MODELING AND ANALYSIS OF A PRINTED CIRCUIT HEAT EXCHANGER FOR SUPERCRITICAL CO₂ POWER CYCLE APPLICATIONS

Ajinkya Meshram¹, Ankush Kumar Jaiswal¹, Sagar D. Khivsara¹, Jesus D. Ortega²,
Clifford Ho², Rucha Bapat¹, Pradip Dutta^{1*}

¹Dept. of Mechanical Engineering, Indian Institute of Science, Bangalore, KA-560012, India

²Sandia National Laboratories, Albuquerque, NM-87185, U.S.

*Corresponding author:

Email: pradip@mecheng.iisc.ernet.in Tel.: +91 80 22933225; fax: +91 80 2360 4536

ABSTRACT

The supercritical carbon dioxide (S-CO₂) based Brayton cycle is a good alternative to conventional power cycles because of high cycle efficiency, compact turbo machinery and compact heat exchangers. In this cycle, the majority of heat transfer (approximately 60-70% of total cycle heat transfer) occurs in the regenerator. For the regenerator, micro-channel heat exchanger is an attractive option because of its high surface-area-to-volume ratio. In this study, the performance of a printed circuit heat exchanger (PCHE) with straight and zigzag channels is evaluated. The study is performed for fully turbulent conditions. The channel diameter and the operating Reynolds number play significant roles in the overall heat transfer and pressure drop of hot and cold channels of S-CO₂. For zigzag channels, it is found that a larger bend angle and smaller linear pitch perform better than a smaller bend angle and large linear pitch combination. Correlations for Nusselt number and friction factor are developed using ANSYS Fluent and are subsequently utilized in one dimensional (1D) thermal modelling of the heat exchanger. For the same thermal capacity, the model indicates that the zigzag channel PCHE volume is significantly smaller than that of a straight channel PCHE because of higher heat transfer coefficient. However, the pressure drop incurred in the former design is larger.

Keywords: Supercritical CO₂, Brayton cycle, PCHE, Regenerator, Zigzag channel

1. INTRODUCTION

The relentless efforts to increase the efficiency of power cycles have lead researchers to explore S-CO₂ as the working fluid in Brayton cycle [1-4]. The major advantages of S-CO₂ Brayton cycle compared to other conventional cycles such as air Brayton and steam Rankine cycles are high cycle efficiency (~50 % @ 700°C turbine inlet temperature), compact turbo-machinery and compact heat exchangers. Moreover, the possibility of small scale (1-10 MW range) S-CO₂ Brayton cycle based concentrated solar power (CSP) plants is another motivation for performing component level evaluations [5]. The key aspect of S-CO₂ Brayton cycle is that a significant amount of heat (approximately 60-70% of total heat transfer) is regenerated after turbine exhaust. Therefore, performance of the cycle is strongly dependent on the regenerator effectiveness and the pressure drop in the hot and cold channels of the heat exchanger. The printed circuit heat exchanger (PCHE) is known to be an attractive option for a regenerator because of its high heat transfer area to volume ratio (650 m²/m³), low approach temperature [1], suitability at high pressure and high temperature conditions [6]. The detailed configurations and manufacturing processes of PCHE are explained by Hesselgreaves [7]. Recently, a review of compact heat exchangers is performed by Li *et al.* [8]. It suggests the potential application of compact heat exchangers (CHE) for solar receivers in CSP plants. There are various PCHE studies which investigate the heat transfer and pressure drop in several geometries with multiple fluids and flow transitional behavior. Geyer et al. [9] investigated the performance of PCHE with a trapezoidal channel. The enhancement in heat transfer was found to be up to four times as compared to the straight channel PCHE, but lower than the zigzag channel configuration. Moreover, the pressure drop of trapezoidal channel PCHE was significantly lower compared to the latter. In another study, Ngo *et al.* [10] performed a comparative study of zigzag and S-shaped fin configurations of PCHEs with carbon dioxide (CO₂) as the working fluid. The Nusselt number and friction factor correlations at low temperatures (~100-300°C) with various Reynolds numbers are also presented. With S-CO₂ as the working fluid, Kim *et al.* [11] evaluated the performance of air foil shaped fin PCHE, in which the heat transfer rate per unit volume is found to be approximately similar to that of the zigzag channel design with significantly lower pressure drop, but the manufacturing of such PCHEs is immensely complex. Mohammed et al. [12] evaluated the performance of PCHE with various configurations of channels and concluded that

