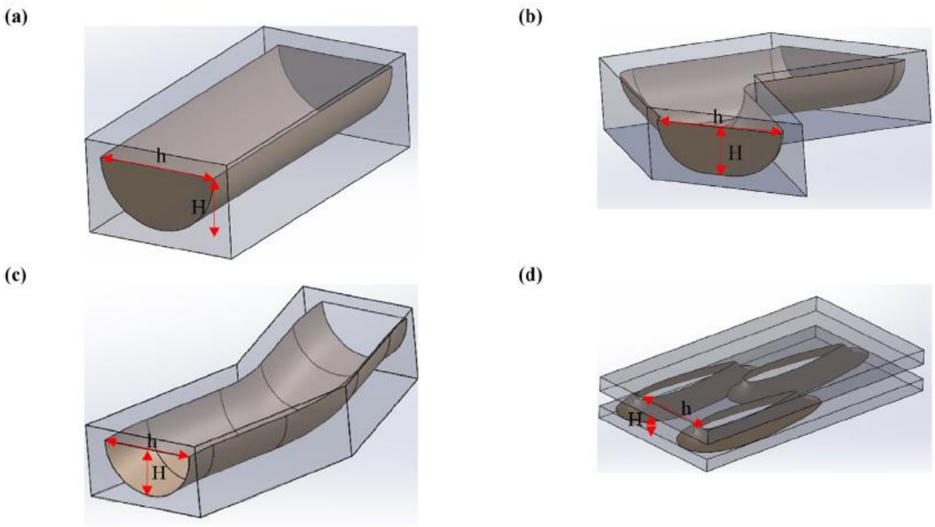


Figure 6. Three-dimensional channel types: (a) Straight; (b) zigzag; (c) wavy; (d) airfoil [8]

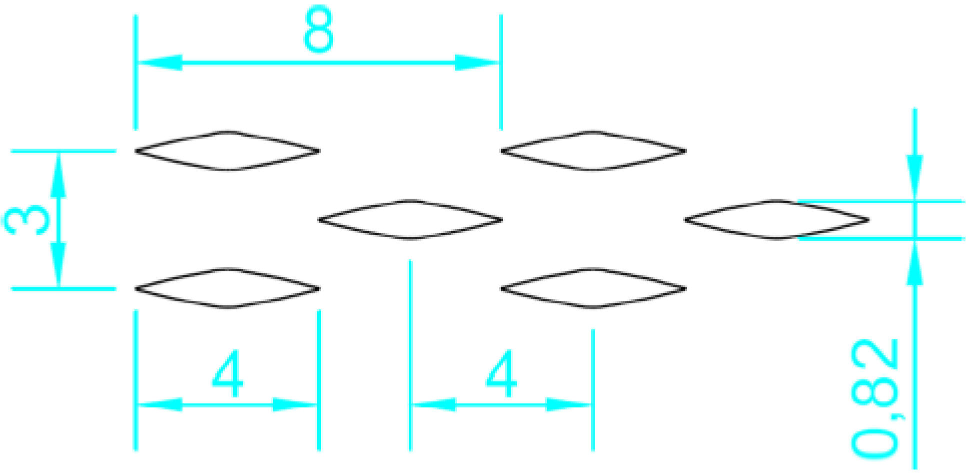


Figure 7. Geometric size of rhombic fin arrangement [9]

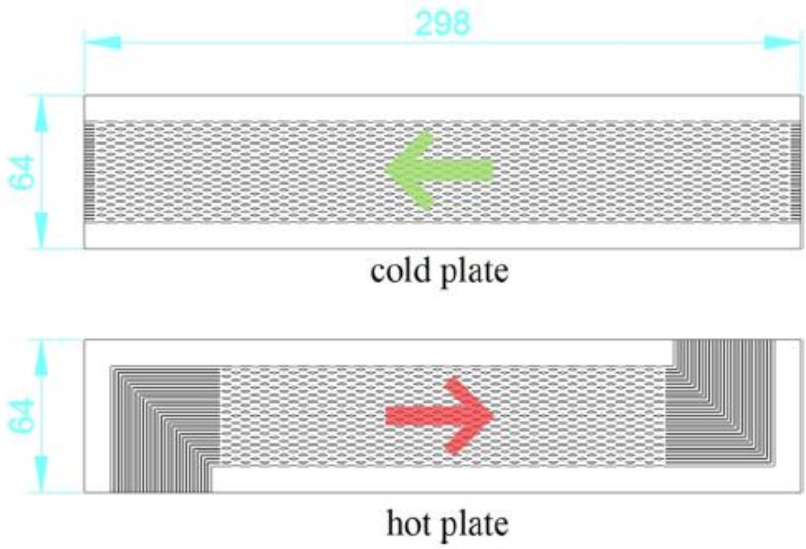


Figure 8. Geometry of the rhombic fin channels [9]

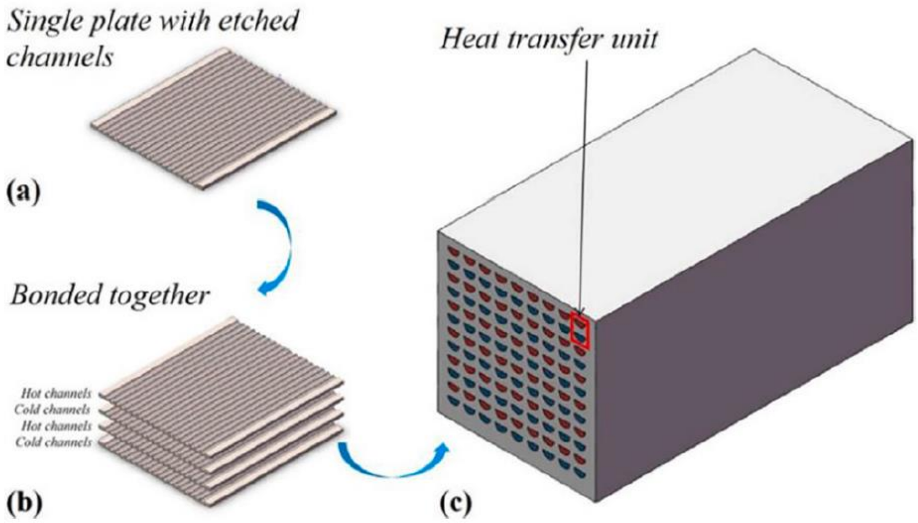


Figure 9. Heat transfer unit structure of PCHE with straight channels [10], [22]

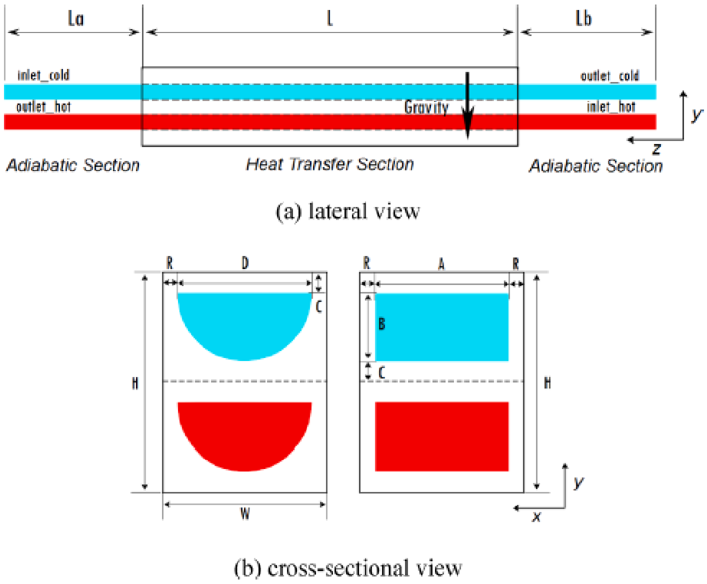


Figure 10. Physical model geometry of calculation unit [10]

Table 3. Geometric data of physical models (Wang et al. [10]).

Parameters	Physical model number				
	Model a	Model b	Model c	Model d	Model e
Cross-section shape	semicircle	rectangle	rectangle	rectangle	rectangle
A	-	1.6 mm	3.2 mm	4.8 mm	6.4 mm
B	-	0.8 mm	0.8 mm	0.8 mm	0.8 mm
D	1.6 mm	-	-	-	-
W	2.4 mm	-	-	-	-
C	0.4 mm				
H	3.2 mm				
L	200 mm				
La	50 mm				
Lb	50 mm				
R	0.4 mm				

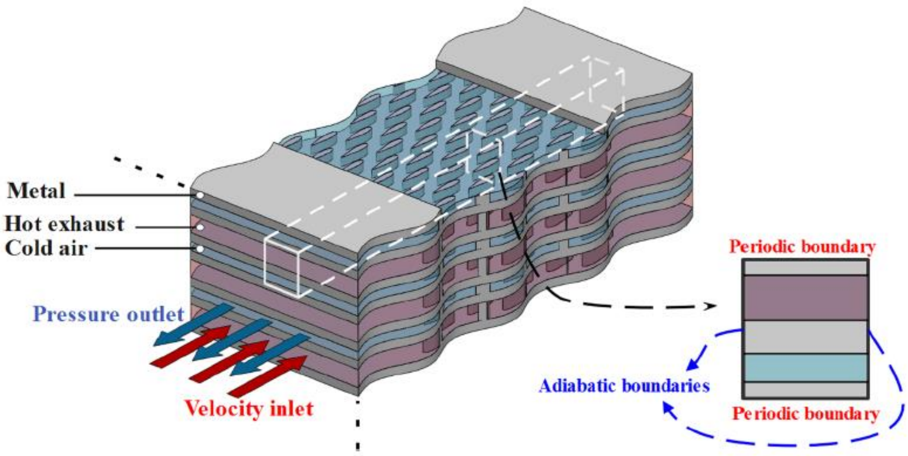


Figure 11. Computational domain and boundary conditions of PCHE [11]

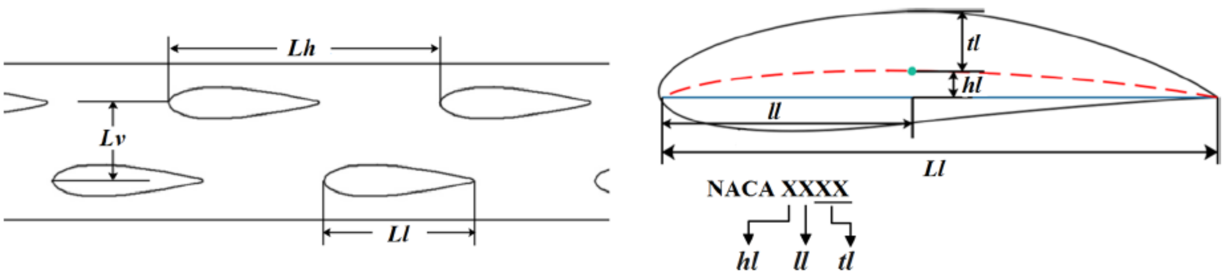


Figure 12. Structure and geometry parameters of PCHE channel with NACA (National Advisory Committee for Aeronautics) airfoil fins [Airfoil fin structure parameters (arc height hl, maximum arc height position ll, airfoil thickness tl), Airfoil fin arrangement parameters (airfoil fin horizontal spacing Lh and airfoil fin vertical spacing Lv), Airfoil chord Ll] [11]

