Design and Analysis of a Spiral Plate Heat Exchanger on a Producer Gas [Biomass Gasification]

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Abstract— Producer Gas (PG) is derived from Biomass Gasification process. Producer gas is a mixture of gases which consists of hydrogen, carbon monoxide, methane, carbon dioxide, water vapour, nitrogen, tar and suspended particulate matter. For satisfactory IC engine operation, an acceptable particle content < 50 〖mg/Nm〗^3and tar content < 100 〖mg/Nm〗^3 is postulated. Tar available in vapor form is separated from Producer Gas by condensation. Hence cooling and cleaning of Producer Gas is an important step for the effective operation of IC engines. Design of spiral plate heat exchanger was created on the basis of heat duty and allowable pressure drop requirement. Geometric model of spiral plate heat exchanger is created by using CREO Parametric 2.0. ANSYS Workbench 16.2 is used for Numerical Analysis (CFD) of geometric model. It is estimated to simulate the model by varying the mass flow rates on both side and by increasing the thermal resistance to heat transfer.

Key words: Biomass Gasification, Tar, Producer Gas, Spiral plate heat exchanger, CFD Analysis

I. Introduction

Gasification is one of the promising technologies to exploit energy from renewable biomass, which is derived from all living matters, and thus is located everywhere on the earth. Biomass gasification means incomplete combustion of biomass resulting in production of combustible gases consisting of Carbon monoxide (CO), Hydrogen (H_2), traces of Methane (〖CH〗_4) and non-useful products like tar and dust. This mixture is called producer gas. Producer gas can be used to run internal combustion engines (both compression and spark ignition).The temperature of Producer Gas coming out of gasifier is usually about 400-500◦C. However, if the gas is to be used in an internal-combustion engine, it must be cooled to prevent pre-ignition, to improve the engine volumetric efficiency, to raise the energy density of gas. As the gas cools, tars begin to condense at temperatures below 150◦C. Tar formation is one of the major problems to deal with because of blocking and fouling process in equipment such as pipes and valves in gas engine. Also the presence of tar will impose serious limitations in the use of producer gas due to engine wear and high maintenance costs.

It is found that no or very little information on using Spiral Plate heat exchangers for the Tar contaminated Producer gas cooling application, is available in the existing Literature, that for 'both fluids flowing in spiral paths'. Spiral plate heat exchanger is a spirally wound plates screen the geometry of Archimedean spiral.

P. Hasler and Th. Nussbaumer (1) gives that Gas cleaning for tar and particle removal is necessary for Internal Combustion (IC) engine application of Producer Gas and that 90% particle removal is easier to achieve than 90% tar removal. A.G.Bhave et.al (2) they proposed that power generation through internal combustion engines, or for certain thermal applications requiring a clean flue gas, it is necessary to cool biomass-based producer gas to ambient temperature, and clean it of tar and particulates before it can be used as a fuel. Pratik N. Sheth and B.V. Babu (3) in this study, the performance of the biomass gasifier system is evaluated in terms of equivalence ratio, producer gas composition, calorific value of the producer gas, gas production rate, zone temperatures and cold gas efficiency. The effect of equivalence ratio on cold gas efficiency is comparatively lower for higher values of equivalence ratio. R.N. Singh et.al. (4) In this study, they make a review on the different tar removal methods for Producer Gas. The tar present in producer gas may create problem, if the tar content in the producer gas is above 50-100mg/Nm^2.Gas cleaning and conditioning systems to control tar levels are being continuously modified for better efficiency and cost effectiveness. Cristopher O. Akudo and Chandra S Theegala (5) they have analysed the sampled impurities from the syngas by mass gravimetric, solvent evaporation and weight differential methods. The average tar and particulate concentrations of sample run was 1.8 to 3.1g/m^3and 5.2 to 6.4g/m^3respectively. Also they have given the higher heating value range from 4.38 to 4.55MJ/Nm^3. Dr. M. S. Tandale & S. M. Joshi [6] provided analytical model to design of spiral tube heat exchanger for waste heat recovery from Producer Gas and its experiments were performed. The experimental results show that the deviation between calculated values of overall heat transfer coefficient from the experimental results and theoretical values obtained from the analytical model are within 12%. Also, the accuracy is found to be within ±8 % in approximation. M. P. Nunez and G. T. Polley [7] presented a method for the sizing of spiral plate heat exchangers. The temperature profiles of the exchanger calculated analytically show the same tendency as those obtained numerically. *R.Rajaved and K. Saravanan* [8] they have investigated the convective heat transfer coefficient for electrolytes using spiral plate heat exchanger. The mass flow rate of water (hot fluid) is 0.636 kg/s and that of electrolytes varied from 0.483 kg/s to 0.704 kg/s. The results of experiment shown that the heat transfer coefficient increases with the increasing in Reynold number of electrolytes. *Probal Guha and Vaishnavi Unde* [9] they have developed a mathematical model to deals with the spiral heat exchanger. The design considerations for spiral heat exchanger is that the flow within the spiral has been assumed as flow through a duct and by using Shah London empirical equation for Nusselt number design parameters are further optimized. *S. Ramachandran and P. Kalaichelvi* [10] Experimental studies were conducted in a spiral plate heat exchanger with hot water as the service fluid and the two-phase system of water – palm oil in different mass flow rates.