the heat transfer coefficient of zigzag channels is maximum compared to curvy and step channels; however, its pressure drop is maximum. In a study of flow characteristics, Figley *et al.* [13] ascertained that the transitional Reynolds number is higher for PCHE operating with helium as working fluid. Dai *et al* [14] performed the experimental study of zigzag channel PCHE with water and found that the transition to transient flow occurs at very low Reynolds number in zigzag channels compared to straight channels within the laminar regime. The S-CO₂ Brayton cycle also consists of a pre-cooler, which can be air or water cooled. Meter [15] has evaluated the performance of PCHE with S-CO₂ and water as working fluids; the operating conditions were approximately similar to S-CO₂ Brayton cycle pre-cooler.

In the present work, PCHE is analyzed as a candidate for S-CO₂ cycle regenerator, which is subjected to high pressures (~200 bar) and temperatures (~800 K). There is not much study reported in literature on PCHE performance at such operating conditions, which are more suitable for regenerator applications in closed cycle S-CO₂ Brayton cycle. A steady state study is performed with straight and zigzag channels in a turbulent regime. The computational study performed in this work can be summarized as follows:

- The entire temperature zone of actual regenerator is divided into two parts (lower and higher temperature range). This leads to a better understanding of the temperature dependent performance. The temperature and pressure variation along the length of PCHE channel is also analysed.
- The dependence of Nusselt number and friction factor on Reynolds number is studied for straight and zigzag channel PCHEs. Correlations for lower and higher temperature range are presented.
- Performance parameters such as overall heat transfer coefficient and pressure drop are evaluated with various channel diameters for a straight channel PCHE; and with several bend angles and linear pitch combinations for a zigzag PCHE.
- The correlations obtained from 3D ANSYS Fluent simulations are further utilized in a 1D thermal-hydraulic study, and the temperature and pressure variation for the extended heat exchanger channel length is presented.

2. SYSTEM DESCRIPTION

The schematic of S-CO₂ Brayton cycle is shown in fig. 1. The major components are turbine, compressor, regenerator, air cooled pre-cooler and receiver, which absorbs concentrated solar energy. In this cycle, the hot fluid from turbine exhaust is utilized to heat the cold fluid from compressor outlet; this heating takes place in the regenerator. The PCHE consists of multiple channels on a plate and multiple such plates joined with the help of diffusion bonding [7]. In this study, a single pair of channel is considered with periodicity assumption on the side walls. The error induced through this assumption will be low because PCHEs usually consist of a very large number of channels. Schematics of straight and zigzag PCHE channel configurations are shown in fig. 2 a) and fig. 2 b), respectively. The corresponding isometric views of computational grids are shown in fig. 3 a) and fig. 3 b), respectively.

3. MATHEMATICAL MODELING

Equations 1-3 show the governing equations of fluid domains for cold and hot S-CO₂ as shown in fig. 2 a) and b). These equations are the conservation equations for mass, momentum and energy. The value of i in momentum equation varies from 1 to 3; and it represents equations in x, y and z directions, respectively, and j is a summation index which varies from 1 to 3.

Continuity equation:

$$\frac{\partial(\rho u_j)}{\partial x_j} = 0 \quad (1)$$

Momentum Equation:

$$\rho \frac{\partial(u_i u_j)}{\partial x_j} = -\frac{\partial p_i}{\partial x_i} + \mu \frac{\partial}{\partial x_j} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) \quad (2)$$

Energy Equation:

$$\frac{\partial}{\partial x_j} \left[u_j (\rho E + p) \right] = \frac{\partial}{\partial x_j} \left((k_f + k_t) \frac{\partial T}{\partial x_j} \right) \quad (3)$$

The energy equation of solid domain is given by eqn. (4).

$$\frac{\partial}{\partial x_j} \left(k_s \frac{\partial T}{\partial x_j} \right) = 0 \quad (4)$$

In this work, the Reynolds-averaged Navier-Stokes (RANS) model [16] is opted for modeling turbulence. The k- ϵ model with standard wall function is utilized for evaluating the Reynolds shear stress. This model solves two additional equations for kinetic energy (k) and dissipation (ϵ). Low computational expense is the major motivation for selection of this model [17]. The mesh needs to be generated with the first cell height (y) such that $y^+ > 32.5$ [18]. Relation for y^+ and y is given by eqn. (5), where the wall shear stress (τ_w) is calculated from eqn. (6).

