Polymer Hollow Fibre Shell and Tube Heat Exchanger for Liquid-Liquid applications

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ABSTRACT

Paper provides comparison of different heat exchangers which differ in length, number and material of fibres. It also deals with an influence of various arrangement of fibres on the efficiency of heat exchangers (HEs). These have a potential to be used in low temperature and low flow rate applications. Number of Shell and tube Polymer heat exchangers were fabricated with various polymers and tested. The heat transfer performance and pressure drops were studied with hot water at 60 and 80 degrees C flowing through the shell side.

It was observed that heat transfer rates (up to 36 kW), and the overall heat transfer coefficients of these devices were 647-1314 W/ (m2K) for water-water applications, and pressure drops are competitive to conventional metal Shell and Tube heat exchanger.

INTRODUCTION

Polymer micro hollow fibre heat exchangers (PHFHE) can reduce weight up to 50% compared with traditional metal heat exchanger. The small diameters of the fibres have thin walls and large surface area so heat transfer intensity is significantly increased. Applications for the PHFHE have been identified in a number of sectors, as follows: Buildings: hollow membrane fibres for liquid desiccant cooling and non-porous capillaries for air heat recuperation, air heaters and fan-coils; Automotive: car radiators with same thermal power as traditional radiators [1]; Electronics; heat transfer units for cooling compact electronic devices; water desalination; air humidification by pervaporation through hollow fibre membranes; Energy Storage: non-porous hollow fibres for encapsulating PCMs can enhance heat transfer for passive cooling and energy storage applications. The implementation of such micro-fibre technology will offer cost effective and recyclable materials with significant reduction in energy consumption and carbon emission.

Heat exchangers are mainly made from metal. Polymer materials provides many advantages in comparison with metal. Easy shaping, chemical resistance, lower density, and lower price are the most important of them. The energy consumed for producing one amount of the polymer is two times lower than to produce same amount of commonly used metal as stainless steel or aluminium [2]. Their main disadvantages are low temperature limit (around 100 °C) and low thermal conductivity around 0.1-0.4 W/ (m K) which is 100-300 times lower than for metals [3].

The PHFHE [4] is a novel technology with the potential not only to significantly improve the current products but to enable entirely new applications and markets. The specific objectives leading to production are characterized as: design, testing and technology. This new approach to the enhanced functionality of polymeric fibre materials used in heat transfer surfaces already has the potential in its current development stage to be competitive in large segments of commercial heat exchangers concerning cost [5], weight, mass scalability, recyclability, resistance to corrosion [6] and low fouling. A previous study showed that small-diameter fibres have a superior overall heat transfer coefficient [7], smooth surface and low surface energy, causing chemical inertness and low adhesion [8]. Polymer fibres are also flexible enough to move in a flow reducing heat exchanger fouling.

PHFHEs are a type of thin-wall polymer heat exchanger which were first proposed by Zarkadas [9] as a useful alternative for lower temperature applications. Small devices containing several hollow PP-based fibres with the liquids in parallel flow at temperatures of up to 74 °C were studied. The overall heat transfer coefficients of these devices were 647-1314 W/ (m2K) for water-water applications. Furthermore, the proposed heat exchangers had very low values of height of a transfer unit (HTU), large numbers of transfer units (NTUs) for comparably short devices, and high values of heat exchanger effectiveness. Their other advantage is quick response to the change of the flow rate. Therefore, they are suitable for temperature control [10]. PHFHEs achieve large area density (The ratio of the heat transfer surface area of a heat exchanger to its volume).