II. DESIGN METHODOLOGY

The problem is to design a suitable heat exchange system for a tar contained producer gas cooling and cleaning application that is to reduce the tar quantity from producer gas. The data regarding producer gas composition, temperature, pressure and volume flow rate of gas, is collected from the actual site. Using this data, heat potential available in the producer gas is calculated by using following equation

$$(Q_h) = m_h * C_{ph} * \Delta T \quad (2.1)$$

The temperature of water at the outlet of spiral plate heat exchanger due to the heat absorbed from Producer Gas is obtained by the energy balance equation is as follows

$$m_h * C_{ph} * \Delta T_h = m_w * C_{pw} * \Delta T_w \quad (2.2)$$

Flow arrangement selected is counter flow type and accordingly LMTD for this stage is calculated as,

$$\Delta T_{lm} = \frac{(\Delta T_1 - \Delta T_2)}{ln(\frac{\Delta T_1}{\Delta T_2})} \quad (2.3)$$

As flow is not truly counter current, so assume *f=0.9*

$$\Delta T_m = f * \Delta T_{lm} \quad (2.4)$$

Assume, $U = 50\ W/m^2K$

To get compact Design $H \simeq D_s$

$$A = \frac{Q_h}{U*(\Delta T_m)} \quad (2.5)$$

$$A_p = A = 2*L*H \quad (2.6)$$

$$D_s = [1.28 * L * (d_c + d_h + 2 * t) + C^2]^{0.5} \quad (2.7)$$

For local heat transfer coefficient,

$$D_e = \frac{4*(d_{h,c}*H)}{2*(d_{h,c}+H)} \quad (2.8)$$

$$Re = \frac{2*m}{H\mu_s} \quad (2.9)$$

Critical Reynold Number is given by

$$R_{ec} = 20000*(\frac{D_{eh}}{D_s})^{0.32} \quad (2.10)$$

For, $Re < R_{ec}$ (Spiral flow no phase change laminar)

$$\frac{h}{Gc_p} = 1.86*(Re)^{(\frac{-2}{3})} * (Pr)^{(\frac{-2}{3})} * (\frac{d_h}{D_s})^{\frac{1}{6}} * (\frac{\mu_f}{\mu_b})^{-0.14} \quad (2.11)$$

$$G = \frac{m}{H*d_h} \quad (2.12)$$

$$P_r = \frac{\mu_b*C_p}{K} \quad (2.13)$$

Overall heat transfer coefficient is given by,

$$\frac{1}{U} = \frac{1}{h_c} + \frac{1}{h_h} + f_c + f_h + \frac{t}{K_m} \quad (2.14)$$

$K_m = 16.26\ W/m\ K$ (stainless steel *G316*)

Pressure Drop,

$Re < R_{ec}$ (Spiral flow no phase change laminar)

$$\Delta P = 36.84 * \frac{L}{\rho} * \left[\frac{W}{H*d_s}\right] [\frac{1.035*\mu_f{}^{0.5}}{(d_s+0.0032)} \left(\frac{H}{W}\right)^{0.5} * (\frac{\mu_f}{\mu_b}) + 1.5 + \frac{16}{L}] \quad (2.15)$$