$$y^+ = \frac{y}{\nu} \sqrt{\tau_w / \rho} \quad (5)$$

$$\tau_w = \frac{0.079 \times u_{av}^2 \times \text{Re}_{d_h}^{-0.25}}{2} \quad (6)$$

ANSYS Fluent 13.0 is used for numerical evaluation of the PCHE. This commercial package uses the finite volume method to discretise the governing equations. The advection terms are discretised by first order upwinding scheme. The hexahedral meshes are created using ANSYS ICEM CFD by employing blocking and mesh extrusion methodology. The grid was refined until doubling the number of grid points yielded less than 1% difference in the solution parameters. The boundary conditions of the computational domain are shown in table 1. The velocity inlet and pressure outlet boundary conditions are specified at inlet and outlet of the flow channels. Periodic boundary conditions are specified on top and bottom walls. The side, front and back walls are specified as adiabatic walls. The alloy Inconel 617 is considered as the material for PCHE due to its suitability and high endurance at extreme temperature and pressure conditions. The thermo-physical properties of the solid domain are assumed constant, and the values of density (ρ), specific heat (c_p) and thermal conductivity (k) are 8360 kg/m³, 0.417 kJ/kg-K and 21 W/m-K, respectively. ANSYS Fluent is coupled with NIST database REFPROP [19] for obtaining accurate properties of S-CO₂. For the 1D thermal-hydraulic modelling, the properties of S-CO₂ are calculated by coupling REFPROP to MATLAB. The heat transfer coefficient, Nusselt number and friction factor are calculated using eqn. (7)-(9). T_b in eqn. 7 is the volume weighted average bulk S-CO₂ temperature and T_w is the area weighted average wall

temperature. The heat transfer coefficient h , defined using T_b and T_w as shown in eqn. 7, is the average heat transfer coefficient over the domain.

$$h = \frac{q}{T_b - T_w} \quad (7)$$

$$Nu = \frac{hd_h}{k_f} \quad (8)$$

$$f = \frac{\Delta p \cdot d_h}{2\rho L_{ch} V^2} \quad (9)$$

$$\text{where, } d_h = \frac{\pi D_{ch}}{2(\pi/2 + 1)} \quad (10)$$

The flow length of zigzag channel is maintained identical to the straight channel for the temperature variation, pressure drop and Reynolds number variation studies. In the temperature variation and pressure drop studies, the mass flow rate of cold and hot S-CO₂ for straight and zigzag channel studies is kept same. For Reynolds number and straight channel diameter variation studies, the values of inlet velocity for hot and cold S-CO₂ are calculated based on the desired inlet Reynolds number. In the straight channel diameter variation study, the diameter is varied from 1.2 mm to 2.6 mm, with a step size of 0.2 mm; and the inlet Reynolds number is varied from 8,500 to 20,500. For the zigzag channel parametric studies, bend angles are varied from 100° to 120° in steps of 10°, and the linear pitch is varied from 8 mm to 12 mm, with a step of 1 mm. The parametric variation study is performed exclusively in a lower temperature range for evaluating the trend.

4. RESULTS AND DISCUSSION

The most important aspects of any numerical study are the geometrical parameters and the boundary conditions. Unless specified otherwise, the width, height, channel diameter and length for the undertaken numerical studies are shown in table 2, while the values of boundary conditions are shown in table 3. Table 3 also indicates the values of higher and lower temperature ranges in which the studies are undertaken. The velocity values shown in table 3 are used in the temperature variation and pressure drop studies only. In the entire study, the channel diameter is fixed at 2 mm, except in the straight channel PCHE diameter variation study. During the straight channel diameter variation study, the values of plate thickness ($H_{ch}/2$) and half wall

thickness $(W_{ch}-D_{ch})/2$ are fixed to 1.6 mm and 0.25 mm, respectively. The zigzag PCHE bend angle and linear pitch variation studies are performed with a fixed Reynolds number of 14,500. The flow length in zigzag PCHE is kept constant at a value of 100 mm.

