ISSN : 0974 - 7443 Volume 11 Issue 2
CHEMICAL TECHNOLOGY
An Indian Journal
Full Paper
CTAIJ 11(2) 2016 [073-084]

Optimal exchanger selection and performance analysis for helium-water heat transfer in fusion energy applications

Karishma Anand[1], Rupsha Bhattacharyya[2]*, K.C.Sandeep[2]
[1]Department of Chemical Engineering, Banasthali University, Rajasthan - 304 022, (INDIA)
[2]Heavy Water Division, Bhabha Atomic Research Centre, Mumbai - 400 085, Maharashtra, (INDIA)
E-mail: rupshabhattacharyya1986@gmail.com

ABSTRACT

Plate heat exchangers (PHEs) and double pipe heat exchangers (DPEs) are possible alternatives for gas-liquid heat transfer applications, particularly for relatively low heat duties of the order of few hundred Watt. One such application is cooling of the helium coolant gas used for heat removal from the first wall of tritium breeder blankets in fusion reactors, before the gas is purified. This work illustrates the selection of an optimal heat exchanger out of PHEs and DPEs for this application based on the total exchanger cost and floor space requirement. Heat transfer area requirements and pressure drop are predicted using appropriate Nusselt number and friction factor correlations. Preliminary cost estimations are made by considering the purchase cost of each unit and the cost of electrical energy required for fluid pumping. Thus the most economical PHE and DPE are identified. The PHE and DPE are then compared on the basis of space requirements. Effects of various design and operating parameters on the total cost of the optimal exchanger type have been evaluated. Heat exchanger effectiveness and exergy analyses have also been performed as part of the rating exercise for the optimal exchanger.

KEYWORDS

Plate heat exchanger;
Double pipe exchanger;
Effectiveness;
Cost analysis;
Exergy analysis.

INTRODUCTION

Plate heat exchangers (PHEs), also called plate and frame heat exchangers are a kind of compact heat exchangers which are quite extensively used in the chemical process industries. They represent a very good alternative, both with respect to the heat transfer area required and the floor space requirements to the more conventional shell and tube heat exchanges, especially when gas-gas or gas-liquid heat transfer has to be carried out. In the field of fusion energy production (e.g. the ITER project[1]), there is an associated tritium extraction system and a coolant purification system for every tritium breeder blanket system[2]. The first wall of the breeder blanket i.e. the structural elements facing the plasma chamber must be cooled by high pressure helium gas which removes the thermal energy generated during fusion. Helium must then be purified and freed of contaminants like moisture, hydrogen isotopes, oxygen and nitrogen. For this it is first necessary to depressurize it and cool it down to about ambient conditions. In such systems associated with the fusion energy sector, which have to be accommodated within limited spaces inside glove boxes to ensure radiation protection and safety in operation, the heat exchange between gas-liquid streams is best carried out in compact heat exchangers, owing to the high heat transfer coefficients and consequently the much reduced heat transfer area and floor space requirements. Therein lies the importance of PHEs for the fusion energy field. For relatively low heat duties such as in the application considered in this study, the more conventional double pipe heat exchangers (DPEs) also represent a possible class of exchangers that may be used[3] and in fact they may be economically competitive with PHEs for gas-liquid service.

In this work, a comparison is made between plate heat exchangers and double pipe heat exchangers for heat duties of upto a few hundred Watt, considering heat transfer between helium and cooling water streams as the case study. Heat exchanger sizing calculations have been performed using Nusselt number correlations specific to each kind of heat exchanger. Economic analysis has been performed to arrive at the optimal PHE and optimal DPE configuration for the service considered in this study and finally the floor space requirements have been compared for selecting the appropriate exchanger for this service. For the optimal exchanger, effectiveness analysis and exergy calculations have also been done to study its performance under conditions different from the design case.

DATA FOR HEAT EXCHANGER SIZING

The base-case stream data as well as the major heat exchanger design parameters considered in this work for the purpose of cost based optimization are presented in TABLE 1. The thermophysical properties of water and helium used in the calculations are shown in TABLE 2. The Nusselt number and friction factor correlations as well as the cost equations used for the two types of heat exchanger are shown in TABLE 3.

Heat exchanger sizing and parametric cost analysis

The major steps followed for sizing the heat exchangers are shown in TABLE 4.