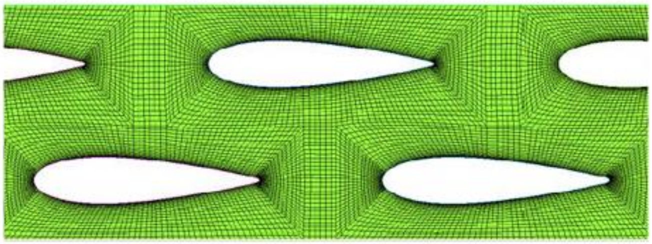
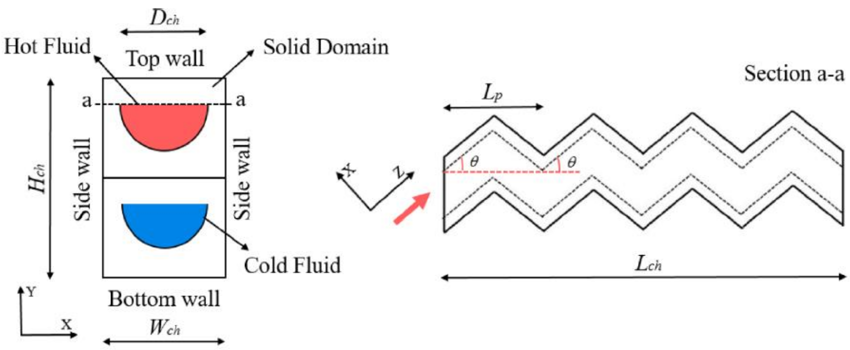
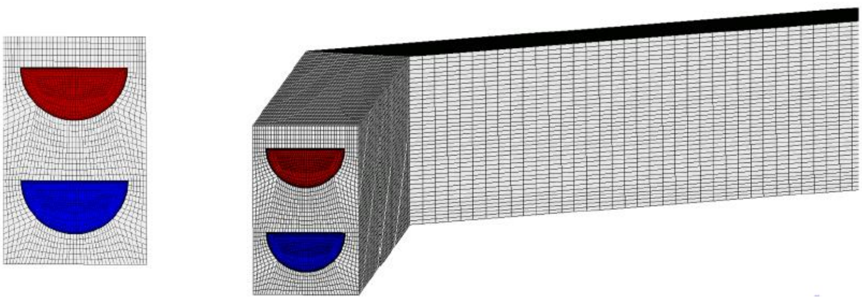


Figure 13. Computational grids of AF-PCHE channel [11]



(a) Geometrical model



(b) Meshing model of the geometrical model

Figure 14. Geometrical and Meshing models of the geometry adopted [12]

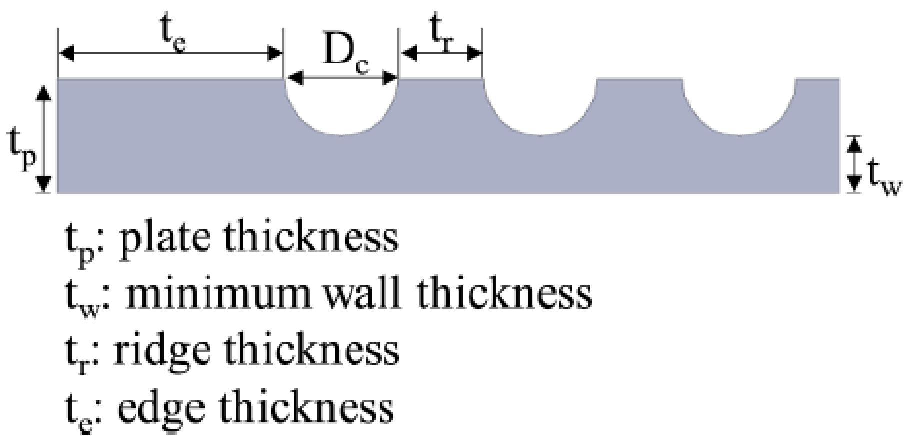


Figure 15. Schematic of PCHE plate [13]

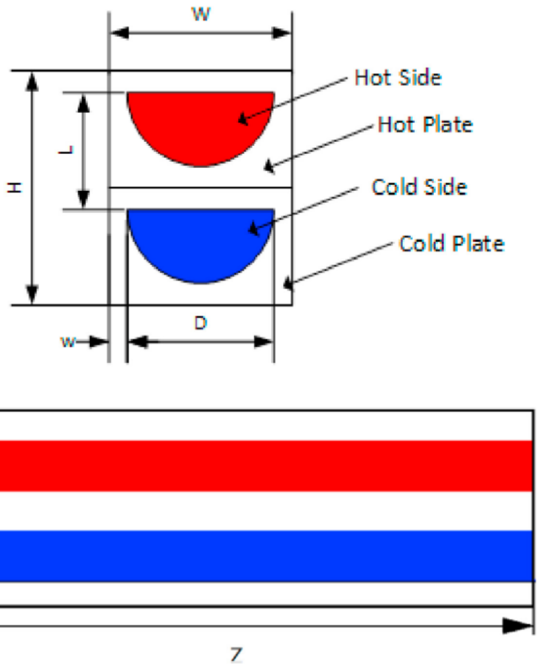


Figure 16. Geometry and dimensions of PCHE [14]

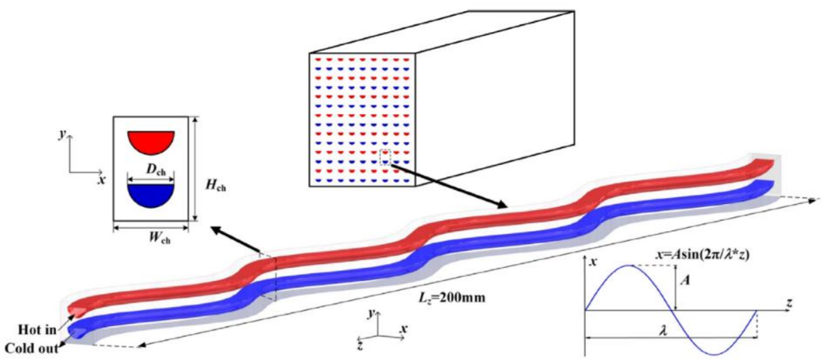


Figure 17. Geometry of the wavy channel [16]

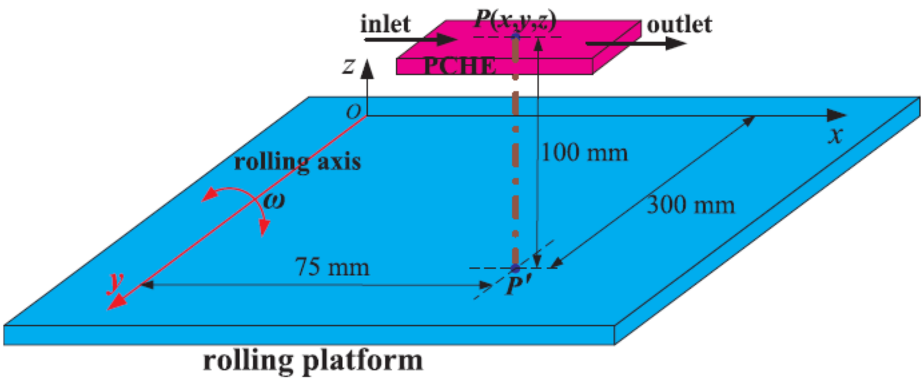


Figure 18. Schematic diagram of the rolling working condition [17]

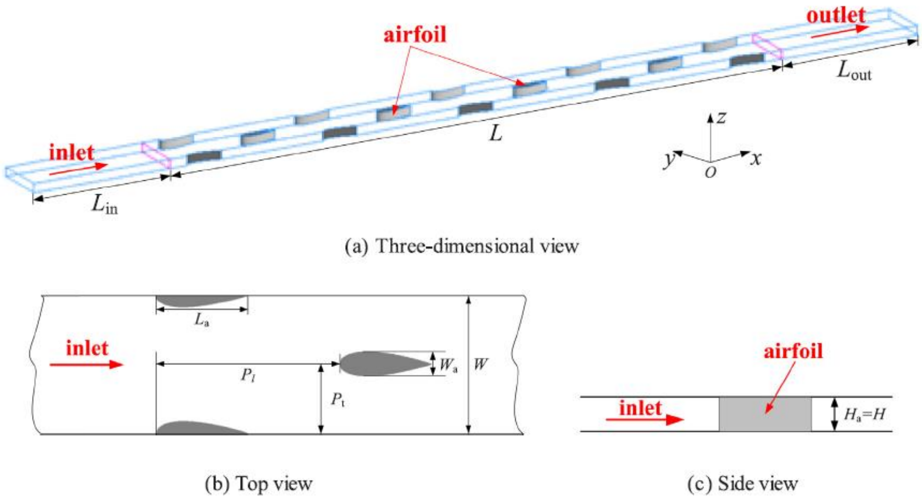


Figure 19. Schematic diagram of the physical model [17]

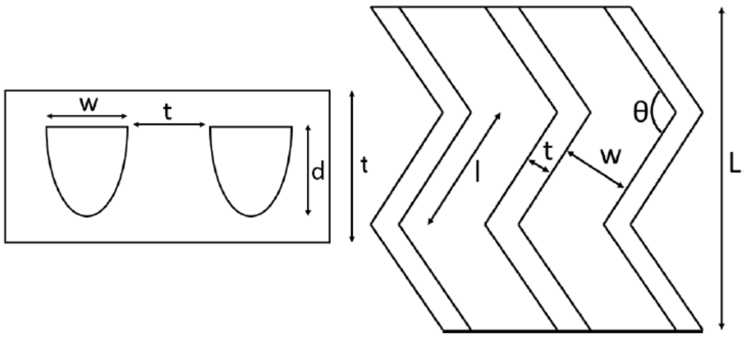


Figure 20. Design geometry [18]

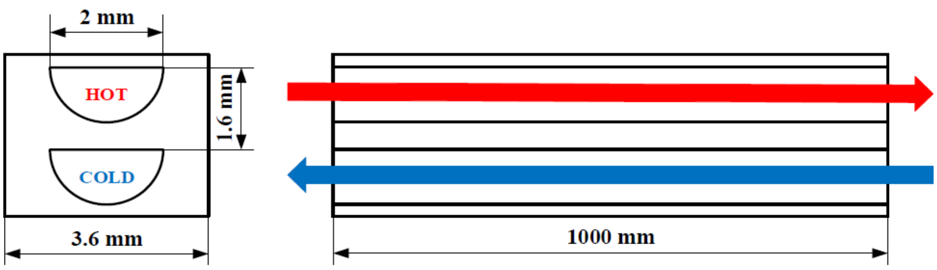


Figure 21. Physical model and geometric dimensions of a single-channel PCHE [20]

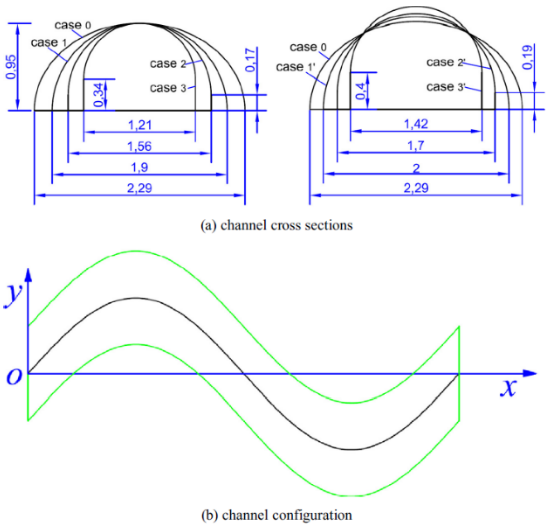


Figure 22. Shapes and sizes of the PCHE channel (unit: mm) [21]

2. Design and Performance Optimization of Heat Exchangers

Simulation and optimization codes (or numerical codes/tools) are used to guide the design of heat exchangers and other similar heat transfer devices/systems used for industrial applications. These numerical tools are used as predictive tools/codes for predicting the effects of the design and design improvements on the performance of heat transfer systems, and therefore play vital role in the design optimization of these heat transfer systems. These numerical tools are also used to support experimental studies by building on the basis provided by experimental results and carrying out further investigations that are complicated and costly to be fully carried out through experimental investigations. Thus, simulation and optimization codes/tools play vital role in research and development activities involving the design and operation of fluid flow and heat transfer systems (Martelli et al. [23], Debrah et al. [24], Shitsi et al. [25]).