The main goal of this paper is to study and compare thermal performance of polymer heat exchangers with different fibre length, diameters and materials with the aim to develop completely new product for variety of industrial heating and cooling applications to transfer heat between corrosive fluids such as aqueous and semi-aqueous cleaning which is normally carried out at elevated temperature to increase the rate of chemical reaction

NOMENCLATURE

A	[m2]	Heat transfer surface area
A_s	[m2]	Shell free flow area
$A_{channel}$	[m2]	Channel Area
C_p	[J/kg K]	Specific heat capacity
C_{min}	W/K	Heat Capacity rate ,minimum
C_{max}	W/K	Heat Capacity rate, maximum
Di ,	[m]	Fibre inside diameter
Do	[m]	Fibre outside diameter
Dis	[m]	Shell inside diameter
DH_{fiber}	[m]	Fibre Hydraulic diameter
DH_{shell}	[m2]	Shell Hydraulic diameter
F		Correction factor
htc	W/m2K	Average Heat Transfer Coefficient
U	W/m2K	Overall Heat Transfer Coefficient
N	[-]	Number of Fibres
R	[-]	Capacity ratio
Q	[kW]	Heat load
LMTD	[°C]	Logarithmic mean temperature difference
NTU		Number of transfer unit
m	Kg/s	Mass flow rate
hf	kJ/kg	Latent heat
T	[K]	Temperature
L	[m]	Heat Exchanger Effective length
Nu	[-]	Nusselt number
Pr	[-]	Prandtl number
Re	[-]	Reynolds number
DT1	[°C]	inlet hot and outlet cold fluid temperature difference
DT2	[°C]	outlet hot and inlet cold fluid temperature difference

Subscripts

i	Inlet
o	Outlet
s	Shell
f	Fibre
c	Cold
h	Hot

Greek Letters

π	[-]	Mathematical constant
μ	[kg/ms]	Dynamic viscosity
ρ	[kg/m3]	Density
ln	[-]	Natural Logarithm
λ	W/mK	Thermal conductivity
$\mathcal{E}$	[-]	Heat Exchanger Effectiveness

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 shows the schematic diagram of the test rig.

Cold water supplied on the tube side of the heat exchangers (secondary) and hot water at constant temperature pumped from tank on the shell side (Primary). During the measurement 3 types of polymer materials, and 3 types of fibres arrangement were tested. Materials were polycarbonate (PC), polypropylene (PP), and polyamide (PA11)
Polycarbonate (PC) has good chemical resistance to acids but poor resistance to alkalis and solvents. It is resistant to mineral acids, organic acids, greases and oils and dissolves in nitrile, polyamide and hot melt. The Glass transition temperature (Tg) of the polymer is 149 ^{0}C. It has a service temperature range of -40 to 130 ^{0}C.
Polypropylene (PP) is rigidly constructed and is only prone to attack by oxidising agents on the tertiary hydrogen of the other hydrocarbon polymers, PP has the highest melting point at 165 ^{0}C. It is non-toxic, non-staining and exhibits excellent corrosion resistance.
Polyamides (PA) are semi-crystalline polymers.
Within this group of polymers, Polyamide 11 (PA11) is one of great importance because of its excellent properties, including resistance to chemicals, a wide range of working temperatures (from −40 to +130 °C), less water absorption or high dimensional stability. Hence, this type of polyamide is willingly used in almost each industrial sectors.

Figure 2 shown bundle off fibres and heat exchangers modules are shown in Fig. 3. Table 1. Shown the details of polymeric heat exchangers and polymer materials used to build the modules. These modules contained anywhere between 440 and 4240 fibres. They were fabricated with the effective fibre length varying between 240 to 750 mm
All Heat exchanger were tested with 6.5 bar differential pressure (pressurized by nitrogen, immersed in 80 °C hot water). Both sides (fibres/shell) were pressurized 4 hours. Pressure was gradually increased to 6.5 bar during first 30 minutes.