For double pipe exchangers the procedure was almost similar, except that the exchanger dimensions were first assumed (e.g. the exchanger length, inner and outer tube diameters) and the overall heat transfer coefficient was calculated using correlations in TABLE 3. The value of the area was then calculated as in step 3 of TABLE 4 and compared with the value assumed initially. If the difference in area was more than 10%, the assumed length was

TABLE 1 : Base-case stream data and heat exchanger parameters[4, 5, 6]

Parameter	Value
Helium flow rate	34 $Nm^3 hr^{-1}$
Helium inlet temperature	80^oC
Helium outlet temperature	30^oC
Water inlet temperature	25^oC
Water outlet temperature	35^oC
Material of construction of exchangers	Stainless Steel
For the PHE	
Effective dimensions of plate	0.043 m (W_e) X 0.279 m (L_e)
Plate spacing	0.002, 0.003, 0.004, 0.005 m
Chevron angle	30, 45, 60 degree
Port diameter	20 mm
Plate thickness	6 mm
For the DPE	
Inner tube	½’’, ¾”, 1” tube, 18 BWG in each case
Outer tube	1”, 1½’’ tube, 18 BWG in each case

TABLE 2 : Thermophysical properties of water and helium[7, 8]

Parameter	Expression / Value
Density of helium (kg m^{-3})	PM_{He}/RT (ideal gas law)
Viscosity of helium (Pa s)	$3.674*10^{-7}*T^{0.7}$
Specific heat capacity of helium (J kg^{-1} K^{-1})	5195
Thermal conductivity of helium (W m^{-1} K^{-1})	$2.682*10^{-3}(1+1.123*10^{-3}P)*T^{(0.71*(1-0.0002P))}$
Density of water (kg m^{-3})	$2.08*10^{-5}(t)^3-6.668*10^{-3}(t)^2+0.04675(t)+999.9$
Viscosity of water (Pa s)	$(21.482[(t-8.435)+?(8078.4+(t-8.435)^2]-1200)^{-1}$
Specific heat capacity of water (J kg^{-1} K^{-1})	$5.2013*10^{-7}t^4-2.1528*10^{-4}t^3+4.1758*10^{-2}t^2-2.6171t+4227.1$
Thermal conductivity of water (W m^{-1} K^{-1})	$0.5692+(t/538)-t^2/133333$

set equal to the calculated length keeping diameters unchanged and calculations were repeated till the results converged. Convergence was attained in about 5 to 6 iterations in each case. Steps 8 and 9 from TABLE 4 were then evaluated for the converged design. Only unfinned tubes have been considered for the DPEs in this work. The results of the converged designs for both PHE and DPE, with various values of the design parameters of the exchanger are presented in TABLES 5 and 6 along with the estimated costs. The most widely used standard Chevron angles for the PHE plates have been used for identification of the exchanger with the least cost. Technically other Chevron angles in the range from 30° to 80° are possible, but they would be non-standard angles which would only raise the purchase cost of the unit.

From the results in TABLES 5 and 6, DPEs have much lower cost compared to PHEs for identical service. But owing to the large heat exchanger lengths required for them, they are not the most compact alternative. From the compactness point of view, the PHE represents the optimal configuration. Moreover for application in the fusion energy field, where the helium gas is likely to be contaminated with radioactive isotopes of hydrogen, a PHE is a better option due to the inherently low fluid hold-up in it. Since in the fusion energy field, availability of floor space represents the most stringent constraint at present for equipment sizing, PHEs are the recommended alternative for this case of helium-water heat transfer. Further analysis reported in this work all pertain to the optimal PHE configuration.

EFFECT OF DESIGN PARAMETERS ON PHE COST

From the standpoint of a heat exchanger design problem, the heat exchanger cost is not only affected by the design parameters of the exchanger but also the operating conditions which may differ from the nominal operating conditions in TABLE 1. This section presents some results, in the form of Figures 1 to 5, demonstrating the effect of various design parameters on the area requirements and hence the total annual cost of the PHE. For parametric analysis of exchanger cost, geometric properties of the PHE like plate spacing, chevron angles, and operating parameters such as helium flow rate, helium inlet temperature and water inlet temperature have been varied.

From Figure 1 it is observed that the cost passes through a broad minimum at chevron angle of 43-48° for a plate spacing of about 2 mm while for the other plate spacing values the cost monotonically increases, displaying relatively low sensitivity to the value of the chevron angle in certain regions. The zones of constant total cost actually reflect the fact that the number of plates in the exchanger is determined by rounding off to the nearest integer. So for several chevron angles, while the actual heat transfer area required for a given duty differs due to the difference in U, in the final configuration they all have the same number of plates and hence the same total available area, and thus the same fixed cost. Hence constant cost sections appear on the cost versus chevron angle curves. Most generally it can

TABLE 3 : Nusselt number, friction factor and purchase cost correlations for PHE and DPE[5, 6, 9, 10, 11]