4.1 Temperature and pressure drop variation along the length

4.1.1 Straight channel

Figures 4 a) and 4 b) show the temperature variation and pressure drop of cold and hot S-CO₂ along the length of the straight channel PCHE for lower and higher temperature ranges. The properties of hot and cold S-CO₂ at the mean temperature and pressure are calculated and presented in table 4. This leads to understanding of dependence of properties on temperature and pressure in the specific cases considered. It can be observed that the variations of both temperature and pressure, is approximately linear in both temperature ranges. The slope of cold S-CO₂ temperature is slightly lower than that of hot S-CO₂ in the lower temperature range. The higher mean specific heat of colder S-CO₂ (can be observed in table 4) is the main reason for this behavior. Whereas, nearly similar specific heats of hot and cold S-CO₂ lead to almost similar temperature slope in the higher temperature range. The pressure drop in the hot S-CO₂ is significantly higher in both temperature ranges. Eqn. (11), obtained by combining eqn. (9) and (10), shows that the pressure drop in a fixed channel geometry and mass flow rate varies inversely with the fluid density. The substantial lower mean density of hot S-CO₂ is the major reason for larger pressure drop compared to cold S-CO₂. Pressure drop in the higher temperature range is slightly larger compared to the lower temperature range for both cold and hot S-CO₂ streams, with the same channel length. The reason for this is the marginally lower density of both fluids in the higher temperature range compared to the lower temperature range, as observed in table 4.

$$\Delta p = \left(\frac{256 f L_{ch} \dot{m}^2 (\pi/2 + 1)}{\pi^3 D_{ch}^5} \right) \frac{1}{\rho} \quad (11)$$

4.1.2 Zigzag channel

Similar to the straight channel PCHE temperature and pressure drop variations, fig. 5 a) and fig. 5 b) show the corresponding variations of cold and hot S-CO₂ streams along the length of zigzag channel PCHE for lower and higher temperature ranges. The bend angle and pitch

considered are 108° and 12 mm, respectively. It can be observed that the temperature difference between hot and cold S-CO₂ streams at any cross section is marginally lower than that of the straight channel PCHE. The outlet temperature values of cold and hot S-CO₂ with zigzag channel PCHE are 522.23, 639.15 and 466.69, 576.69 K for lower and higher temperature ranges, respectively, while the corresponding values are 498.45, 615.48 and 494.37, 601.83 K with straight channel PCHE. The large amount of heat transfer with zigzag channel leads to higher cold and lower hot S-CO₂ outlet temperatures. Compared to straight channel PCHE, in both temperature ranges, the pressure drops are significantly higher.

4.2 Reynolds number variation

4.2.1 Straight channel

The Nusselt and friction factor variation of straight channel PCHE with Reynolds number for hot and cold S-CO₂ are shown in figs. 6 a) and 6 b) for lower and higher temperature ranges, respectively. It can be observed that the Nusselt number increases approximately linearly with the Reynolds number, for both cases. The Nusselt number of cold S-CO₂ is marginally higher compared to hot S-CO₂ for all cases; the slightly higher Prandtl number for cold S-CO₂ is the main reason for this behavior, as observed in table 4. The friction factor decreases with increase in Reynolds number for both cases. However, the friction factor of cold S-CO₂ is slightly larger than that of the hot S-CO₂. The Reynolds number is varied in the range of 5,000 to 26,000. The Nusselt number and friction factor are observed to vary between 20 to 70 and 0.005 to 0.011, respectively.

4.2.2 Zigzag channel

The results of Reynolds number variation study for zigzag channel PCHE are shown in fig. 7 a) and 7 b). The Nusselt number and friction factor variation with Reynolds number is analogous to the straight channel PCHE. However, the values of Nusselt number and friction factor with zigzag PCHE is slightly higher compared to straight channel design. The range of Reynolds number variation analyzed for zigzag channels is 5,000 to 32,000. The Nusselt number and friction factor are observed to vary in the range of about 25 to 120 and 0.044 to 0.050, respectively.