Performance tests were carried out installing heat exchangers in horizontal and vertical orientation. The primary and secondary piping were insulated to reduce the heat loss from the system. Heat exchanger system heated water at constant set point temperature using steam and stored in hot water (20) tank and pumped to shell side of the heat exchangers by a centrifugal pump

The inlet and the outlet temperatures (6,8,17,19), pressure (5,7,16,18) ,differential pressure and flow rates (9,15) of the shell-side and fibre-side were respectively measure by thermocouples, pressure transducers, differential pressure transducer and turbine flow meters. Accuracy of thermocouples were estimated to be ± 0.4% of the measured values, pressure and differential pressure transducer +- 0.014 bar and flow meter 0.5% of reading

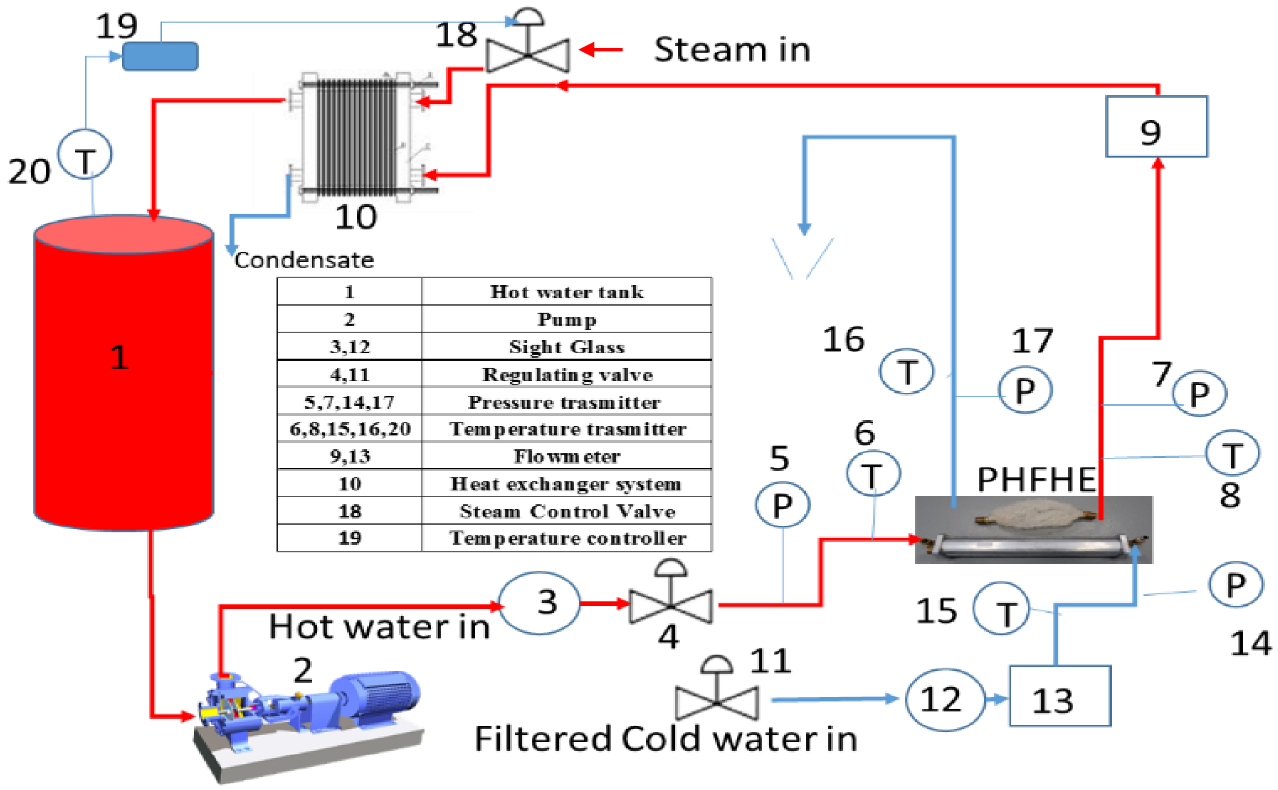

Fig.1. Schematic diagram of the test rig

HEX	No. of fibres	Fibre OD [mm]	Fibre ID [mm]	Heat transfer Area [m²]	Potting area diameter [mm]	Type	Effective length [mm]	Temp. limit [°C]	Pressure limit [bar]	Material	Fibres potting area%
AB1	740	0.8	0.65	0.52	40	twisted fibres by 45°	280	80	4	PC	30%
AB2	820	0.8	0.65	0.54	40	parallel fibres	260	80	4	PC	33%
HE2	2000	1.3	1.05	1.96	80	twisted fibres by 45°	240	80	6.5	PA11	53%
HE3	4240	0.8	0.65	2.56	80	twisted fibres by 45°	240	80	6.5	PA11	42%
H7	1800	0.8	0.65	1.54	40	twisted fibres by 45°	340	60	4	PC	72%
H8	1800	0.8	0.65	1.54	40	twisted fibres by 45°	340	60	4	PC	72%
575-576	440	0.8	0.65	0.83	20	random bundle of twisted fibres	750	80	4	PP	70%

Table .1. Properties of polymer heat exchanger, PC – polycarbonate, PA11 – polyamide, Polypropylene (PP

All measuring instruments were calibrated and calibration curves were created with which measured data were conditioned during the data-processing stages.