Parameter	Value / Expression
Friction factor equation for PHE	$\frac{1}{\sqrt{f}} = \frac{\cos\left(\frac{\pi}{2}-\beta\right)}{\left(0.18\tan\left(\frac{\pi}{2}-\beta\right)+0.36\sin\left(\frac{\pi}{2}-\beta\right)+\frac{f_o}{\cos\left(\frac{\pi}{2}-\beta\right)}\right)^{0.5}} + \frac{1-\cos\left(\frac{\pi}{2}-\beta\right)}{\sqrt{3.8 f_1}}$ For $Re_h < 2000$ $f_o = \frac{64}{Re_h}$ and $f_1 = \frac{597}{Re_h} + 3.85$ For $Re_h >= 2000$ $f_o = \frac{1}{(1.8\log_{10}(Re_h) - 1.5)^2}$ and $f_1 = \frac{39}{(Re_h)^{0.289}}$ $Re_h = \emptyset Re$ and $Re = \frac{(2a)u\rho}{\mu}, \emptyset = 1.17$
Nusselt number correlation for PHE	$Nu_h = 0.122\,Pr^{0.333}\left(f Re^2 \sin 2\left(\frac{\pi}{2}-\beta\right)\right)^{0.374}$ $Nu_h = \emptyset Nu$ and $Nu = \frac{h(2a)}{k_f}$
Purchase cost correlation for PHE	$(7000(10.78 * A)^{0.42})$, cost in Dollars
Friction factor equation for DPE	For Re<=2100 $f = \frac{64}{Re}$ For Re>2100 and Re<=20000 $f = \frac{0.316}{Re^{0.25}}$ For Re>20000 $f = \frac{0.184}{Re^{0.2}}$
Nusselt number correlation for DPE	For Re<=2100, 0.6<=Pr<=5 $Nu = 1.86\left(\frac{RePr}{\frac{L}{D}}\right)^{\frac{1}{3}}$ For Re>2100, 0.5<=Pr<=2000 $Nu = \frac{\frac{f}{8}(Re-1000)Pr}{1+12.7\sqrt{\frac{f}{8}}\,(Pr^{0.67}-1)}$
Purchase cost correlation for DPE	$0.851 * 3 * \exp(7.1248 + 0.16\ln(10.78 * A))$, cost in Dollars
Fouling factors	10000 W m^{-2} K^{-1} for helium, 8000 W m^{-2} K^{-1} for water
Chemical Engineering Plant Cost Index (CEPCI) for heat exchangers (April 2015)	607.9
Dollar to Indian Rupee exchange rate (June 2015)	1 $ = Rs 65
Cost of electrical energy for industrial facilities	Rs 12 $kWhr^{-1}$

be said that for a given plate spacing, with increase of chevron angle the value of the heat transfer

TABLE 4 : Heat exchanger sizing algorithm[3, 12]

Step Number	Calculation
1	Calculation of the flow rate of water using stream data in TABLE 1.
2	Calculation of average stream temperatures, LMTD, NTU and heat duty.
3	Calculation of exchanger area required based on an assumed value of U.
4	Calculation of number of plates and number of channels using effective dimensions of plates from manufacturers' specification sheets.
5	Calculation of water side and helium side heat transfer coefficients and friction factors using correlations in TABLE 3 and assumed exchanger layout.
6	Calculation of U using the individual coefficients and fouling factors on each side.
7	Comparison of assumed and calculated U. If the values deviate by more than 10%, assumed value is set equal to the calculated value of U and steps 3 to 7 are repeated till convergence is achieved.
8	Pressure drop calculations are performed for the finalized exchanger configuration using the friction factors calculated.
9	Estimation of capital cost for the exchanger is made using cost equations and operating costs are calculated based on the total electrical power required for pumping the two fluids through the exchanger, assuming the exchanger to operate for 8000 hours per annum.
10	Calculation of exchanger volume.

TABLE 5 : Area and cost estimates for PHE

Chevron angle (deg)	Plate spacing (m)	Area required (m^2)	Pressure drop (Pa)		Total exchanger cost (Rs)	Exchanger volume (m^3)
			Helium side	Water side		
30	0.002	0.036	60937	1059	609353.46	0.0005
30	0.003	0.066	7939	157	678750.39	0.0009
30	0.004	0.116	1304	44	817564.25	0.0016
30	0.005	0.190	292	27	981107.07	0.0030
45	0.002	0.044	27120	483	596805.32	0.0005
45	0.003	0.112	1376	45	817591.07	0.0013
45	0.004	0.208	209	25	1027990.74	0.0027
45	0.005	0.318	61	23	1191408.95	0.0046
60	0.002	0.089	3478	81	752328.09	0.0008
60	0.003	0.209	255	26	1028007.83	0.0021
60	0.004	0.365	51	23	1262578.07	0.0044
60	0.005	0.578	16	22	1530225.20	0.0083

TABLE 6 : Area and cost estimates for DPE

Inner tube size	Outer tube size	Length (m)	Area required (m^2)	Pressure drop (Pa)		Total exchanger cost (Rs)	Exchanger volume (m^3)
				Helium side	Water side		
Helium in inner tube, water in the annulus							
½”	1”	4.31	0.172	6536	39	354478.58	0.002
¾”	1½”	5.31	0.318	809	7	388669.54	0.006
1”	1½”	4.72	0.377	154	25	399239.15	0.0054
Helium in the annulus, water in inner tube							
½”	1”	7.45	0.297	1270	237	384792.90	0.0035
¾”	1½”	14.12	0.845	190	65	454266.90	0.0161
1”	1½”	8.05	0.642	414	10	434867.79	0.0092