Optimization involves selection of a parameter or a function to be optimized by decreasing (minimizing) or increasing (maximizing) the quantity (value) of the selected parameter or function. This function, which shows various aspects or features (parameters or characteristics) that are important to be considered in the optimization, is normally referred to as Objective Function. Parameters such as design parameters (geometry and material parameters including weight, volume, etc.); system input parameters or operational parameters (mass flow rate, fluid inlet temperature, pressure, heat flux, etc.); costs (capital, initial, maintenance and product costs); system performance parameters (heat exchanger effectiveness, compactness, LMTD, overall heat transfer coefficient, rate of energy consumption, efficiency, overall profit, safety, environmental effects, etc.); and system output parameters (heat transfer coefficient or heat transfer rate, wall temperature, fluid outlet temperature, pressure drop, etc.) are mostly optimized in fluid flow thermal systems as an objective function (Martelli et al. [23], Navaie [26]). The system output parameters and system performance parameters are discussed based on the design, costs and system input (operational) parameters to help determine the optimized values for these design, costs and operational parameters without comprising system safety.

Heat exchangers form integral components of thermal systems (fluid flow and heat transfer systems) intended for particular use/application. Various optimization studies are carried out prior to the design of heat exchangers and other similar heat transfer devices/systems in order to have information (or data) on the optimized design parameters, input (operational) parameters, output parameters, performance parameters and estimate costs involved in the design and operation of heat exchangers. According to Webb [27], and Gee and Webb [28], design and optimization has three basic conditions of design objectives and constraints such as increasing heat transfer capability while keeping volume and pumping power constant, reducing pumping power while keeping heat transfer capability and volume constant, and reducing volume while keeping heat transfer capability and pumping power constant. The design objectives and constraints normally vary with the application condition. The performance of PCHE is evaluated from the perspective of heat transfer rate per unit volume and pressure (Yang et al. [21], Aneesh et al. [29]).

Geometry design (or Mechanical design) studies intended to optimize the performance of PCHE involves investigating the effects of geometrical parameters such as cross-sectional shape of the channel (circular, semi-circular, square, rectangular, airfoil, rhombic, etc.) and channel type (straight, zigzag, wavy, airfoil, S-shaped, etc.) on the overall performance of PCHE. Kern and Log Mean Temperature Difference (LMTD) methods which are normally used in heat exchanger design could be used to develop PCHE code for evaluation of thermal-hydraulic performance and economics of PCHEs (Lee et al. [8, 30]). Log Mean Temperature Difference (LMTD) and ε-NTU methods are the two (2) conventional methods used in modelling of heat exchangers to predict system performance in the view of heat transfer but modifications to these methods are required to predict thermal performance if PCHEs are to be used for pre-coolers and recuperators near critical conditions and at supercritical conditions (Kwon et al. [19]).

The estimate cost evaluation of PCHE takes into consideration the material for the design, volume occupied by the material, pumping work and fluid pressure drop. The total cost estimation of the design and operation of the PCHE is the sum of the capital cost and operating cost. The capital cost is determined based on the volume and material for the design, excluding the fabrication and interfacing costs, which depend on the manufacturer and seller of the PCHE (Lee et al. [8], Thulukkanam [31]).

The total material needed for the design is given by equation (1). The total cost of the material is given by equation (2), Cm is the unit cost of the material. The capital cost is determined using equation (3), *r* is the interest rate (in percentage %) and *N* is the duration of operation of the PCHE (in years) (Kim and Sherman [32]).

$$M = \rho * V\ [kg] \qquad (1)$$

$$C = C_M * M\ [\$] \qquad (2)$$

$$C\ capital = \frac{C * r * (1+r)^N}{[(1+r)^N - 1]}\ [\$] \qquad (3)$$

The pumping work *Wp* which is determined using the working fluid volumetric flow rate *V* and working fluid pressure drop Δ*P* in the PCHE is given by equation (4). The operating cost is determined using equation (5), O is the operating cost per pumping work, which could be more than 36.4 $/MWh. The total cost estimation of the design and operation of the PCHE is determined using equation (6) (Kim and No [33]).

$$Wp = \Delta P * V\ [MW] \qquad (4)$$

$$C\ operating = O * (Wp * 3600s) = O * (Wp * 1h)\ [\$] \qquad (5)$$

$$C\ total = C\ capital + C\ operating\ [\$] \qquad (6)$$

The correlations used for calculation of the Nusselt number and friction factor are important for determination of the performance of PCHE in relation to heat transfer rate (heat transfer coefficient) and pressure drop. That is, the Nusselt number and friction factor correlations are used in the thermal hydraulic design of heat transfer systems. The correlations used for the calculation of Nusselt number and friction factor corresponding to the different channels of PCHE are given in Table 4. These heat transfer and friction factor correlations could be applicable to fluids such as H2O, CO_2, He, CH4 and N2 among others. Huang et al. [5] provides heat transfer and friction factor correlations specific to CO_2 and He with respect to the PCHE channel types such as airfoil, zigzag, straight and S-shaped channels.

Table 4. Correlations for heat transfer (Nusselt number) and friction factor calculation

Correlations for Nusselt number (heat transfer) and friction factor calculation for different channel types
Straight channel (Lee et al. [8], Kim et al. [34, 35], Lu et al. [36], Bannett and Chen [18]) $Nu = \frac{(F/8)(Re-1000)Pr}{1+12.7(F/8)^{1/2}(Pr^{2/3}-1)}\ where\ F = (0.79ln(Re) - 1.64)^{-2}, \frac{1}{\sqrt{f}} = -1.8log\left(\left(\frac{\varepsilon}{3.7D}\right)^{1.11} + \frac{6.9}{Re}\right)$ Straight channel (Yang et al. [9], Liu et al. [2]) $Nu = 0.1229Re^{0.6021}Pr^{0.3}\left(\frac{Cp_w}{Cp_b}\right)^{0.1310}, f = 0.05776Re^{-2192} \quad 3600 \le Re \le 36500$ Straight channel (Gnielinski [37], Saeed et al. [38]) $Nu = \frac{(f/8)(Re-1000)Pr}{1+12.7\left(\frac{f}{8}\right)^{0.5} * \left(Pr^{\frac{2}{3}}-1\right)}, f = \left(\frac{1}{18log_{10}Re - 1.5}\right)^2, (3000 \le Re \le 60000)(0.7 \le Pr \le 1.2)$
Zigzag channel (Lee et al. [8], Kim et al. [34, 35], Lu et al. [36], Bannett and Chen [18]) $Nu = 2.2631 * 10^6(Re)^{1.0515*10^{-3}}m^{6.5405*10^{-1}}\alpha^{-2.0665*10^{-4}}\left(\frac{L}{D}\right)^{8.0885*10^{-5}}$ $Nu = 2.2967 * 10^6 m^{6.5429*10^{-1}}T^{-1.3319*10^{-3}}, \quad f = 1.288 * 10^{-1} - 3.231 * 10^{-3}Re^{3.58*10^{-1}} + 8.929 * 10^{-2}(\pi - \alpha)^{1.799} +$ $1.636 * 10^{-2}\left(\frac{r}{D}\right) + 1.349 * 10^{-1}\left(\frac{w}{D}\right)^{2.972*10}$

Zigzag channel (Yang et al. [9], Ngo et al. [39]) $Zigzag: Nu = 0.1696Re^{0.629}Pr^{.371} \quad f = 0.192Re^{-0.091} \quad 3500 \leq Re \leq 23000, 0.7 \leq Pr \leq 2.2$ Zigzag channel (52°C) (Ngo et al. [39]); Zigzag channel (40°C) (Ishizuka et al. [40], Saeed and Kim [41], Saeed et al. [38]) $h = 0.2104Re + 44.16, f = -2 \times 10^{-6}Re + 0.1023, (5000 \leq Re \leq 13000)(0.7 \leq Pr \leq 1.2);$ $Nu = 0.169Re^{0.629}Pr^{0.317}, f = 0.1924Re^{-0.091}, (3500 \leq Re \leq 22000)(0.7 \leq Pr \leq 2.2);$ $Nu = 0.041Re^{0.83}Pr^{0.95},$ $Hot\ side; f = 0.115Re^{-0.13}, Cold\ side; f = 0.115Re^{-0.089}, (3500 \leq Re \leq 22000)(0.7 \leq Pr \leq 2.2)$
Wavy channel (Lee et al. [8], Kim et al. [34, 35], Lu et al. [36], Bannett and Chen [18]) $Nu = 0.4Re^{0.64}Pr^{1/3}\left(\frac{2w}{\Lambda}\right)^{0.75}, f = 4.8Re^{-0.36}\left(\frac{2w}{\Lambda}\right)^{1.5}$
Airfoil channel (Lee et al. [8]; Kim et al. [34, 35], Lu et al. [36], Bannett and Chen [18]) $Nu = 0.02671Re^{0.8}Pr^{n}\xi_h{}^{-0.009177}\xi_v{}^{-0.01118}\ (n = 0.3\ for\ cooling, n = 0.4\ for\ heating),$ $f = 0.05754Re^{-0.1923}\xi_h{}^{-0.22647}\xi_v{}^{-0.03108}$ Airfoil channel (Yoon et al. [42]; Saeed et al. [38]) $Nu = 0.027Re_{min}^{0.78}Pr^{0.4}, fRe_{min} = 9.31 + 0.028Re_{min}^{0.86}, (3000 \leq Re \leq 1.5 \times 10^5)(0.6 \leq Pr \leq 0.8)$
S-shaped channel (Ngo et al., 2007) $S - shaped: Nu = 0.174Re^{0.539}Pr^{0.43}\ f = 04545Re^{-0.34} \quad 3500 \leq Re \leq 23000, 0.7 \leq Pr \leq 2.2$
C-shaped (40°C) (Saeed and Kim [41], Saeed et al. [38]) $Nu = 0.050Re^{0.8}Pr^{0.86},$ $Hot\ side; f = 0.019Re^{-0.0054}, Cold\ side; f = 0.025Re^{0.038}, (3000 \leq Re \leq 60000)(0.7 \leq Pr \leq 1.2)$
Rhombic fin channel (Yang et al. [9]) $Nu = 0.02661Pe^{0.8145}\left(\frac{\rho_w}{\rho_b}\right)\left(\frac{Cp_w}{Cp_b}\right)^{0.6421}\left(\frac{\mu_w}{\mu_b}\right)^{0.2765}\left(\frac{\lambda_w}{\lambda_b}\right)^{-0.8897}$ $f_h = 1.67363 - 1.47907\left(1 - e^{-Re/2282.74698}\right) - 1.29913 \times 10^6\left(1 - e^{-Re/3.67732\times10^{11}}\right)$ $f_c = 3.84694 - 3.59004\left(1 - e^{-Re/1200.44046}\right) - 5.61481 \times 10^6\left(1 - e^{-Re/3.49261\times10^{11}}\right)$

There are other formulations/correlations for design and optimization of heat exchangers in addition to heat transfer and friction factor correlations. These formulations could be different depending on the flow type in the heat exchanger (parallel flow, counter-flow or countercurrent flow, cross-flow, multi-flow), channel type (straight, zigzag, wavy, airfoil, etc.) or the flow cross-section of the channel type (circular, square, rectangular, semi-circular, airfoil, etc). These formulations include heat transfer effectiveness ε, number of transfer units *NTU*, log (logarithmic) mean temperature difference *LMTD*, overall heat transfer coefficient *U*, the net heat transfer rate Q_{net}, etc. Most of these formulations were obtained for tubular (circular) cross-section straight channel heat exchangers. PCHEs are basically counter-current fluid flow designs and therefore, the design and optimization formulations provided in this section are formulations based on counter-current flow design of heat exchangers.