Test were carried out, at steady state condition where primary and secondary water flow rate, and primary hot water temperature kept nominally constant, at 80 Deg.C (6).Different hot and cold water flow rate were utilised. For each flow rate setting values of the inlet and outlet temperatures of the primary and secondary water streams, the flow rates of the inlet and exit pressure and differential pressure readings for tube side and shell side were recorded every 10 seconds using data logger for a period of 30 minutes and averaged used to carry out the performance calculations.

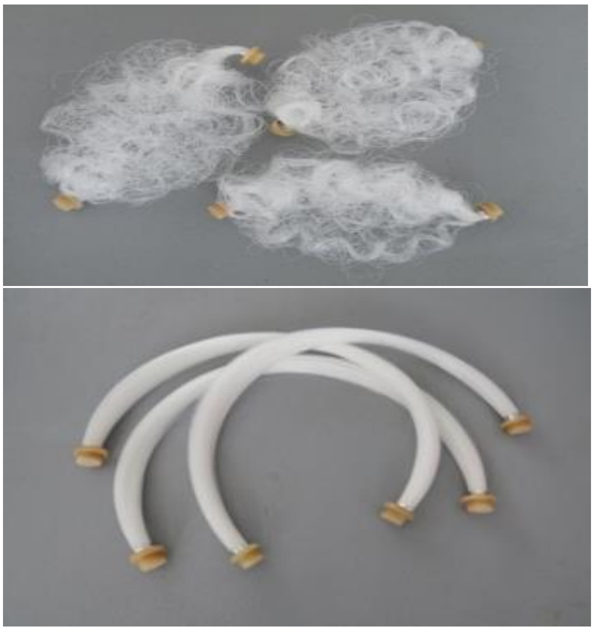

Figure.2. Bundel of fibres

Fig.3 Polymer Hollow Fibre Heat Exchangers(PHFHE)

DIMENSIONLESS PARAMETERS

The primary modes of heat transfer across the tube surfaces within the polymeric heat exchanger are forced convection and conduction through the polymer tube. In order to solve the convection coefficient for the primary hot fluid (h_h) and secondary cold fluid (h_c) an empirical relation is sought to resolve

Prandtl number; $\mathrm{Pr} = \frac{C_p \cdot \mu}{k}$ (1)

Reynolds number; $\mathrm{Re} = \frac{\rho \cdot u \cdot D_H}{\mu}$ (2)

Reynolds number is expressed in terms of mass flow rate;

$\mathrm{Re} = \frac{\dot{m} \cdot D_H}{A \cdot \mu}$ (3)

Nusselt number,

$Nu_D = fn\left(Re, Pr, \frac{Di}{L}\right)$ (4)

The following empirical relation for fully developed laminar flow in tubes at constant wall temperature used to calculate Nusselt number [8]

$$Nu = 3.66 + \frac{0.0668\left(\frac{D}{L}\right) Re\,Pr}{1 + 0.04\left[\left(\frac{D}{L}\right) Re\,Pr\right]^{2/3}} \quad (5)$$

Where $\left(\frac{D}{L}\right) Re\,Pr$ is defined as Gratez number

The heat transfer coefficient calculated from this relation is the average value over the entire length of the fibre tube.