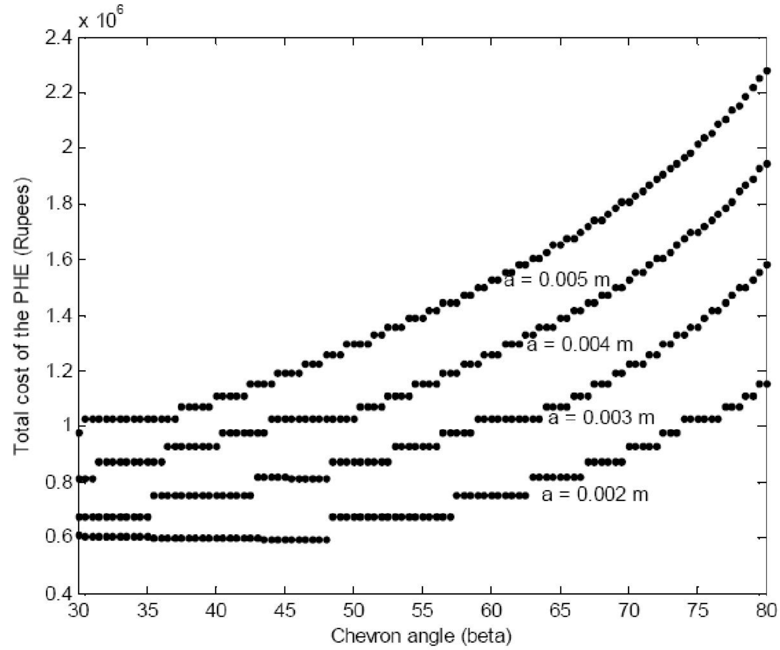

Figure 1 : Effect of plate spacing and chevron angle on total cost of the PHE, (all other fixed parameters as in TABLE 1)

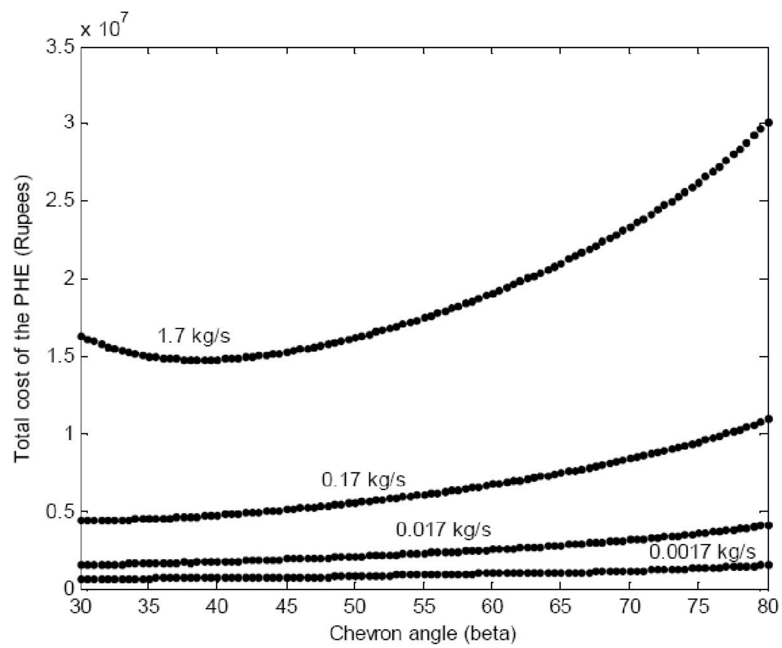

Figure 2 : Effect of helium flow rate and chevron angle on total cost of the PHE, (a = 0.003 m, all other fixed parameters as in TABLE 1)

coefficient decreases, so a higher heat transfer area is required for the given heat duty. For a given chevron angle, increasing the plate spacing reduces the fluid velocities per pass and lowers the heat transfer coefficients, which once again necessitates a higher heat transfer area and consequently a higher exchanger cost. The lowered cost of fluid pumping on increasing the plate spacing does not affect the total cost significantly since the pressure drop and the pumping power cost is a very small fraction of the total cost, owing to the low gas and liquid flow rates considered in this work.

In Figure 2, the effect of increasing the gas flow rate (and consequently the liquid flow rate as well to match the increasing heat duty) has been shown. It is seen that at a helium flow rate of 1.7 kg s^{-1}, a