III. CFD ANALYSIS

Parameter	Quantity	Unit
Density(ρ)	1000	Kg/m^3
Specific Heat (Cp)	4.175	KJ/KgK
Bulk Viscosity (μb) $* 10^{-6}$	740	Pa – S
Thermal Conductivity (K) $* 10^{-3}$	621	W/mK

Table 1: Properties of Water

Spiral plate heat exchanger geometry is created in CREO 2.0 environment, Meshing and modelling is created in Meshing Modeller and Design Modeller of ANSYS WORKBENCH 16.2 and is imported into CFX solver to predict the temperature profiles of the spiral plate heat exchanger. Simulations are carried out for 12 cases of fluid system. Coarse mesh is used in the core region and Inflation layer mesh is used for all other parts of the spiral plate heat exchanger. The conformal mesh is used for CFD analysis. The final grid consisted of 2,772,530 nodes and 6,953,330 elements. All calculations are performed in a double precision segregated steady state solver.

Parameter	Quantity	Unit
Density(ρ)	0.9	Kg/m^3
Specific Heat (Cp)	1.312	KJ/KgK
Bulk Viscosity (μb) $* 10^{-6}$	26.91	Pa – S
Thermal Conductivity (K) $* 10^{-3}$	49.52	W/mK

Table 2: Properties of Producer Gas

Sr.No.	Parameter	Value	Unit
1	Total heat transfer area	0.8	m^2
2	Width of the channel plate	100	mm
3	Thickness of the channel plate	1	mm
4	Core diameter of the heat exchanger	75	mm
5	Outer diameter of the heat exchanger	252	mm
6	Channel spacing	5	mm

Table 3: Technical Specifications of SPHE

Case No.	Gas Flow Rate (Kg/s)	Water Flow Rate (Kg/s)	T_{wi} (° C)	Thermal Conductivity (W/m K)
1	0.0011	0.334	30	16.26
2	0.001	0.334	30	16.26
3	0.001	0.167	30	16.26
4	0.001	0.167	40	16.26
5	0.0006	0.167	30	16.26
6	0.0006	0.167	40	16.26
7	0.001	0.334	30	0.125
8	0.001	0.334	40	0.125
9	0.001	0.167	30	0.125
10	0.001	0.167	40	0.125
11	0.0006	0.167	30	0.125
12	0.0006	0.167	40	0.125