4.3 Parametric variation

4.3.1 Straight channel diameter variation

The variation of overall heat transfer coefficient and pressure drop in the straight channel PCHE with varying channel diameter for various Reynolds numbers is shown in figs. 8 a), 8 b) and 8 c). It can be observed that the overall heat transfer coefficient decreases with diameter, while it increases with Reynolds number. Similarly, the pressure drops also decrease with diameter and increase with Reynolds numbers. The reason for higher pressure drop in the hot stream is already explained in section 4.1. The net gain in overall heat transfer coefficient with increase in Reynolds number is approximately same for all channel diameters under consideration. However, the rise in pressure drop of hot and cold S-CO₂ with increase in Reynolds number is significantly higher for smaller channel diameter PCHEs. As seen in eqn. (12), obtained by replacing $\dot{m}$ in terms of Re in eqn. (11), the pressure drop is proportional to the square of Reynolds number and inversely proportional to third power of channel diameter. Hence, the rise in pressure drop with increase in Reynolds number is significantly larger with small channel diameter. The maximum allowable pressure drop will be the constraint to finalize the channel diameter and operating Reynolds number:

$$\Delta p = \left(\frac{16 f L_{ch} (\pi/2 + 1)^3 \mu^2}{\rho \pi^3} \right) \frac{\text{Re}^2}{D_{ch}^3} \quad (12)$$

4.3.2 Zigzag pitch and channel bend angle

The key geometric design parameters of zigzag channel PCHE are the bend angle and linear pitch. Therefore, the study of zigzag channel PCHE performance for various values of bend angle and linear pitch is carried out and presented in figs. 9 a), 9 b) and 9 c). Figure 9 a) shows that the overall heat transfer coefficient increases with bend angle; the higher bend angle leads to increase in the transverse velocity, which results in better mixing and higher heat transfer coefficient. Furthermore, the overall heat transfer coefficient also decreases with linear pitch. For the same flow length, the larger number of bends for lower linear pitch results in higher heat transfer coefficient. Figures 9 b) and 9 c) show the pressure drop variation with bend angle and linear pitch for cold and hot S-CO₂, respectively. The pressure drop variation is approximately similar to the overall heat transfer coefficient variation: increases with bend angle, while decreases with linear pitch. The reason for this behavior is same as that in the case of the overall

heat transfer coefficient variation. Therefore, increasing the bend angle and decreasing the linear pitch are two ways to augment the overall heat transfer, in addition to reducing the channel diameter and increasing the Reynolds number. The pressure drop increases simultaneously with increase in overall heat transfer coefficient achieved by any of these variations. The channel bend angle and linear pitch should be selected in such a way that it leads to maximum overall heat transfer coefficient, but within the maximum allowable pressure drop.

The correlations for Nusselt number and friction factor for all considered cases are represented in table 5.

4.4 One-Dimensional analysis

The above study leads to a basic understanding of the main predominant parameters for design of a PCHE as regenerator for S-CO₂ Brayton cycle. However, 3D simulations of an entire heat exchanger with multiple channels are computationally expensive. Therefore, 1D thermal-hydraulic model for designing and evaluating the heat exchanger performance, using numerically derived correlations from channel-level CFD simulations, is an attractive alternative. The computational algorithm for the 1D modelling is represented in fig. 10. This methodology of heat exchanger modelling is referred to as sub heat exchanger modelling [20]. In this method, the heat exchanger is divided into various sub heat exchangers having same amount of heat transfer. Initially, the pressure field is guessed to calculate fluid properties which are further iterated until the error is within specified limit. The NTU and UA are evaluated for each sub heat exchanger, and the required length of sub heat exchangers is evaluated after calculation of overall heat transfer coefficient. The pressure field is updated according to the pressure drop in the sub heat exchangers.

The correlations mentioned in table 5 are used for calculation of Nusselt number and friction factor in the sub heat exchangers. In this study, the regenerator is designed for a 1.5 MW net power S-CO₂ Brayton cycle based CSP plant. The thermodynamic state points of the cycle are calculated from an algorithm proposed by Garg *et al.* [3], while neglecting the leakage loss. The state points are shown in fig. 1 and the required values are specified in table 6. It is found that the required heat capacity and mass flow rate of regenerator are 8.9 MW and 15.3 kg/s, respectively.

The maximum allowable pressure drop was fixed for cold side S-CO₂ at 75kPa. Major performance parameters such as temperature variation and pressure drop of hot and cold S-CO₂

with straight and zigzag channel PCHE are shown in figs. 11 a) and 11 b). The zigzag channel length (0.9 m) is much smaller compared to the straight channel PCHE (1.5 m) for transferring same amount of heat, because of its higher heat transfer coefficient. Fig 11 a) shows that the temperature variation is non-linear along the length of heat exchanger channel, which is inconsistent with the above mentioned ANSYS Fluent simulation results. The smaller length of computation domain and the temperature range split (lower and higher temperature range) are the key reasons for the linear variation observed earlier. However, Fig. 11 b) shows that the pressure drop in zigzag PCHE is significantly higher than that in the straight channel PCHE for hot S-CO₂.