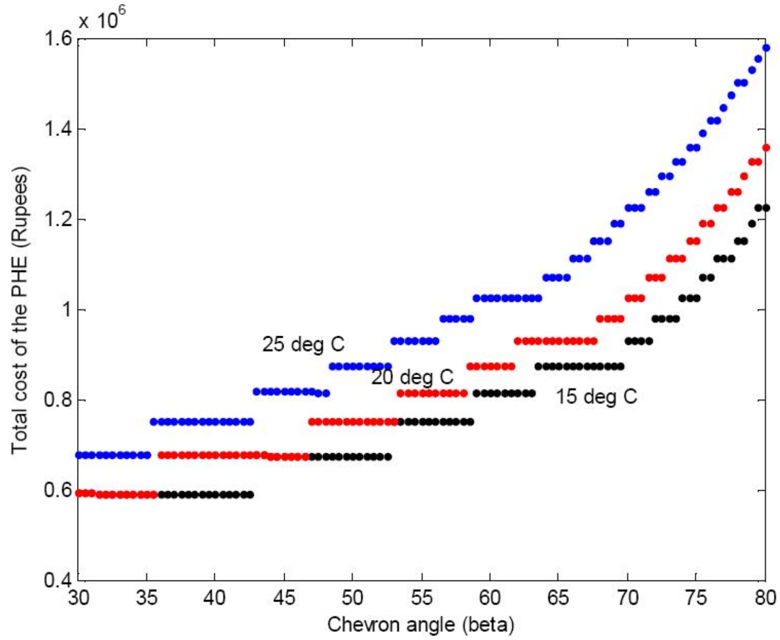

Figure 3: Effect of water inlet temperature and chevron angle on total cost of the PHE, (a = 0.003 m, all other fixed parameters as in TABLE 1)

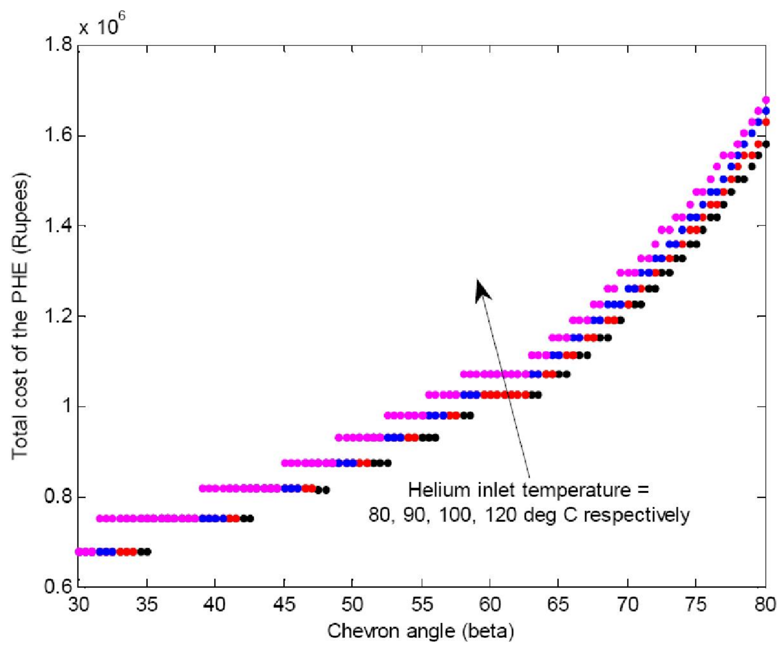

Figure 4: Effect of helium inlet temperature and chevron angle on total cost of the PHE, (a = 0.003 m, all other fixed parameters as in TABLE 1)

minimum appears in the total cost at a chevron angle of about 37^{o} for a plate spacing of 3 mm. At a given flow rate, with increase of chevron angle, the heat transfer coefficient and the friction factor (hence the pressure drop) both decrease. When flow rates and hence heat transfer coefficients and pressure drops are low, in general the total cost increases monotonically with increasing chevron angle, since fluid pumping costs are not significant and the total cost is controlled by the the fixed cost. But when flow rates are high, fluid pumping costs are comparable with the fixed cost of the exchanger and upto a certain value of the chevron angle, the lowered pumping cost outweighs the enhanced area requirements and hence the total cost is seen to decrease. Beyond this angle the rise in fixed cost

Full Paper

is greater than that in the pumping cost and the total cost again increases with chevron angle. It is also possible that the total cost decreases with chevron angle, if fluid pumping cost is significantly greater than the fixed cost of the exchanger. In arriving at the results of Figure 3, the size of each plate was kept the same for all cases, to ensure uniformity in cost comparison. But for an actual heat exchanger handling these much enhanced fluid flow rates, larger plates, based on the available standard sizes will actually have to be used, so that the required heat transfer area is obtained with fewer number of plates, which in turn implies higher channel velocity of the fluids, a larger U and lower cost.

Figure 3 shows that lower the cooling water inlet temperature, lower is the exchanger cost since a greater temperature difference driving force is available for heat transfer to take place. In Figure 4, the change in the inlet temperature of helium is seen to affect the total exchanger cost to a much lower extent than the change in the cooling water inlet temperature. Higher helium inlet temperature at a given flow rate and a fixed outlet temperature indicates a higher heat duty but also a higher mean temperature difference for heat transfer. Thus the area requirements are not dramatically increased with rise in helium temperature. In Figure 5, the effect of selecting different plates for the same duty (i.e. different effective length and width) has been demonstrated. For a given heat duty, if a plate of larger effective area is selected, fewer number of thermal plates will be required and thus the number of channels for fluid flow will be reduced. This leads to increase in the fluid velocities per channel and ultimately higher overall heat transfer coefficients and thus lower total cost.