Table 4: Cases for CFD Simulation

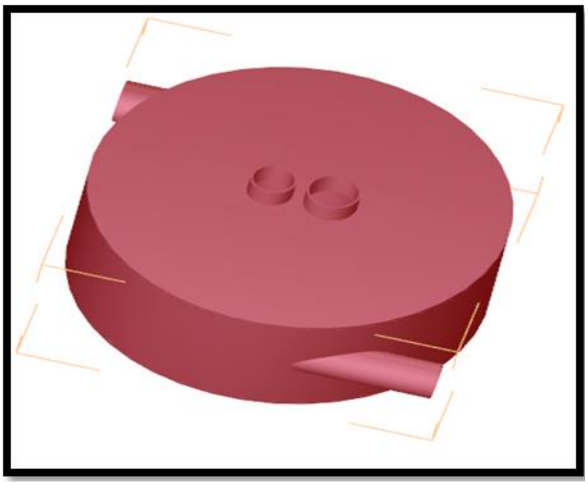


Fig. 1: Geometric Model of spiral Heat exchanger

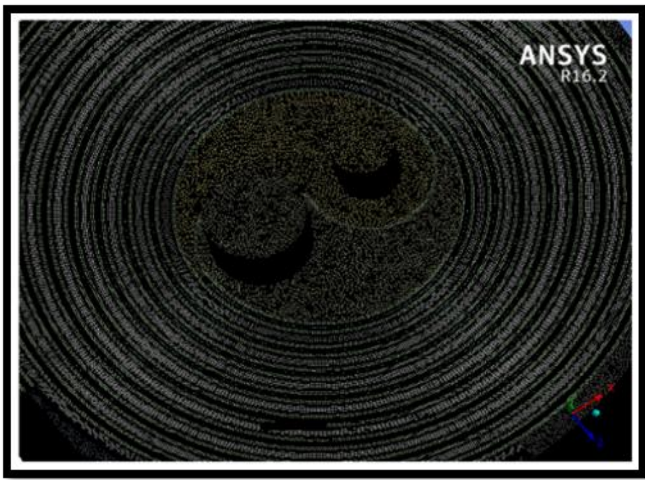


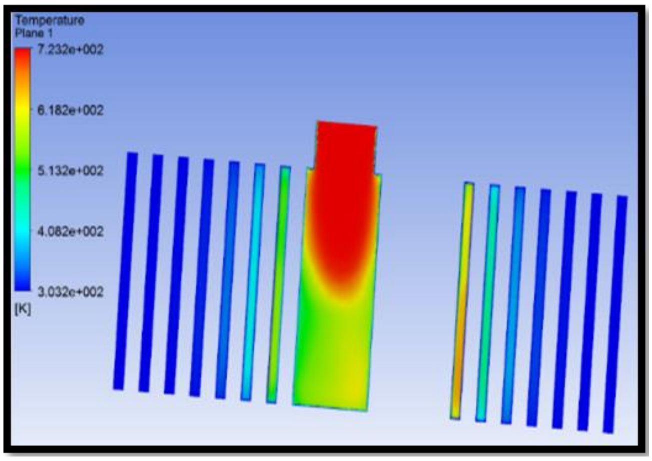
Fig. 2: Mesh Quality of SPHE

In the simulations of flows, for turbulence modelling, namely the k-ε model is used. Properties of the spiral plate material are set to those of stainless steel, with a thermal conductivity of 16.26 W/m K, density of 7881.8 Kg/m^3 and a specific heat of 502 J/kg K. Producer Gas material Model is created by considering constant property gases. Assumption is made that 20 % water vapors at 100◦C and 80 % producer gas concentration in the Producer Gas Material Model. Hence we would able to simulate the actual behavior of particles present in the Producer Gas. Issue regarding the heat transfer effect due to tar vapor condensation on the spiral plates of heat exchanger is resolved by assuming that 0.5 mm tar film is get formed throughout the spiral plate. For hot and cold fluid flow rates, the temperatures at the inlet of the hot fluid producer gas is 450 ° C. Solutions are considered to have converged when the residuals of continuity, components of velocity and energy components are less than 10^{-6}.

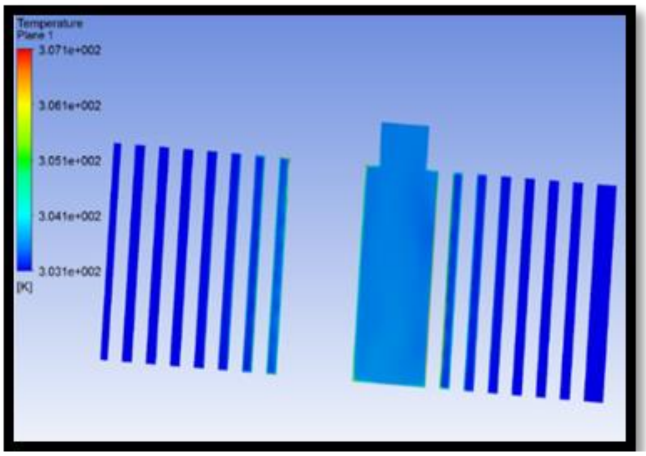
IV. RESULTS AND DISCUSSION

In this section results from the post processing using CFD post is observed and represented. Plots are plotted for parameters which are responsible for performance of the spiral plate heat exchanger. The following are some of the results:

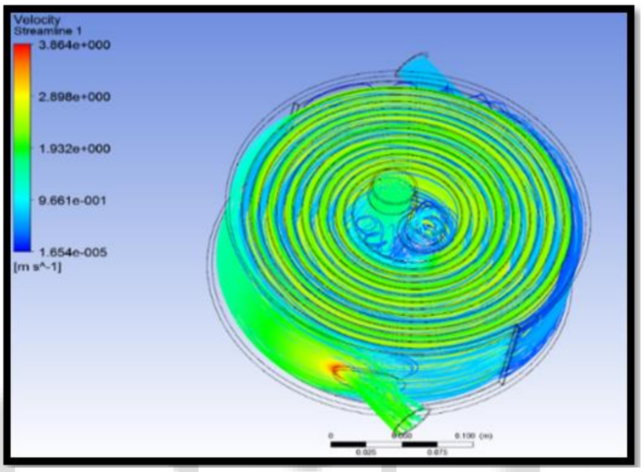
A. *For Case 2*



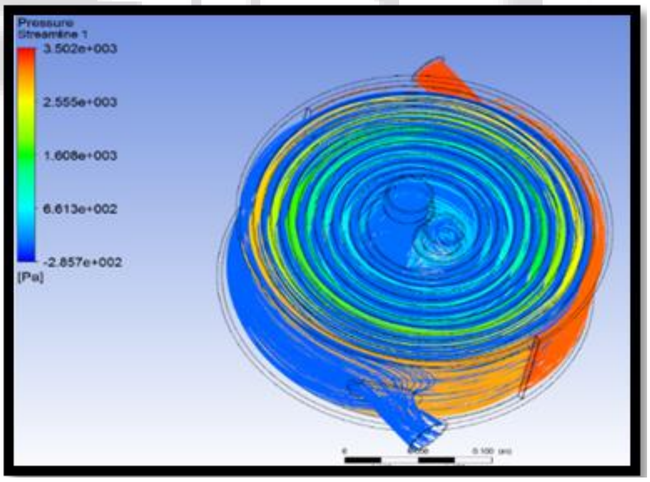
(a) Gas Temperature Plot



(b) Water Temperature Plot



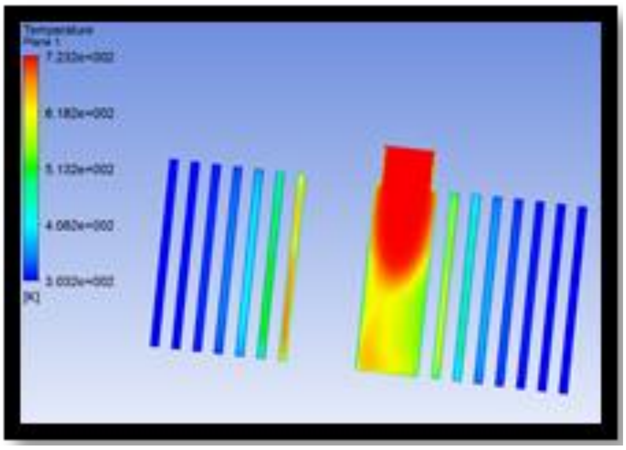
(c) Velocity Plot



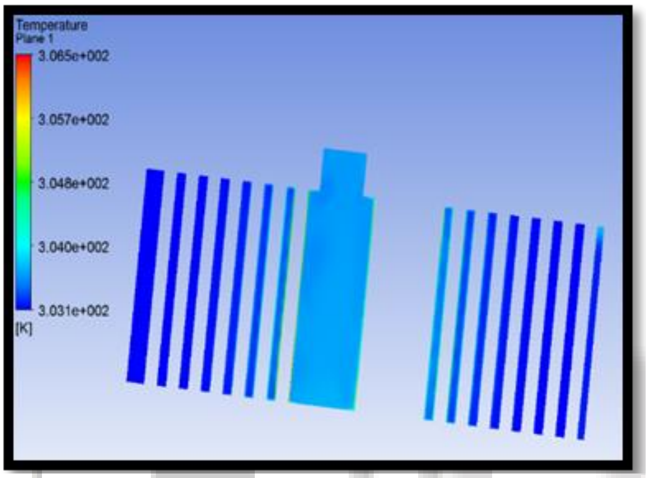
(d) Pressure Plot

Fig. 3: Performance Parameters plot for Case 2

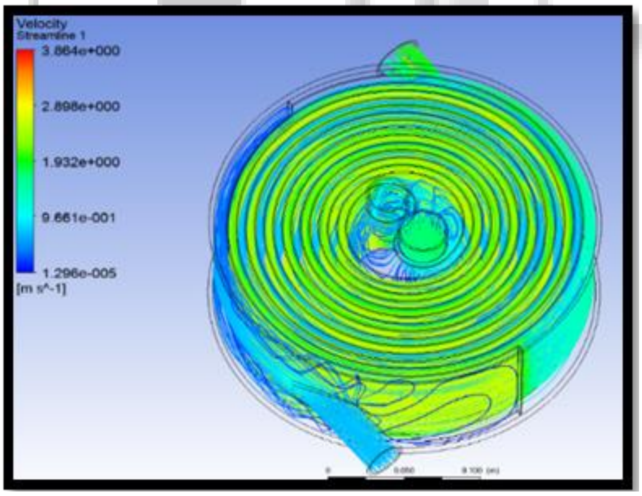
B. For Case 7



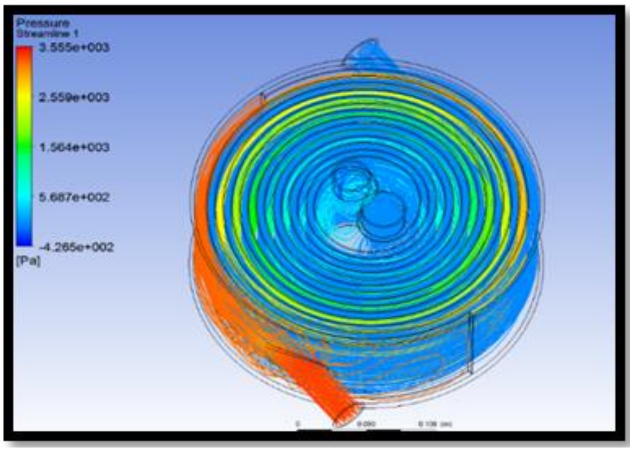
(a) Gas Temperature Plot



(b) Water Temperature Plot



(c)Velocity Plot



(d) Pressure Plot

Fig. 4: Performance Parameters plot for Case 7

The fluid particles were undergoing an uninterrupted rhythm inside the both spiral domain. Along the outer side of the spiral plate the velocity and pressure values were higher in comparison to the inner spiral plate due to the centrifugal force. The shear stress at wall of inner spiral plate is greater than the wall of outer spiral. In addition, the value of overall heat transfer coefficient was found to increase with increase in mass flow rate (i.e. inlet velocity). Here we would come to know that in Cases 7 to 12, the percentage variation in overall heat transfer coefficient is less than ± 10 %.

Cases	Empirical (U) W/m^2K	Simulation (U) W/m^2K	% Variation
1	39.78	54.76	27.35
2	38.76	51.47	24.69
3	38.26	51.14	25.18
4	38.26	51.09	25.32
5	32.08	36.28	11.57
6	32.08	36.28	11.57
7	26.60	28.23	5.77
8	26.60	28.22	5.78
9	26.37	28.18	6.41
10	26.37	28.18	6.41
11	23.82	22.96	-3.74
12	23.82	22.96	-3.74

Table 5: Overall Heat Transfer Coefficient (U) In Different Cases

V. CONCLUSION