If the pressure drop of cold S-CO₂ is allowed to vary, the pressure drop of cold fluid and the heat exchanger volume variation with cold S-CO₂ inlet Reynolds number is obtained as shown in fig. 12. The larger length of channel leads to higher straight channel PCHE volume compared to zigzag channel PCHE. The heat exchanger volume can be reduced by operating in higher Reynolds number regime, but with significantly larger pressure drops.

5. CONCLUSIONS

Performance evaluation of straight and zigzag channel PCHE for S-CO₂ cycle regenerator has been performed. This study leads to following main inferences:

- The temperature variation in the straight and zigzag channel is approximately linear in both lower and higher temperature range study. The pressure drop in the hot S-CO₂ stream is significantly larger (approximately 3 times) than that in the cold stream, because of lower density.
- The Reynolds number variation study suggests that the Nusselt number increases linearly with Reynolds number in both types of channel geometries. The cold S-CO₂ Nusselt number is higher compared to that of hot S-CO₂ because of higher Prandtl number. The Nusselt number varies in the range of 20 to 70 and 25 to 120 in straight and zigzag channel PCHEs, respectively.
- The overall heat transfer coefficient is sensitive to the operating Reynolds number and channel diameter. The smaller diameter results in higher heat transfer coefficient, but the pressure drops also increase with decrease in channel diameter, in inverse cubic proportion.

Similarly, in zigzag channel PCHE, the overall heat transfer coefficient increases with bend angle and decreases with linear pitch, but with a higher pressure drop penalty.

- 1D simulations suggest that the heat exchanger volume can be reduced by operating in a higher Reynolds number regime, but at the cost of higher compressor work.

The regenerator analysis can be further coupled to the S-CO₂ Brayton cycle analysis for selecting appropriate operating conditions and sizing. A smaller heat exchanger will result in lower capital expenditure, but the higher pressure drop will reduce cycle efficiency and will result in lower power production. Hence, a complete thermo-economic analysis is required for arriving at optimized operating conditions and heat exchanger size.

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NOMENCLATURE

c_p	specific heat (kJ/kg-K)
D_{ch}	diameter of the channel (mm)
d_h	hydraulic diameter of channel (mm)
E	energy (J)
f	Fanning friction factor
h	heat transfer coefficient (W/m ² -K)
H_{ch}	height of computational domain (mm)
k	turbulent kinetic energy (J)
k_f	thermal conductivity of fluid (W/m-K)
k_s	thermal conductivity of solid (W/m-K)
k_t	turbulent thermal conductivity (W/m-K)

L_p	linear pitch of zigzag channels (mm)
L_{ch}	channel length (mm)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt Number
p	pressure (kPa)
Pr	Prandtl Number
q	heat flux (W/m ²)
Re	Reynolds Number
Re_{d_h}	Reynolds number based on hydraulic diameter
T	temperature (K)
T_b	bulk fluid temperature (K)
T_w	wall temperature (K)
u_{av}	average velocity (m/s)
u_j	velocity component in j direction (m/s)
V	mean flow velocity (m/s)
W_{ch}	width of computational domain (mm)
y	first cell height (mm)

Greek symbols

α	bend angle (°)
Δp	frictional pressure drop (kPa)
ΔT_{reg}	regenerator approach temperature (K)
δ_{ij}	kronecker delta
ϵ	dissipation rate (J/kg-s)
η_{turb}	turbine isentropic efficiency
η_{comp}	compressor isentropic efficiency
τ_w	wall shear stress (N/m ²)
μ	dynamic viscosity (N/m ² -s)
ν	kinematic viscosity (m ² /s)
ρ	fluid density (kg/m ³)

Subscripts

1	compressor inlet
3	turbine inlet
f	fluid
hf	hot fluid
cf	cold fluid

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CAPTIONS TO TABLES

Table 1: Boundary conditions

Table 2: Computational domain size parameters

Table 3: Boundary conditions for temperature and pressure drop variation along the length study