The heat capacity rate for hot fluid is defined by equation (7).

$$C_h = \dot{m}_h C_{p.h} \quad [W/K] \tag{7}$$

The heat capacity rate for cold fluid is defined by equation (8).

$$C_c = \dot{m}_c C_{p.h} \quad [W/K] \tag{8}$$

The heat capacity ratio, which is the ratio of minimum heat capacity rate to the maximum heat capacity rate, is defined by equation (9).

$$C_r = \frac{C_{min}}{C_{max}} \tag{9}$$

C_{min} can be C_h or C_c, or C_{max} can be C_h or C_c. That is if C_{min} is C_h, then C_{max} will be C_c; if C_{min} is C_c, then C_{max} will be C_h.

Heat exchanger effectiveness is defined by equation (10).

$$\text{Effectiveness } \varepsilon = \frac{\text{Actual heat transfer rate } q_{act}}{\text{Maxumum possible heat transfer rate } q_{max}} \tag{10}$$

Where q_{act} and q_{max} are defined by equations (11, 12).

$$q_{act} = C_h(T_{h.i} - T_{h.o}) = C_c(T_{c.o} - T_{c.i}) \quad (11)$$

$$q_{max} = C_{min}(T_{h.i} - T_{c.i}) \quad (12)$$

The actual heat transfer in terms of heat exchanger effectiveness becomes equation (13).

$$q_{act} = \varepsilon\, C_{min}(T_{h.i} - T_{c.i}) \quad (13)$$

In a heat exchanger, an overall heat transfer coefficient U is often used to combine the effects of convection and conduction of the heat exchanger core (Kyekyeku [43], Cengel [44]). For a semi-circular flow channel, the overall heat transfer coefficient based on the inner surface areas of the tube/channel per length L without the effects of fouling (fouling factors) is given by equation (14).

$$\mathrm{U_i} = \frac{1}{\frac{1}{\mathrm{h_i}} + \frac{\mathrm{A_i}\ln\left({}^{\mathrm{r_o}}/_{\mathrm{r_i}}\right)}{\pi \mathrm{K_f L}} + \frac{\mathrm{A_i}}{\mathrm{A_o}}\frac{1}{\mathrm{h_o}}} \quad (14)$$

The number of transfer units (NTU) is defined by equation (15).

$$NTU = \frac{UA}{C_{min}} \quad (15)$$

Heat exchanger effectiveness is also defined by equations (16, 17).

$$\varepsilon = \frac{1 - e^{-NTU(1-C_r)}}{1 - C_r e^{-NTU(1-C_r)}}, \quad [C_r < 1] \quad (16)$$

$$\varepsilon = \frac{NTU}{1 + NTU}, \quad [C_r = 1 \; or \; C_h = C_c] \quad (17)$$

NTU is also defined by equations (18, 19).

$$NTU = \frac{1}{C_r - 1} ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right), \quad [C_r < 1] \quad (18)$$

$$NTU = \frac{\varepsilon}{1 - \varepsilon}, \quad [C_r = 1 \; or \; C_h = C_c] \quad (19)$$

The LMTD for counter-current flow heat exchanger is expressed by equation (20) (Cengel, 2002).

$$LMTD = \frac{\theta_2 - \theta_1}{Log_e \frac{\theta_2}{\theta_1}} = \frac{\theta_2 - \theta_1}{ln \frac{\theta_2}{\theta_1}}, \theta_1 = T_{h.i} - T_{c.o}, \theta_2 = T_{h.o} - T_{c.i} \quad (20)$$

The net heat transfer rate Q_{net} of the two fluids can be expressed by equation (21) (Cengel, 2002).

$$\mathrm{Q_{net}} = \mathrm{UA(LMTD)} \quad (21)$$

Wen et al. [16] used equation (22) for calculation of heat exchanger effectiveness and equation (23) for evaluation of overall performance of heat exchanger to carried out performance analysis of a semi-circular cross-section sinusoidal wavy channel PCHE with supercritical CO_2 as working fluid.

$$\varepsilon = \frac{H_{h,in} - H_{h,out}}{H_{h,in} - H\left(T_{h,out} - T_{c,in}\right)} \quad (22)$$

$$\eta = \frac{j/j_0}{(f/f_0)^{1/3}} \quad (23)$$

Where j, Nu, Pr and f are given respectively by equations (24, 25, 26, 27).

$$j = \frac{Nu}{RePr^{1/3}} \quad (24)$$

$$Nu = \frac{hD_h}{k} \; or \; \frac{hd_h}{\lambda_f} \quad (25)$$

$$Pr = \frac{\mu C_p}{k} \quad (26)$$

$$f = \frac{\Delta P D_h}{2u^2 \rho L} \; or \; \frac{\Delta P . d_h}{2\rho L V^2} \quad (27)$$

where j_0 and f_0 correspond to the case of straight channel.

Li et al. [12] used equation (28) for calculation of thermal performance index (TPF) of heat exchanger to carried out performance analysis of a semi-circular cross-section zigzag channel PCHE with supercritical methane as working fluid.

$$TPF = \frac{Nu/Nu_0}{(f/f_0)^{1/3}} \quad (28)$$

Where f, Nu, d_h, and h are respectively given by equations (27, 25, 29, 30)

$$d_h = \frac{\pi D_{ch}}{2(\pi/2 + 1)} \quad (29)$$

$$h = \frac{q}{T_w - T_b} \quad (30)$$

Several design and performance optimization studies have been performed in which straight channel PCHE has been compared with non-straight channel PCHE, and the study results show that the non-straight PCHEs performed better than the straight channel PCHEs in terms of reduced pressure drop (lower frictional resistance), enhancement of heat transfer and reduced volume (Saeed et al. [38], Ma et al. [45], Cheng et al. [46], Ngo et al. [39, 47], Wen et al. [16], Kim et al. [48], Saeed and Kim [41], Chen et al. [49], Pidaparti et al. [50]). Supercritical carbon dioxide (SCO2 or sCO2 or S-CO2) has been applied to PCHEs and analyses based on Brayton power cycle have shown that SCO2 could be applied in the fields of solar energy, nuclear energy, distributed energy and waste heat utilization because of its excellent capabilities such as high efficiency, high flexibility, compact size, and safety among others. SCO2 Brayton power cycle has been found to be effective because of its capability of producing high temperature of heat addition and low temperature of heat rejection (Binotti et al. [51]; Liu et al. [2]; Kouta et al. [52]; Alfani et al. [53]; Kwon et al. [54]; Yang et al. [55]). S-CO_2 Brayton cycle provides relative benefits such as (1) the thermal efficiency of the s-CO_2 cycle is higher than that of the supercritical water-steam Rankine cycle, (2) the carbon dioxide is an inertial fluid that produces a weak chemical reaction with the solid matter, which further increases the temperature of the steam at the turbine inlet, (3) the s-CO_2 cycle is a Brayton cycle. The entire system is operated at high pressures (>7.38 MPa), resulting in high fluid density, which significantly reduces the sizes of the compressor, turbine and cooler components (Ma et al. [20], Holcomb et al. [56], Dostav [57], Lv et al. [58]). That is, SCO_2 is a potential fluid/gas to be used for power production in addition to the use of steam for power production.

3. Optimization of Brayton and Rankine Power Cycles

Brayton and Rankine cycles are respectively used in gas and steam turbines. They are used to describe the performance as well as optimize the performance of gas and steam systems that depends on heating and cooling applications to increase/maximize the outputs of these power generation systems. Figures 23 and 24 respectively show closed-cycle system of components and process diagrams for Brayton cycle. Figures 25 and 26 respectively show cycle system of components and process diagrams for Rankine cycle. The Brayton cycle is more efficient than the Rankine cycle, but the Rankine cycle can generate more power because it uses water as its working fluid (Ighodaro and Osikhuemhe [59]). Table 5 compares Brayton and Rankine cycles.

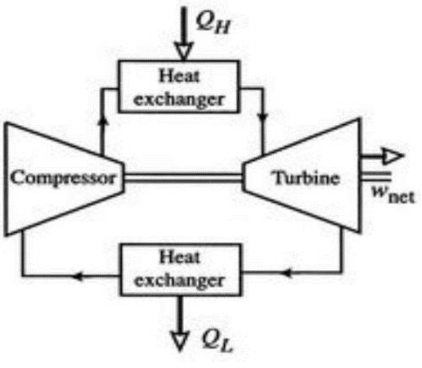


Figure 23. Closed-cycle system (Liu et al. [60]).

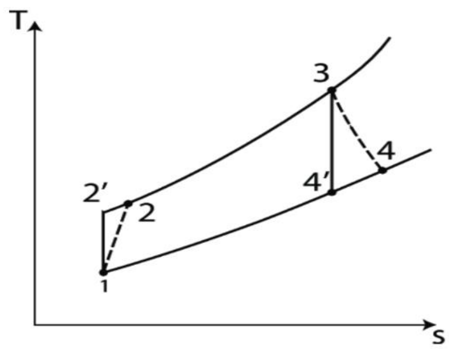


Figure 24. Process diagrams for the Brayton cycle [60]

The efficiency of Brayton cycle or Rankine cycle is given by equation (31).

$$\eta_{BR\ or\ RA} = \frac{Turbine\ Work - Compressor\ Work\ or\ Pump\ Work}{Heat\ added\ in\ the\ Combustor\ or\ in\ the\ Boiler/Evaporator} \quad (31)$$

The ideal efficiency and actual efficiency of Brayton cycle are given by equations (32) and (33) (see Figure 24).

$$\eta_{IDEAL\ BR} = 1 - \frac{T_1}{T_{2'}} \quad (32)$$

$$\eta_{ACTUAL\ BR} = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2} \quad (33)$$

The h_2 and h_4 are optimized enthalpies corresponding to respective temperatures or entropies, $T_2\ (or\ s_2)$ and $T_4\ (or\ T_2)$.

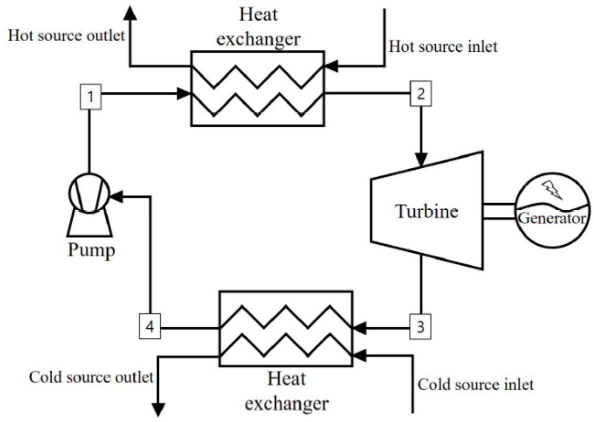


Figure 25. A schematic diagram of a simple Rankine cycle, showing the correct location of the components and the correct direction of energy and mass flows [61], [62]

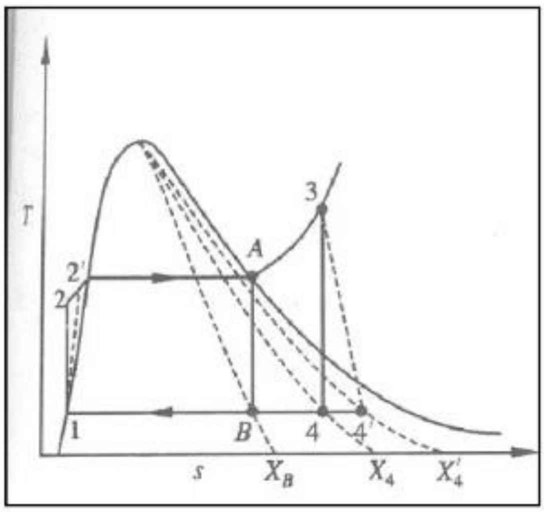


Figure 26. Temperature–entropy process diagram for a basic binary cycle using a working fluid with normal condensing properties [61], [62]

The ideal efficiency and actual efficiency of Rankine cycle are given by equations (34) and (35) (see Figure 26).

$$\eta_{IDEAL\ RA} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2)} \qquad (34)$$

$$\eta_{ACTUAL\ RA} = \frac{(h_3 - h_{4'}) - (h_{2'} - h_1)}{(h_3 - h_{2'})} \qquad (35)$$

The $h_{2'}$ and $h_{4'}$ are optimized enthalpies corresponding to respective entropies or temperatures, $s_{2'}\ (or\ T_{2'})$ and $s_{4'}\ (or\ T_{4'})$

Efficiency can be defined as the ratio of output to input. Efficiency can be increased by increasing the output at constant input or decreasing the input for constant output or increasing the output while reducing the input. Methods of regeneration, reheating, intercooling, recuperation, recompression, and stages of turbine expansion (two-stage turbine expansion) among others could be used to improve the efficiency of Brayton cycle or Rankine cycle. The combination of these methods could also be used to improve the efficiency of Brayton cycle or Rankine cycle.