The distribution of flow inside the spiral domains is uniform in all of the cases which is close enough to the literature results. The contour plots of temperature, velocity and pressure obtained in different cases are similar to those obtained in literature. The overall heat transfer coefficient in Case 2 and Case 7 is high, in these cases mass flow rate of both the fluid is same and high as compared to others hence high Reynolds number leads to the high heat transfer coefficient for spiral heat exchanger as investigated in literature. Also we could conclude from the contour plots of pressure in different cases, that increasing the feed flow rate for both the domain results into the increase of pressure drop and vice versa which is initial investigated in literature. So all those conclusions validate the developed CFD model of spiral plate heat exchanger.

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NOMENCLATURE

(ρ) Density (Kg/m^3)
C_p Specific Heat at constant pressure (KJ/KgK)
K Thermal Conductivity (W/mK)
μ_f Dynamic Viscosity at film temperature *(Pa-s)*
μ_b Dynamic Viscosity at bulk temperature *(Pa-s)*
ΔP Pressure Drop
U Overall heat transfer Coefficient (W/m^2K)
Q Heat Transfer rate *(W)*
ΔT Temperature Drop
LMTD Log mean temperature Difference
h Local heat transfer coefficient (W/m^2K)
A_p Provided Area of heat transfer (m^2)
A_r Required area of heat transfer (m^2)
Re Reynold Number
Re_c Critical Reynold Number
Pr Prandtl Number
m Mass flow rate (Kg/s)
G Mass Velocity of Fluid (Kg/m^2s)
T Temperature $(\circ C)$
K_m Effective thermal conductivity (W/mK)

SUBSCRIPTS

i Inlet
o Outlet
h Hot side
c Cold
a Ambient
w Water

ABBREVIATIONS

CFD Computational Fluid Dynamics
SPHE Spiral Plate Heat Exchanger.