Table: 4 Mean properties in straight channel

Table: 5 Nusselt and friction factor correlation for various cases

Table 6: Inputs for thermodynamic calculation of regenerator

Table 1: Boundary conditions

Boundary	B.C's
Fluid Inlet	Velocity Inlet
Fluid Outlet	Pressure Outlet
Top wall	Periodic
Bottom wall	Periodic
Side walls	Adiabatic
Front wall	Adiabatic
Back wall	Adiabatic

Table 2: Computational domain size parameters

Variable	Values (<i>mm</i>)
D_{ch}	2
H_{ch}	3.2
W_{ch}	2.5
L_{ch}	200

Table 3: Boundary conditions for temperature and pressure drop variation along the length study

Property		Cold S-CO ₂		Hot S-CO ₂	
		Inlet	Outlet	Inlet	Outlet
Straight channel					
Lower	T(K)	400	-	630	-
Temperature	p(bar)	-	225	-	90
range	V(m/s)	0.842	-	4.702	-
Higher	T(K)	500	-	730	-
Temperature	p(bar)	-	225	-	90
range	V(m/s)	1.518	-	6.118	-
Zigzag channel					
Lower	T(K)	400	-	630	-
Temperature	p(bar)	-	225	-	90
range	V(m/s)	0.806	-	1.345	-
Higher	T(K)	500	-	730	-
Temperature	p(bar)	-	225	-	90
range	V(m/s)	1.876	-	7.564	-

Table: 4 Mean properties in straight channel

Property	Lower Temperature range		Higher Temperature range	
	Cold S-CO ₂	Hot S-CO ₂	Cold S-CO ₂	Hot S-CO ₂
<i>T (K)</i>	438.47	554.15	558.71	670.1
<i>p (bar)</i>	225.07	90.16	225.08	90.26
<i>c_p (kJ/kg-K)</i>	1.59	1.14	1.28	1.16
<i>ρ (kg/m³)</i>	342.48	88.87	223.65	71.33
<i>μ (μPa-s)</i>	31.24	27.14	30.90	31.26
<i>k_f (mW/m-K)</i>	45.91	40.70	47.50	49.16
<i>Pr</i>	1.08	0.76	0.83	0.74

Table: 5 Nusselt and friction factor correlation for various cases

Case	Nusselt number	Friction factor
Straight channel		
Hot Fluid (500K < T _b < 630K)	$Nu = 0.0493 Re^{0.77} Pr^{0.55}$	$f = 0.8386 Re^{-0.5985} + 0.00295$
Hot Fluid (600K < T _b < 730K)	$Nu = 0.0514 Re^{0.76} Pr^{0.55}$	$f = 0.8385 Re^{-0.5978} + 0.00331$
Cold Fluid (400K < T _b < 500K)	$Nu = 0.0718 Re^{0.71} Pr^{0.55}$	$f = 0.8657 Re^{-0.5755} + 0.00405$
Cold Fluid (500K < T _b < 600K)	$Nu = 0.0661 Re^{0.743} Pr^{0.55}$	$f = 0.8796 Re^{-0.5705} + 0.00353$
Zigzag channel		
Hot Fluid (470K < T _b < 630K)	$Nu = 0.0174 Re^{0.893} Pr^{0.7}$	$f = 0.867 Re^{-0.522} + 0.040$
Hot Fluid (580K < T _b < 730K)	$Nu = 0.0205 Re^{0.869} Pr^{0.7}$	$f = 0.819 Re^{-0.671} + 0.044$
Cold Fluid (400K < T _b < 520K)	$Nu = 0.0177 Re^{0.871} Pr^{0.7}$	$f = 0.869 Re^{-0.512} + 0.041$
Cold Fluid (500K < T _b < 640K)	$Nu = 0.0213 Re^{0.876} Pr^{0.7}$	$f = 0.804 Re^{-0.711} + 0.045$
Straight channel diameter variation		
Hot Fluid (500K < T _b < 630K)	$Nu = 0.0685 Re^{0.705} (D_{ch}/2)^{-0.122}$	$f = 0.0648 Re^{-0.254} (D_{ch}/2)^{-0.0411}$
Cold Fluid (400K < T _b < 500K)	$Nu = 0.0117 Re^{0.843} (D_{ch}/2)^{0.0405}$	$f = 0.0759 Re^{-0.241} (D_{ch}/2)^{0.089}$
Zigzag pitch and turning angles		
Hot Fluid (470K < T _b < 630K)	$Nu = 87.56 \left(\frac{L_p}{12} \right)^{-0.178} \left(\frac{\alpha}{116} \right)^{-0.9306}$	$f = .0375 \left(\frac{L_p}{12} \right)^{-0.9639} \left(\frac{\alpha}{116} \right)^{-2.409}$
Cold Fluid (400K < T _b < 520K)	$Nu = 85.95 \left(\frac{L_p}{12} \right)^{-0.171} \left(\frac{\alpha}{116} \right)^{-0.8912}$	$f = .0395 \left(\frac{L_p}{12} \right)^{-0.9479} \left(\frac{\alpha}{116} \right)^{-2.239}$