Organic Rankine cycle (ORC) systems are similar to conventional steam power generation systems but use organic fluids instead of water as working fluid. Potential working fluid for ORC systems include hydrocarbons, hydrofluorocarbons, chlorofluorocarbons, hydrofluoroolefins (HFOs), among others (Zhang et al. [63]; Tchanche et al. [64]; Mago et al. [65]; Wang et al. [66]). However, on the view of safety and environmental influence, HFOs are the better selection since they are nonflammable, nontoxic, and environment friendly with zero-ozone depletion potential values and a low global warming potential value (Zhang et al. [63]; Yang et al. [67]; Fang et al. [68]).

Table 5. Comparison of Brayton and Rankine cycles

BRAYTON CYCLE	**RANKINE CYCLE**
The Brayton cycle is a gas turbine cycle and a thermodynamic cycle, which is used in gas turbines.	The Rankine cycle is a steam turbine cycle and a thermodynamic cycle, which is used in steam turbines.
The working input fluid is a gas and the output fluid from the combustor is also a gas which is fed into the gas turbine for expansion producing shaft work for power generation.	The working input fluid is a liquid and the output fluid from the boiler/evaporator is a steam/vapor which is fed into the steam turbine for expansion producing shaft work for power generation.
The different processes that the working fluid undergoes in closed Brayton cycle are (see Figure 24): **1) Isentropic compression (process 1-2):** Ambient air is drawn inside the compressor and compressed isentropically. **2) Isobaric heat addition (process 2-3):** Heat is added to the compressed air at constant pressure. **3) Isentropic expansion (process 3-4):** Air/gas is expanded in a turbine isentropically. **4) Isobaric heat rejection (process 4-1):** Heat is rejected from the system at constant pressure. Isentropic compression and expansion processes denote an ideal cycle. Usually, the process is not completely isentropic due to irreversibilities and friction losses in turbine and compressor. The isentropic efficiency of turbine and compressor denote the magnitude of useful output that can be obtained from given conditions. The fourth step of the Brayton cycle (cooling of the working fluid) is normally omitted, as gas turbines are open systems that do not reuse the same air/gas.	The different processes that the working fluid undergoes in closed Rankine cycle are (see Figure 26): **1) Isentropic compression (process 1-2):** Pressure of working fluid increases in this process. The low-pressure liquid is pumped to a boiler/eveporator increasing its pressure. **2) Isobaric heat addition (process 2-3):** Heat is added to the working liquid inside the boiler/evaporator. The heat addition process is isobaric. The high-pressure liquid is converted at constant temperature to high pressure steam inside the boiler. The steam exits at high pressure and enters the turbine at point 3. **3) Isentropic expansion (process 3-4):** The high-pressure steam rotates the turbine propellers as a result causing turbine shaft to start rotating, producing mechanical energy which is eventually converted to electrical energy by electric generator. During this process of turbine expansion, the high-pressure steam reduced to low pressure steam. The low-pressure steam enters the condenser at point 4. **4) Isobaric heat rejection (process 4-1):** The steam is converted at constant temperature back to liquid state inside the condenser. The heat is rejected/extracted from the steam at constant pressure.

	Note that condenser and boiler/evaporator are devices that change the state of working fluid without changing the temperature and pressure.
The Brayton cycle (gas turbine cycle) is the simplest gas turbine cycle. In this cycle, the air/gas is compressed, heated by a heat source, then expanded through a turbine to produce work that powers the compressor and an electric generator.	The Rankine cycle (steam turbine cycle) is a more efficient variant of the Brayton Cycle and is used in most electric power generation plants. In this cycle, water is heated to form steam, which expands through a turbine to produce work that powers an electric generator.

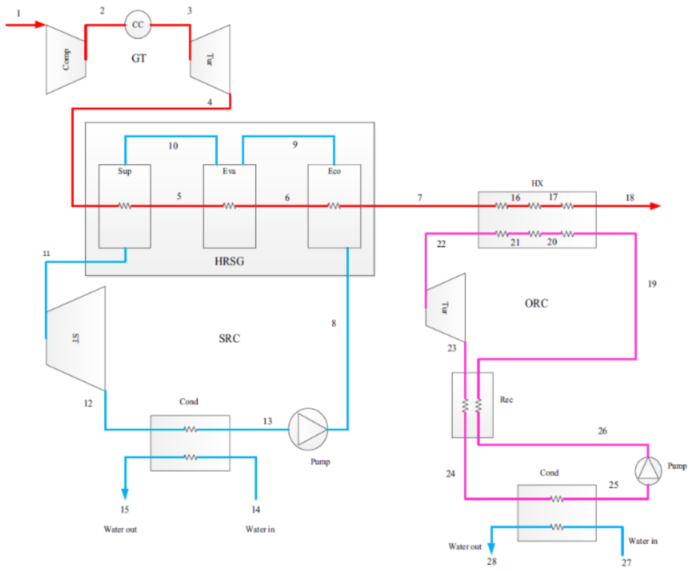
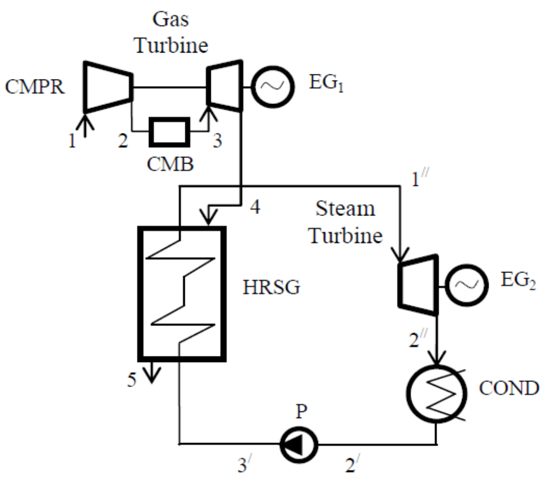


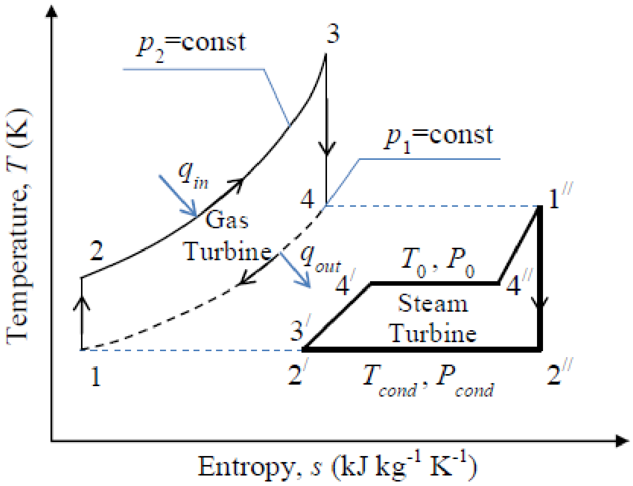
Figure 27. Proposed combined gas turbine, steam, and organic Rankine cycle [69]



CMPR – air compressor; CMB – combustor; EG$_1$ – gas turbine electric generator; HRSG – heat recovery steam generator; EG$_2$ – steam turbine electric generator; COND – condenser; P – pump

Figure 28. Combined-cycle power plant schematic diagram [70]

Some industrial applications may require the combination of Brayton cycle (or gas turbine), Rankine cycle (steam turbine) and Organic Rankine cycle (ORC turbine) in addition to the combination of regeneration, reheating, intercooling, recuperation, recompression and two-stage turbine expansion processes to efficiently use the waste heat arising from the industrial application. For example, the exhaust gases (flue gas) with huge source of waste energy from gas turbine can be passed through heat recovery steam generator (HRSG) with steam rankine cycle (SRC), and the exhaust gases from the HRSG can be further passed through organic rankine cycle (ORC) in order to remove substantial amount of waste heat energy for power/electricity generation in the SRC and ORC in addition to the power/electricity generated from the gas turbine as shown in Figure 27 (Mohammadi et al. [69]).



1–2–3–4 is the idealized Brayton cycle; $1^{//}–2^{//}–2^{/}–3^{/}–4^{/}–4^{//}$ is the idealized Rankine cycle; q_{in} and q_{out} are the heat supplied to and removed from the exhaust gases in the isobaric process at pressures p_2 and p_1, respectively

Figure 29. The idealized combined Brayton-Rankine cycle [70]

Table 6. Thermal efficiency calculation models of four cycles (Figures 30 and 31) [60]

Cycle	Thermal Efficiency
Simple recuperative cycle	$\eta_{simple} = \dfrac{(h_4 - h_5).\eta_G - (h_2 - h_1)/\eta_M}{(h_4 - h_3)}$
Reheating cycle	$\eta_{reheating} = \dfrac{(h_4 - h_5 + h_6 - h_7).\eta_G - (h_2 - h_1)/\eta_M}{(h_4 - h_3 + h_6 - h_5)}$
Recompression cycle	$\eta_{recompression} = \dfrac{(h_5 - h_6 + h_7 - h_8).\eta_G - [SR.(h_2 - h_1) + (1 - SR).(h_{3''} - h_{10})]/\eta_M}{(h_5 - h_4 + h_7 - h_6)}$
Intercooling cycle	$\eta_{intercooling} = \dfrac{(h_5 - h_6 + h_7 - h_8).\eta_G - [SR.(h_{12} - h_{11} + h_2 - h_1) + (1 - SR).(h_{3''} - h_{10})]/\eta_M}{(h_5 - h_4 + h_7 - h_6)}$

SR, split ratio; η_M, efficiency of the motor that drives the compressor; η_G, efficiency of the electric generator that is connected to the turbine for power generation.