ANALYSIS OF THE OPTIMAL PHE

PHE rating analysis

Based on floor space constraints as well as minimum total cost, the PHE with a plate spacing of 0.002 m and chevron angle of 45°C has thus been identified as the most optimal configuration for helium water heat transfer, under the base case operating conditions. In this section results of heat exchanger rating analysis have been presented for the cost optimized PHE. The well known effectiveness (ε)-NTU method of analysis has been used for exchanger rating, details of which are very well documented in literature[3, 11]. Typically in such an analysis the value of U is taken as a constant while evaluating the exchanger performance. In this study, for every set of inlet conditions, the value of U was

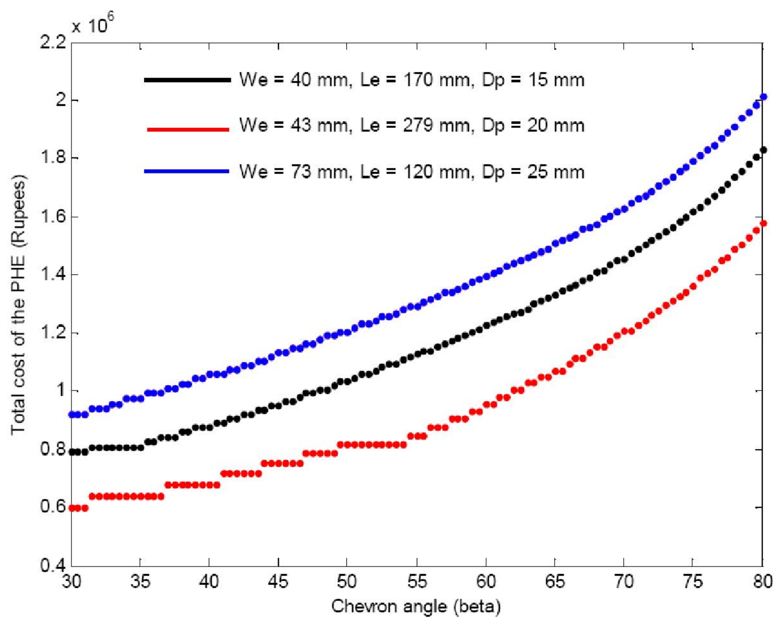

Figure 5 : Effect of plate and port dimensions and chevron angle on total cost of the PHE, (a = 0.003 m, all other fixed parameters as in TABLE 1)

first calculated using values of the fluid physical properties at the inlet temperature, after which the NTU and then effectiveness were evaluated. Once the exit temperatures were calculated for both streams, the calculations were repeated by evaluating U at the arithmetic mean temperature of each fluid and then again evaluating the exit temperatures. The process was repeated till convergence, which was attained in about 3 to 4 iterations for each condition. Flow rate variations to the extent of ± 20% of the design value and inlet temperatures upto 40°C greater than the design values have been considered for the exchanger's performance analysis. Such variations are expected during normal operation of the exchanger, owing to variations in the fusion reactor's performance.

Exergy analysis of the PHE

The transfer of heat from the hot helium stream to the cooling water stream takes place across a finite temperature difference driving force, thereby making the heat transfer process an irreversible one. Moreover there are frictional pressure losses on both sides as the fluids flow through the exchanger[13, 14]. All these irreversibilities contribute to exergy destruction or 'lost work' during operation of the heat exchanger. In this section, the extent of irreversibility and the thermodynamic second law efficiency or exergetic efficiency of the optimal PHE are estimated.

For a substance that undergoes a change from state 1 to 2, the change in exergy can be written as follows:

$$\psi_2 - \psi_1 = (h_2 - h_1) - T_o(s_2 - s_1) \quad (1)$$

For ideal gases and liquids the change in enthalpy from state 1 to 2 is calculated as

$$(h_2 - h_1) = \int_{T_1}^{T_2} C_p dT \quad (2)$$

For an ideal gas undergoing a non isothermal and non isobaric process, the change in entropy is given by Equation 3 while for an incompressible fluid like a liquid, the entropy change is calculated from equation 4. In this study helium has been considered as an ideal gas and water has been taken as an incompressible fluid. Specific heat data used for these calculations have been obtained from TABLE 2. The pressure drop on the water side is negligible as observed from TABLE 5, hence the frictional component of exergy loss of the water stream has not been considered. Frictional pressure drop and the resultant exergy destruction on the helium side have been taken into account, along with the thermal component of exergy loss.

$$(s_2 - s_1)_{helium} = \int_{T_1}^{T_2} \frac{C_p}{T} dT - R \ln\left(\frac{P_2}{P_1}\right) \quad (3)$$

$$(s_2 - s_1)_{water} = \int_{T_1}^{T_2} \frac{C_p}{T} dT \quad (4)$$

The second law efficiency of the heat transfer process is expressed as the ratio of the exergy gained by water to the exergy lost by helium in the heat exchanger, Equation 5, assuming adiabatic conditions.

$$\eta_{2nd\ law} = \frac{m_w\left(\psi_{w_{out}} - \psi_{w_{in}}\right)}{m_h\left(\psi_{h_{in}} - \psi_{h_{out}}\right)} \quad (5)$$

The irreversibility of the process, per unit mass flow rate of cooling water is calculated from Equation 6:

$$I = -\left(\psi_{w_{out}} - \psi_{w_{in}}\right) + \frac{m_h}{m_w\left(\psi_{h_{in}} - \psi_{h_{out}}\right)} \quad (6)$$

The effect of changing flow helium flow rate and the helium inlet temperature on the exchanger's thermal performance and the extent of irreversibility in the process have been studied and results have been reported in TABLE 7.