Table 6: Inputs for thermodynamic calculation of regenerator

Variable	Values
$T_3 (K)$	973
$T_l (K)$	313.15
$p_3 (bar)$	225
$p_l (bar)$	90
$\eta_{turb} (\%)$	83
$\eta_{comp} (\%)$	69
$\Delta p/p_{in} (\%)$	0.5
$\Delta T_{reg} (K)$	5

CAPTIONS TO FIGURES

Figure: 1 Schematic of S-CO₂ Brayton cycles

Figure: 2 Schematic of PCHE Computational domains with a) straight b) zigzag channel

Figure: 3 A small section of a) straight b) zigzag channel PCHE for general mesh representation

Figure: 4 Temperature (left ordinate) and pressure drop (right ordinate) variation of straight channel along the length a) lower temperature range b) higher temperature range

Figure: 5 Temperature (left ordinate) and pressure drop (right ordinate) variation of zigzag channel along the length a) lower temperature range b) higher temperature range

Figure: 6 Nusselt number (left ordinate) and friction factor (right ordinate) variation of straight channel vs. Reynolds number a) lower temperature range b) higher temperature range

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Figure: 8 a) Overall heat transfer coefficient b) pressure drop of hot fluid c) pressure drop of cold fluid vs. channel diameters in straight channel for various Reynolds number (in thousands)

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Figure: 10 Schematic of algorithm for 1D model

Figure: 11 a) Temperature b) pressure drop variation along the length of straight and zigzag channel heat exchanger

Figure: 12 Pressure drop (left ordinate) and heat exchanger volume (right ordinate) variation with cold S-CO₂ inlet Reynolds number with Straight and zigzag channel

Figure: 1 Schematic of S-CO₂ Brayton cycles

Figure: 2 a)

Figure: 2 b)

Figure: 2 Schematic of PCHE Computational domains with a) straight b) zigzag channel

Figure: 3 a)

Figure: 3 b)

Figure: 3 A small section of a) straight b) zigzag channel PCHE for general mesh representation

Figure: 4 a)

Figure: 4 b)

Figure: 4 Temperature (left ordinate) and pressure drop (right ordinate) variation of straight channel along the length a) lower temperature range b) higher temperature range

Figure: 5 a)

Figure: 5 b)

Figure: 5 Temperature (left ordinate) and pressure drop (right ordinate) variation of zigzag channel along the length a) lower temperature range b) higher temperature range

Figure: 6 a)

Figure: 6 b)

Figure: 6 Nusselt number and friction factor variation of straight channel vs. Reynolds number a) lower temperature range b) higher temperature range

Figure: 7 a)

Figure: 7 Nusselt number (left ordinate) and friction factor (right ordinate) variation of zigzag channel vs. Reynolds number a) lower temperature range b) higher temperature range

Figure: 8 a)

Figure: 8 b)

Figure: 8 c)

Figure: 8 a) Overall heat transfer coefficient b) pressure drop of hot fluid c) pressure drop of cold fluid vs. channel diameters in straight channel for various Reynolds numbers (in thousands)

Figure: 9 a)

Figure: 9 b)

Figure: 9 c)

Figure: 9 a) Overall heat transfer coefficient b) pressure drop of hot fluid c) pressure drop of cold fluid vs. linear pitch in zigzag channel for various turning angles

Figure: 10 Schematic of algorithm for 1D model

Figure: 11 a)

Figure: 11 b)

Figure: 11 a) Temperature b) pressure drop variation along the length of straight and zigzag channel heat exchangers

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