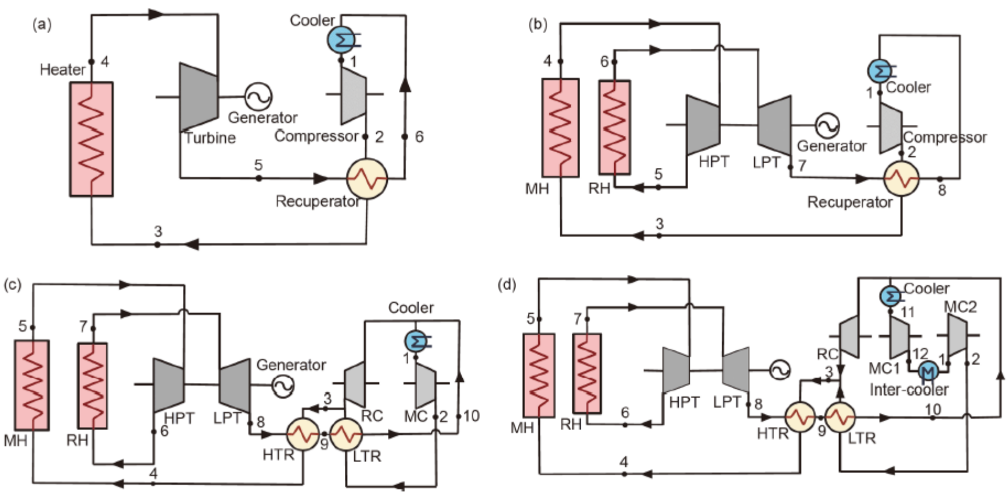


Figure 30. Configurations of (a) simple recuperative cycle, (b) reheating cycle, (c) recompression cycle, and (d) intercooling cycle. MH: main heater, RH: re-heater, HPT: high-pressure turbine, LPT: low-pressure turbine, MC: main compressor, RC: re-compressor, HTR: high-temperature recuperator, LTR: low-temperature recuperator [60]

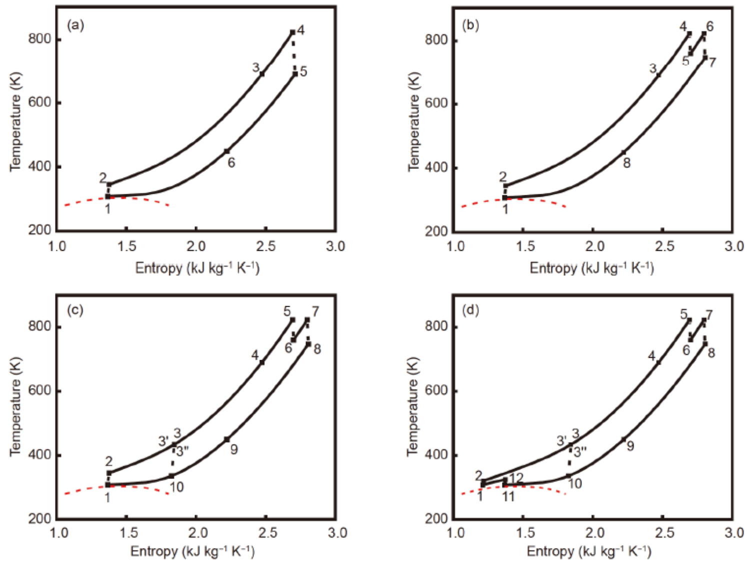


Figure 31. (Color online) T-s diagrams of (a) simple recuperative cycle, (b) reheating cycle, (c) recompression cycle, and (d) intercooling cycle [60]

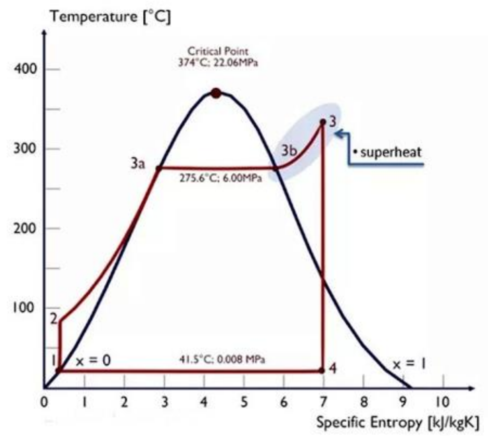


Figure 32. Rankine cycle with superheat of saturated steam [62]

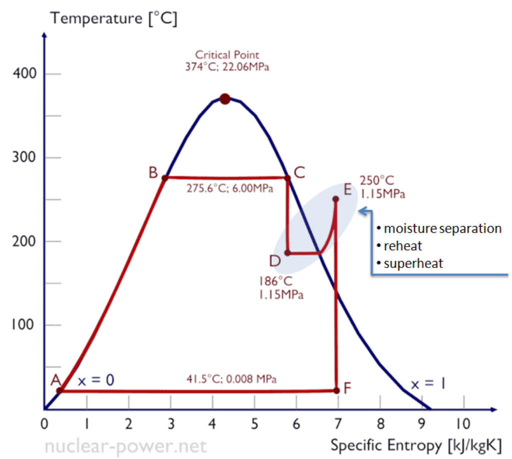


Figure 33. Rankine cycle with reheat of wet steam and superheat of saturated steam [61], [62]

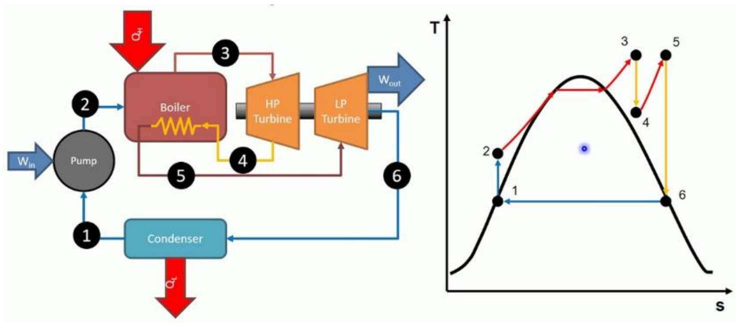


Figure 34. Rankine circle with reheat of superheated steam [74]

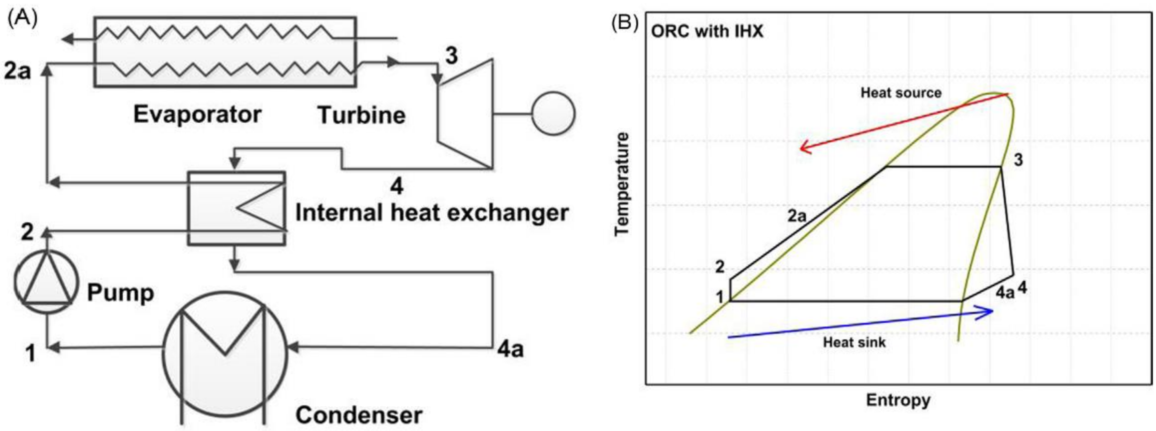


Figure 36. Schematics of a reheat ORC system: (A) system schematic and (B) T-S diagram [71]

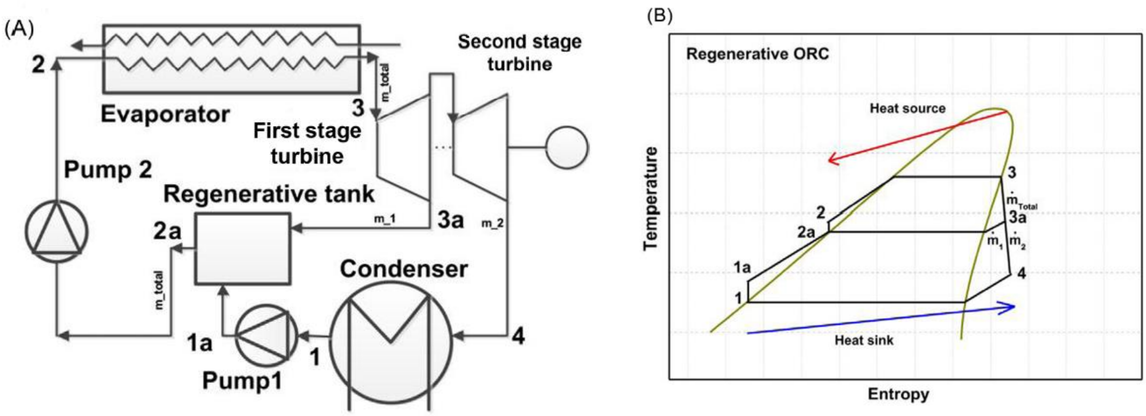


Figure 37. Schematics of a regenerative ORC system with recompression (two pumps): (A) system schematic and (B) T-S diagram [71]

Figures 28 and 29 respectively show components and process diagrams for combined cycle power plant of gas turbine and steam turbine in which the waste heat energy in exhaust gases from the gas turbine is utilized for power generation by the application of HRSG and steam turbine (Tarasova et al. [70]). The formulation of performance or optimization parameter varies according to the optimization processes involve in getting the maximum efficiency from particular industrial application. For example, Table 6 provides different efficiency formulations involving the use of Brayton cycles for recuperative cycle, recuperative cycle with reheating, recuperative cycle with reheating and recompression, and recuperative cycle with reheating, recompression and intercooling (Liu et al. [60]). Figure 30 shows the Brayton cycles with components for the various recuperative cycles and Figure 31 shows the process diagrams for the various recuperative cycles with reheating, recompression and intercooling with regards to the process optimization of Brayton cycle (Liu et al. [60]).

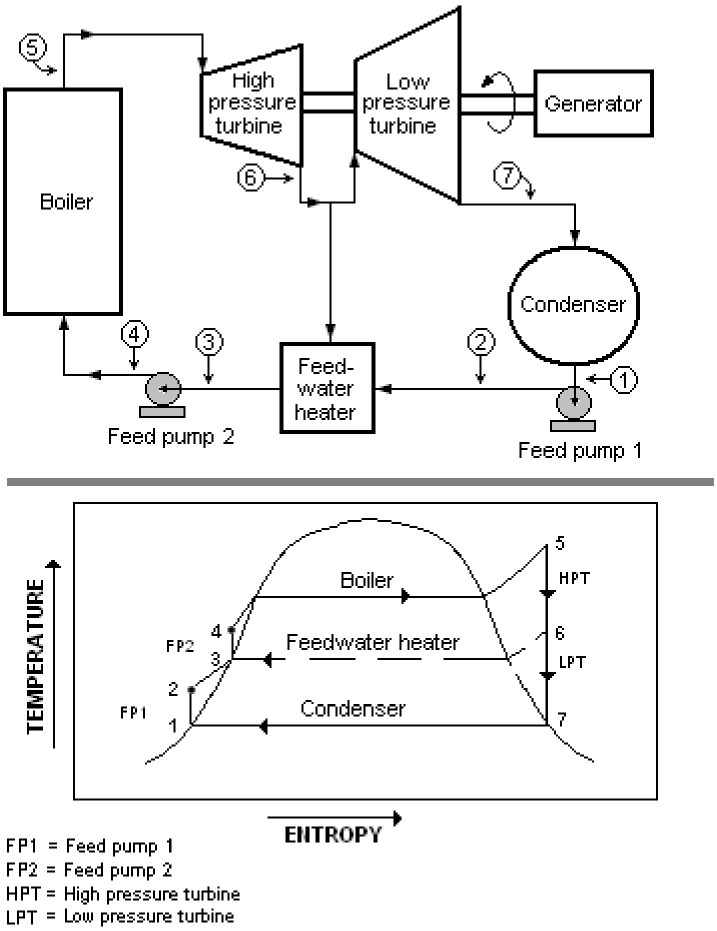


Figure 38. A Rankine cycle with two pumps, two steam turbines and a single open feedwater heater [73]; [62]

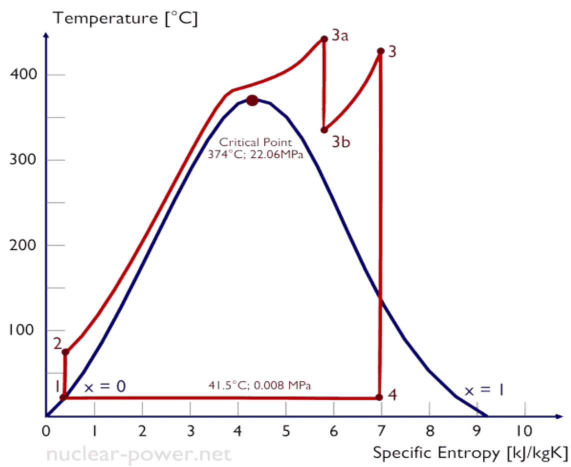


Figure 39. Process diagram for Supercritical Rankine cycle with reheat [74]

Rankine cycle power efficiency can be increased by selection of optimum increased pressure of boiler/evaporator, by selection of optimum decreased pressure of condenser, by making use of the processes of reheating and superheating possible with steam turbine expansion, and by adoption the process of heat regeneration. Rankine cycle power efficiency can also be increased by using supercritical Rankine cycle which is applicable to SCWRs and supercritical fossil fuel power plants.