From TABLE 7 it is observed that in the PHE, for a fixed helium flow rate, as its inlet temperature increases and water inlet temperature and flow rate remain unchanged, the exchanger effectiveness increases and the second law efficiency of the process decreases. The heat transfer from the helium takes place across a higher temperature difference and hence increases the exergy destruction and process irreversibility, leading to a lowering of the exergetic efficiency. For a fixed helium inlet temperature, as its flow rate increases, the effectiveness decreases and second law efficiency increases. When the helium flow increases, the exit temperatures of the helium and water streams also

Full Paper

TABLE 7 : Results of the PHE rating and exergy analysis (T_o = 293 K)

Helium flow rate (kg s^{-1})	Helium inlet temperature (°C)	Cooling water flow rate (kg s^{-})	Cooling water inlet temperature (°C)	NTU	ε	Helium outlet temperature (°C)	Cooling water outlet temperature (°C)	Process Irreversibility per kg water (J kg^{-1} water)	Process second law efficiency (%)
0.0017	90	0.0106	25	3.9608	0.9661	27.20	37.51	4052	32.52
0.0017	100	0.0106	25	3.9754	0.9665	27.51	39.44	5264	31.63
0.0017	110	0.0106	25	3.9898	0.9669	27.81	41.37	6604	30.98
0.0017	120	0.0106	25	4.0041	0.9673	28.11	43.29	8064	30.51
0.00136	90	0.0106	25	4.4310	0.9797	26.32	35.14	3380	29.68
0.00136	100	0.0106	25	4.4461	0.9799	36.51	36.71	4398	28.67
0.00136	110	0.0106	25	4.4610	0.9802	26.69	38.27	5524	27.91
0.00136	120	0.0106	25	4.4758	0.9804	26.86	39.83	6753	27.35
0.00204	90	0.0106	25	3.5668	0.9488	28.33	39.74	4679	34.96
0.00204	100	0.0106	25	3.5813	0.9493	28.79	42.01	6069	34.19
0.00204	110	0.0106	25	3.5956	0.9499	29.26	44.29	7604	33.65
0.00204	120	0.0106	25	3.6099	0.9505	29.71	46.58	9274	33.26
0.0017	90	0.0106	15	3.8811	0.9639	17.71	29.39	5606	7.69
0.0017	100	0.0106	15	3.8962	0.9643	18.03	31.32	7023	9.59
0.0017	110	0.0106	15	3.9112	0.9648	18.35	33.25	8579	11.13
0.0017	120	0.0106	15	3.9261	0.9652	18.66	35.18	10245	12.41

increase, thereby giving rise to higher process irreversibility, but the gain in exergy of the water stream rises as its flow rate and inlet temperature remain unchanged, thereby raising the exergetic efficiency of the process. Not considering the frictional loss of exergy on the helium side leads to calculated second law efficiency greater by about 1 to 2% compared to the values in TABLE 7. The cooling water is obtained from the ambient and its inlet temperature may vary seasonally. At a temperature of 15°C, which is lower than the dead state temperature of 20°C considered here, it is seen that exergetic efficiency drops drastically, owing to much larger exergy loss by helium and relatively low exergy gain by the water stream.

SUMMARY AND CONCLUSIONS

Comparative total cost analysis has been used as the basis for optimizing PHEs and DPEs for the case of helium water heat transfer. This has practical relevance to the field of fusion energy. The most simple exchanger types have been studied here to allow a preliminary identification of the optimal type. The cost was estimated from the thermal design of the exchanger without detailed mechanical design considerations. It was determined that while the cost of the DPE is much lower than that of the PHE for a given service, the space requirement is much larger for the DPE and hence it is not the optimal choice for fusion energy applications where space requirements appear as the most stringent constraint. It is expected that use of finned inner tubes will reduce the DPE dimensions but also increase the cost slightly. Thus the PHE was chosen as the optimal exchanger for helium water service and its design parameters were selected on the basis of minimum total annual cost. For the optimal PHE, performance analysis for operating conditions other than the design values was carried out using the ε-NTU method. True counter flow characteristics of the exchanger were assumed for it. Effectiveness values were quite high, ranging from 0.94 to 0.98 for the conditions examined. Exergy analysis was also performed for the exchanger, and exergetic efficiency was found to lie between 9 and 35%, assuming adiabatic operation.