The temperature in the boiler/evaporator and condenser depends on the pressure set in these heat exchangers. The higher the pressure in the boiler the better the steam quality, and the lower the pressure in the condenser the lower the saturation temperature of the water/liquid. These pressure conditions in the boiler and condenser favors higher efficiency of Rankine cycle. However, the selection of the pressure in the boiler and in the condenser may also depends on the other factors of consideration. The pressure conditions in the boiler and condenser guide the selection of optimum range at which pressure should be decreased in the turbine. That is the steam expansion (or decrease in pressure) in the turbine should end at some point to avoid damages caused to steam turbine blades by low-quality steam resulting/arising from continuous pressure decrease in the turbine.

Superheating which involves further heating of saturated vapor (or dry steam) at constant pressure to obtain superheated steam prior to turbine expansion, favors higher efficiency of the Rankine cycle (see Figure 32). Reheating involves heating of wet steam from turbine expansion at constant pressure to obtain saturated vapor (or dry steam) (see Figure 33). The saturated vapor can then be superheated at constant pressure to obtain superheated steam prior to another turbine expansion. Increasing the Rankine cycle efficiency by the two processes of reheating and superheating is carried out by two-stage turbine expansion (see Figure 33). Increasing the Rankine cycle by the process of reheat of superheated steam is also obtained by two-stage turbine expansion (see Figure 34). Increasing the Rankine cycle efficiency by only the process of superheating is carried out by one stage turbine expansion (see Figure 32).

Expanded steam could be extracted from turbine expansion and fed directly into the feedwater thereby raising/increasing the temperature of the feedwater, or wet steam or saturated steam extracted from steam lines and fed directly into the feedwater thereby raising/increasing the temperature of the feedwater, is generally referred to as Regeneration. Efficiency of the Rankine cycle could be increased by heat regeneration because of heat flow into the cycle occurring at a higher temperature, which eventually increases the working steam quality. The optimization processes showing schematic of components and process diagrams for Organic Rankine Cycle (ORC) are shown in Figure 35 (Regeneration/recuperation), Figure 36 (Reheating) and Figure 37 (Recompression with two pumps and Regeneration with two-stage turbine expansion) (Li [71, 72], Zhang et al. [73]). Figure 38 combines the processes of recompression (two pumps), two-stage turbine expansion (two steam turbines) and regeneration/recuperation (feedwater heater using expanded steam or low-pressure steam) for Rankine cycle.

With the design of materials that can withstand high pressure and temperature conditions, steam can be obtained at supercritical pressures beyond the critical point of water (T_{CR} = 374 °C; PCR = 22.1 MPa) and fed directly into steam turbine using SWCRs and supercritical fossil fuel power plants. Efficiency of SCWRs operated for power generation could be increased up to 45% compared to that of light water reactors LWRs which could be increased up to 33%. Efficiency of supercritical fossil fuel power plants operated for power generation could be increased up to 48% when processes of reheating are incorporated in the supercritical Rankine power efficiency optimization. Figure 39 shows process diagram for reheat supercritical Rankine cycle with two-stage turbine expansion, which could increase the efficiency of the Rankine cycle more than that of Rankine cycle at subcritical pressures.

4. Description of Heat Transfer Devices/Systems

Heat transfer devices /systems are designed mainly to increase or decrease the temperature of a fluid (gas or liquid) from initial temperature to final temperature by absorption of sensible heat, or to change a phase of a fluid to a different phase by the absorption or release of latent heat. Most of the heat transfer devices/systems are normally referred to as heat exchangers and may include boiler, steam generator, condenser, radiator, evaporator, cooling tower, regenerator, recuperator, heater, cooler, precooler, economizer and superheater. Several studies are being carried out for the possibility of using PCHEs as some of these heat transfer devices/systems such as intermediate heat exchangers of nuclear power plants, precoolers, recuperators and condensers among others. Table 7 provides brief descriptions of heat transfer devices/systems used for industrial applications requiring heating and cooling of working fluid to obtain the desired outputs as in power generation and automobile industries (Patel et al. [75], Zohuri [76], Pra et al. [77]).

Table 7. Brief description of heat transfer devices/systems used for specific industrial applications (Patel et al. [75], Zohuri [76]).

Industrial Heat transfer device/system	Description/Definition	Application/Fields of application
Heat Exchanger	A heat exchanger is a system used to transfer heat between two or more fluids. Heat exchangers are used in both cooling and heating processes. The fluids may be separated by a solid wall to prevent mixing or they may be in direct contact	Specific applications for heat exchangers include: (i) Heating a cooler fluid using the heat from a hotter fluid

	The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air. Another example is the heat sink, which is a passive heat exchanger that transfers the heat generated by an electronic or a mechanical device to a fluid medium, often air or a liquid coolant.	(ii) Cooling a hot fluid by transferring its heat to a cooler fluid (iii) Boiling a liquid using the heat from a hotter fluid (iv) Boiling a liquid while condensing a hotter gaseous fluid (v) Condensing a gaseous fluid by means of a cooler fluid. Heat exchangers are widely used in space heating, refrigeration, air conditioning, power stations, chemical plants, petrochemical plants, petroleum refineries, natural-gas processing, and sewage treatment.
Steam generator	A steam generator is a device/system that uses a heat source to boil liquid water and convert it into its vapor phase (or steam) under pressure higher than atmospheric pressure. The heat source is a hot fluid stream rather than the products of combustion. The hot fluid heat source for a steam generator is mostly the direct or indirect coolant of primary system of a nuclear power plant for power or electricity generation.	Steam generators used in nuclear power plants produce steam mostly for power or electricity generation.
Steam boiler	Steam boilers are typically larger pressure vessels capable of providing energy to industrial operations. They accomplish this by boiling water at subcritical pressures with intricate fuel systems. The heat may be derived from the combustion of a fuel such as coal, petroleum fuel oil, natural gas, municipal waste or biomass, and other sources. Steam boilers are used to generate/produce steam/vapor for industrial operations/applications. The steam generation/production takes place at a particular pressure and hence the required temperature of a steam for particular operation/application is determined by the system Pressure. A Boiler is the biggest and most critical part of a thermal power plant.	Boilers can be applied in (i) Operating steam engines. (ii) Operating steam turbines. (iii) Operating reciprocating pumps. (iv) Industrial process work in chemical engineering. (v) Producing hot water required to be supplied to room in very cold areas. (vi) Thermal power stations. The heat content of the steam is large and thus it is suitable for process heating in many industries like sugar mills, textile mills, dairy industry and also in chemical industries.
Cooler/Cooling system	Cooler/cooling system is an apparatus employed to keep the temperature of a structure or device from exceeding limits imposed by needs of safety and efficiency.	Cooling systems are employed in automobiles, industrial plant machinery, nuclear reactors, power plants, air-conditioning in buildings, and many other types of machinery and applications that require cooling to ensure system safety and also to produce the desired outputs.
Regenerator	A regenerative heat exchanger (commonly known as a Regenerator) exchanges heat from one flowing process fluid to an intermediate solid heat storage medium, then that medium exchanges heat with a second flowing process fluid. The two flows are either separated in time, alternately circulating through the storage medium, or are separated in space and the heat storage medium is moved between the two flows.	Regenerator is used in the hot blast process on blast furnaces; in glass melting furnaces and steel making, to increase the efficiency of open-hearth furnaces; in high pressure boilers; and in chemical and other applications.
Recuperator	A recuperator is a special purpose counter-flow energy recovery heat exchanger positioned within the supply and exhaust air streams of an air handling system, or in the exhaust gases of an industrial process, in order to recover the waste heat. Generally, they are used to extract heat from the exhaust gases and used it to preheat air entering the combustion system. In this way, they use waste energy to heat the air, offsetting some of the fuel, and thereby improve the energy efficiency of the system as a whole. The temperature of the air is increased by sensible heat absorption. Regenerators and recuperators are heat exchange systems that recover heat by cycling through heat sinks (regenerators)	The heat recovered by a regenerator or recuperator is most commonly used to preheat combustion air to a furnace. Regenerators and recuperators may also be attachments to thermal oxidizers that preheat the exhaust air from industrial processes prior to treatment for HAP (Hazardous Air Pollutant) and VOC (Volatile Organic Chemical) contaminants.

	or through a high temperature metallic heat exchanger (recuperators).	
Evaporator	An evaporator is a device used to turn the liquid form of a chemical substance, such as water, into its gaseous form - vapor. In this process, liquid is converted to vapour/steam at the same temperature through latent heat absorption (liquid is evaporated or vaporized). That is, change of state of the liquid to vapor/steam occurs at constant temperature and pressure.	Evaporators find applications in a broad spectrum of processes in various industrial sectors including power generation, pharmaceuticals, food & beverages, pulp & paper, and chemicals.
Radiator	Generally, a radiator is a heat exchanger that is used to transfer thermal energy from one medium to another for the essence of cooling and heating. Radiators consist of a large area of the cooling surface and used the stream of air to take away the surrounding heat. Radiators are common types of heat exchangers designed to transfer heat from hot coolant to the atmosphere. This is achieved by a coolant fan sucking the heat from the radiator through a blown air to the atmosphere.	Radiators are used for cooling internal combustion engines, mainly in automobiles but also in piston-engined aircraft, railway locomotives, motorcycles, stationary generating plants and other places where heat engines are used.
Condenser	A condenser is a device used for reducing a gas or vapour to a liquid. Vapour/steam is converted to liquid at the same pressure and temperature through the release of latent heat in a condenser.	Condensers are employed in power plants to condense exhaust steam from turbines and in refrigeration plants to condense refrigerant vapors, such as ammonia and fluorinated hydrocarbons. The petroleum and chemical industries employ condensers for the condensation of hydrocarbons and other chemical vapors. In distilling operations, the device in which the vapor is transformed to a liquid state is called a condenser.
Cooling tower	A cooling tower is a heat rejection device that rejects waste heat to the atmosphere through the cooling of a coolant stream, usually a water stream to a lower temperature. Cooling towers use the evaporation of water to remove process heat and cool the working fluid to near the wet-bulb air temperature. In the case of closed-circuit cooling towers or dry cooling towers, they rely solely on air to cool the working fluid to near the dry-bulb air temperature using radiators.	Cooling towers are used for several applications including cooling the circulating water used in oil refineries, petrochemical and other chemical plants, thermal power stations, nuclear power stations and HVAC (heating, ventilation, and air conditioning) systems for cooling buildings.
Economiser	An Economiser is a heat transfer device used to increase the temperature of feedwater before it enters boiler/evaporator system. The temperature of the liquid is increased by sensible heat absorption. Economisers can be comparable to feedwater preheaters or feedwater heaters.	A common application of economizers in steam power plants is to capture the waste heat from boiler stack gases (flue gas) and transfer it to the boiler/evaporator feedwater. This raises the temperature of the boiler/evaporator feedwater needed for the rated boiler/evaporator output.
Superheater	Superheaters are specially designed to increase the temperature of the saturated steam (or dry steam) and to help control the superheated steam outlet temperature. For example, in HRSGs, they are simple single-phase heat exchangers with steam flowing inside and the flue gas passing outside, generally in the cross flow, eventually increasing the temperature of the steam.	Superheaters are used in power generation industries for electricity generation. Superheated steam is used in steam turbines for electricity generation, in steam engines, and in processes such as steam reforming.
Reheater	Reheater is a heat transfer device used to elevate the temperature conditions of a wet steam back to the required saturated steam temperature conditions after the steam losses its energy going through systems (e.g., turbine), eventually reducing its temperature and pressure. That is, Reheater is used to heat a fluid again after the fluid loses its heat while going through a system (e.g., going through turbine expansion).	Reheaters are used in power generation industries for electricity generation.
Heater	A heater is a piece of heat transfer equipment or a machine which is used to raise the temperature of a fluid (gas or liquid) for industrial application, for domestic use or for any other purpose such as air inside a room or a car. A European study estimated that: • 30% of industrial heating applications required heat below 100°C	The industrial sector uses heat for many applications; including washing, cooking, sterilizing, drying, preheating boiler feed water, viscosity control, and many other processes. Some specific industries that require heat in one form or another are Gas Processing Industry, Chemical and Petrochemical Industries, Power

	• 27% of industrial heating applications required heat between 100°C and 399°C • 43% of industrial heating applications required heat above 399°C	Generation, Aerospace, Food & Beverage, Agriculture, and General Manufacturing industries. There are four primary types of heat used in industrial processing: Fossil fuel, Steam, Electric and Hydronic.
Preheater	In general, a preheater is a heat exchanger that increases the temperature of a gas or liquid before it is fed into another process. Air preheaters and feedwater heaters are normally used to preheat a gas or liquid. An air preheater or air heater is a general term to describe any device designed to heat air before another process (for example, combustion in a boiler) with the primary objective of increasing the thermal efficiency of the process. They may be used alone or to replace a recuperative heat system or to replace a steam coil. A feedwater heater is a power plant component used to pre-heat water delivered to a steam generating boiler. In large steam systems, or in any process requiring high temperature, the input fluid is usually preheated in stages, instead of heating it in one step from ambient to the final temperature. Preheating in stages increases efficiency and minimizes thermal shock stress to components, as compared to injecting ambient temperature liquid into a boiler or other device that operates at high temperatures.	Various types of air-preheaters find application in papers mills, sugar mills and in power plants. Feedwater heaters are used in both fossil- and nuclear-fueled power plants.
Intercooler and Aftercooler	Intercooler and After-cooler are almost same mechanical device serving the same purpose in cooling charge air. An intercooler or aftercooler is also referred to as a Charge-Air Cooler. An intercooler or aftercooler (or charge air cooler) could also be defined as a mechanical heat exchanger device used to cool a compressed air before an automotive or industrial application.	Aftercoolers and Intercoolers perform similar functions. Aftercoolers and Intercoolers are used in many applications, including air compressors, air conditioners, refrigerators and gas turbines, and are widely known in automotive use as an Air to Air Cooler or air-to-liquid cooler for forced induction (turbocharged or supercharged) internal combustion engines to improve their volumetric efficiency by increasing intake air charge density through nearly isobaric (constant pressure) cooling. Intercoolers are mostly used in supercharged or turbocharged internal combustion engines (compared to aftercoolers) to cool the air compressed by the turbocharger or supercharger before it enters the internal combustion engine, eventually reducing it to a lower temperature and allowing more air to enter for increased fuel combustion, and hence increase in both the maximum power output and efficiency of the engine. All industrial systems require an aftercooler (compared to intercooler), which is the most important part of an air compressor.
Precooler	Precooler is a kind of tube heat exchanger, which is used to pre-cool the incoming high-temperature intake air/gas before the active compression process. Precooler could also be defined as a heat exchanger device used for cooling a fluid or gas before it is used by a mechanical device such as compressor and other devices requiring cooling before the system can function safely to produce the desired output.	Precoolers are used in the optimization of gas energy cycles such as Brayton cycles as shown in Figures 30, 40, and 41. That is Precoolers are mostly used together with Recuperators in power generation industry to improve the performance and hence improve the efficiency of gas energy/power cycles.

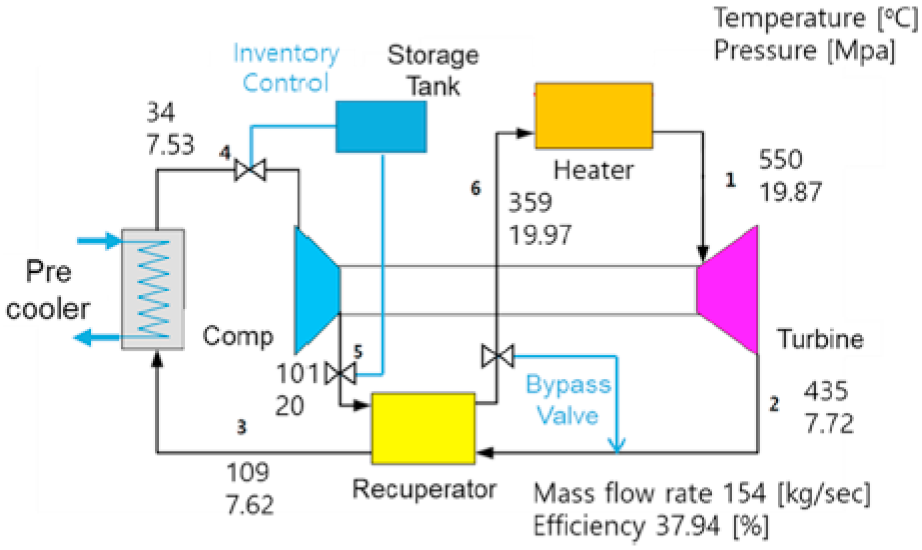


Figure 40. The configuration of the simple recuperated cycle [19]

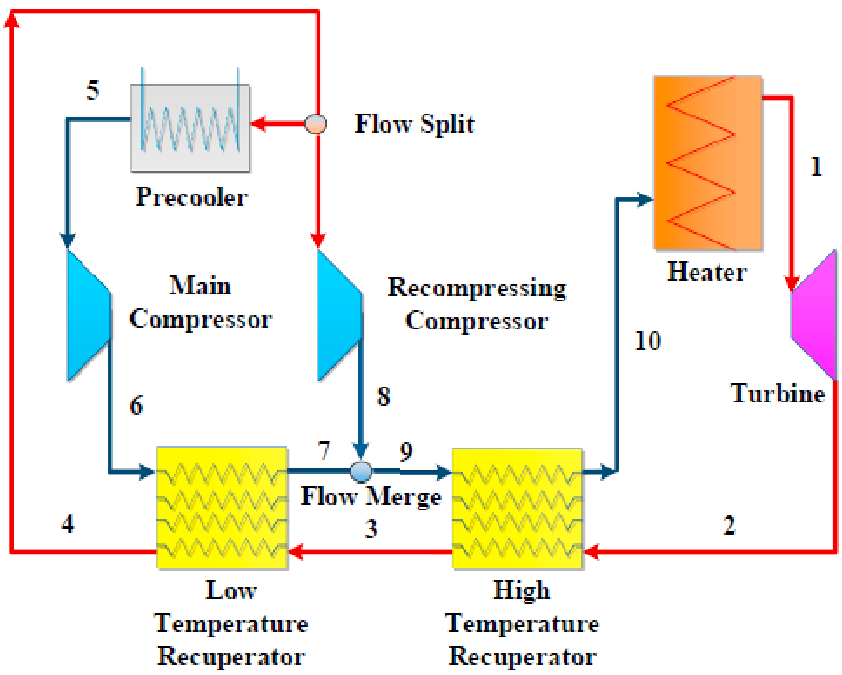


Figure 41. The configuration of the recompression cycle [78], [19]

5. Conclusion

PCHEs are promising heat exchangers and can be used in many industrial installations requiring heating and cooling applications because they have compact size and also have capability of withstanding high pressure and temperature conditions. A brief review has been carried out on PCHEs focusing on recent studies existing in literature. The review among other things covers overview of reviews carried out on PCHEs and other heat exchanger types, design and optimization of PCHEs, optimization of Brayton and Rankine or organic Rankine cycles, and brief descriptions of heat transfer devices/systems. The following findings have been obtained as a result of the review:

1. Depending on the particular application of PCHE, geometry/structure design (mechanical design) and operational (system input) parameters could be optimized based on system output and performance parameters to obtain optimized design and operational parameters of the PCHE that could be applied for the particular application.

2. In addition to the system optimization using mechanical design parameters and system input parameters (operational parameters), industrial systems that require the use of Brayton, Rankine and Organic Rankine cycle can be further optimized using optimization processes such as reheating, recompression, intercooling, recuperation, and regeneration. The use of Rankine cycle, the organic Rankine cycle or the combination of Rankine and organic Rankine cycles can be used to derive maximum waste heat energy/power from exhaust gases (flue gas) arising from the operation of gas turbine for power production.
3. Studies providing more data on design/geometry to help readers know various ways design/geometry can be modified to help obtain optimum design/geometry data should be carried out. These studies would help simplify system geometry design and reduce the overall cost of geometry design and manufacturing/fabrication.
4. Studies providing more data on various ways to help decrease pressure drop, increase overall heat transfer coefficient, decrease compactness factor, increase efficiency, increase effectiveness, increase heat transfer rate, increase log mean temperature difference LMTD, etc. should be carried out as these conditions support data optimization on design and operation of PCHEs.
5. Heat transfer and friction factor correlations have been developed for some specific channel types (such as straight, zigzag, wavy and airfoil). But it has not been established that the developed heat transfer and friction factor correlation for particular channel type could predict fluid flow and heat transfer characteristics well for various types of the flow cross-sections (circular, square, rectangular, semi-circular, airfoil, etc.) applicable to the channel type. In this regard, more experimental studies should be carried out to support validation of numerical studies being performed to help develop universal heat transfer and friction factor correlations to cover different flow cross-sections and channel types of PCHEs, and also to cover wider range of geometry and operational/flow parameters for accurate prediction of heat transfer, pressure drop, and system performance data on design and operation of PCHEs.
6. Studies based on mechanical design, thermal hydraulic design and optimization of PCHEs are being carried out using supercritical carbon dioxide (sCO_2) Brayton cycle for the possibility of using sCO_2 for power generation in addition to the use of steam Rankine cycle for power production. Similar studies are also being carried out for the possibility of replacing steam Rankine cycle with sCO_2 Brayton cycle in the design of SFR (sodium-cooled fast reactor) to eliminate the consequences of accidental sodium-water reaction.
7. There are specific formulations/correlations for design evaluation (e.g., Log Mean Temperature Difference (LMTD) and ε-NTU methods), performance evaluation and economic evaluation of PCHEs and other heat exchanger types. Power/energy cycle efficiency formulation for gas or rankine cycle is used to evaluate the performance of power plants.

This review among other things would be useful to readers and scientists seeking basic information on design and optimization of PCHEs, optimization of Brayton and Rankine cycles used for performance assessment of power cycles, and differences between heat transfer devices/systems. The broad research findings on PCHEs and other heat exchanger types provided in Tables 1 and 2 would guide the design of new experimental and numerical studies on fluid flow and heat transfer, and design and optimization of heat exchangers.

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Acknowledgement

This study received no source of funding.

Conflict of interest

The Authors has no conflicts of interest to declare that they are relevant to the content of this article.

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