Nomenclature

a	Plate spacing (m)
A	Heat exchanger area (m^2)
C_p	Specific heat capacity (J kg^{-1} K^{-1})
D	Tube diameter (m)
D_p	Port diameter for PHE (m)
f	Friction factor
f_o, f_1	Constants for friction factor correlation
h	Convective heat transfer coefficient (W m^{-2} K^{-1})
h_1, h_2	Specific enthalpy of a fluid stream at state 1 and 2 respectively (J kg^{-1})
I	Process irreversibility per unit mass of water (J kg^{-1})
k_f	Fluid thermal conductivity (W m^{-1} K^{-1})
L	Length of heat exchanger (m)
L_e	Effective length of heat exchanger plate (m)
m_h	Helium mass flow rate (kg s^{-1})
m_w	Water mass flow rate (kg s^{-1})
M_{He}	Helium molecular weight (kg mol^{-1})
Nu	Nusselt number
Nu_h	Modified Nusselt number for PHE
NTU	Number of transfer units in the heat exchanger
P	Fluid pressure (bar)
P_1, P_2	Fluid pressure at state 1 and 2 respectively (bar)
Pr	Prandtl number
R	Gas constant for helium (J kg^{-1} K^{-1})
R_e	Reynolds number
R_{eh}	Modified Reynolds number for PHE
s_1, s_2	Specific entropy of a fluid stream at state 1 and 2 respectively (J kg^{-1} K^{-1})
t	Stream temperature (Celsius)
T	Stream temperature (K)
T_o	Ambient temperature (K)
u	Channel velocity in PHE (m s^{-1})
U	Overall heat transfer coefficient in an exchanger (W m^{-2} K^{-1})
ε	Heat exchanger effectiveness
ψ_1, ψ_2	Specific availability of a fluid stream at state 1 and 2 respectively (J kg^{-1})
$\psi_{h\ in}$, $\psi_{h\ out}$	Specific availability of helium at state 1 and 2 respectively (J kg^{-1})
$\psi_{w\ in}$, $\psi_{w\ out}$	Specific availability of water at state 1 and 2 respectively (J kg^{-1})
$\eta_{2nd\ law}$	Second law thermodynamic efficiency
ρ	Fluid density (kg m^{-3})
μ	Fluid viscosity (Pa s)
φ	Surface enhancement factor for PHE

REFERENCES

[1] N.Holtkamp; "An overview of the ITER project", Fusion Engineering and Design, **82(5-14)**, 427-434 **(2013)**.

[2] A.Ciampichetti, F.S.Nitti, A.Aiello, I.Ricapito, K.Liger, D.Demange, L.Sedano, C.Moreno, M.Succi; "Conceptual design of tritium extraction system for the european HCPB test blanket module", Fusion Engineering and Design, **87(5-6)**, 466-471 **(2012)**.

[3] B.K.Dutta; Heat transfer principles and applications, Prentice Hall of India Pvt.Ltd., **(2006)**.

[4] GBS series – brazed plate heat exchangers technical data sheet, Available at http://www.gea-heatexchangers.com/products/plate-heat-exchangers/brazed-plate-heat-exchangers/gbs-series/, last accessed on 03.08.2015.

[5] S.Kakac, H.Liu, A.Pramuanjaroenkij; Heat exchangers: Selection, Rating, and thermal design, 3rd Edition, CRC press, Taylor and Francis Group, **(2012)**.

[6] L.Wang, B.Sunden, R.M.Manglik; Plate heat exchangers: Design, Applications and Performance, WIT Press, **(2007)**.

[7] H.Petersen; "The properties of helium: Density, Specific heats, Viscosity, and thermal conductivity at pressures from 1 to 100 bar and from room temperature to about 1800 K", Report No.224, Danish atomic energy commission research establishment riso, **(1970)**.

[8] J.A.W.Gut, J.M.Pinto, "Modeling of plate heat exchangers with generalized conûgurations", International Journal of Heat and Mass Transfer, **46**, 2571-2581 **(2003)**.

[9] W.D.Seider, J.D.Seader, D.R.Lewin; Product and process design principles – synthesis, Analysis and Evaluation, 2nd Edition, John Wiley and Sons Inc., **(2009)**.

[10] Chemical Engineering, **122(7)**, 64, July, **(2015)**.

[11] F.P.Incropera, D.P.De Witt, T.L.Bergman, A.S.Lavine; Fundamentals of heat and mass transfer, 6th Edition, John Wiley and Sons Inc., **(2006)**.

[12] G.Towler, R.Sinnott, Chemical engineering design - Principles, Practice and economics of plant and process design, Elsevier Inc., **(2008)**.

[13] R.E.Sonntag, C.Borgnakke, G.J.Van Wylen; Fundamentals of thermodynamics, 6th Edition, John Wiley and Sons Inc., **(2005)**.

[14] T.J.Kotas; The exergy method of thermal plant analysis, 1st Edition, Krieger Publishing Company, **(1995)**.