Analogy between heat, momentum, and mass transfer is known as Colburn J-factor [11] and is a given by.

$$\mathrm{j} = \frac{Nu.\,Pr^{-1/3}}{Re}$$

PROPERTIES OF WATER

The averages measured inlet and outlet temperatures for the shell-side and fibre-side water were used to determine the density, dynamic viscosity ,thermal conductivity, specific heat constant and enthalpy of the water in each run using The IAPWS (named as IAPWS R7-97(2012) [12] between 0 °C and 100°C

HEAT EXCHANGER PERFORMANCE CALCULATIONS

Thermal performance of a heat exchanger is its capacity expressed in term of, heat flow rate, fluid flow rate, temperature, temperature difference, pressure drop, heat transfer coefficient that can determined by measurement or calculated from measure parameters.

The heat load of a heat exchanger can be derived from the equations (6) and (7).

Q= m Cp DT (6)

Q = U A LMTD (7)

LMTD (the log mean temperature difference) is defined by equation (19)

Total heat transfer area, $A = N\pi D_o L$, (9)

Shell free flow area,

$A_s = \frac{\pi}{4}(D_{is}^2\ \ ND_o^2)$ (9)

Channel area

$A_{channel} = \pi N \frac{D_i^2}{4}$ (10)

Hydraulic diameter for compact heat exchanger,

$DH_{shell} = \frac{4A_0 L}{A}$ (11)

where A is total heat transfer area, A_0 is free flow area, and L is flow length,

$DH_{shell} = \frac{(D_{is}^2\ \ ND_o^2)L}{NDoL,}$ (12)

Hydraulic diameter of the fibre

$$DH_{fiber} = D_i \quad (13)$$

$$Q(fibres) = \frac{m^{\cdot} f}{3600} c_p\left(T_{fo} - T_{fi}\right) \quad (14)$$

$$Q(shell)[kW] = \frac{m^{\cdot} s}{3600} c_p(T_{si} - T_{so}) \quad (15)$$

$$error[\%] = \frac{Q(fibres) - Q(shell)}{Q(fibres)} * 100 \quad (16)$$

$$latent\_heat\_load\_fibres = \frac{(h_{f\_fibres\_out} - h_{f\_fibres\_in}) * flow_{fibre}}{3600} \quad (17)$$

$$latent\_heat\_load\_shell = \frac{(h_{f\_shell\_in} - h_{f\_shell\_out}) * flow_{shell}}{3600} \quad (18)$$

$$DT1 = T_{si} - T_{fo};\ DT2 = T_{so} - T_{fi}$$

$$LMTD = \frac{DT1 - DT2}{\ln\left(\frac{DT1}{DT2}\right)} \quad (19)$$

$$U = \frac{Q*100}{A*LMTD} \quad (20)$$

$$R(capacity\_ratio) = \frac{T_{si} - T_{so}}{T_{fo} - T_{fi}} \quad (21)$$

The effectiveness $\mathcal{E}$ is a measure of thermal performance. It is define as a ratio of the actual heat transfer rate from the hot fluid to the cold fluid to the maximum possible heat transfer rate thermodynamically permitted.

$$\mathcal{E} = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer}$$

Correction factor *F*.

$$F = \frac{\sqrt{R+1} * \ln\left(\frac{1-\mathcal{E}R}{1-\mathcal{E}}\right)}{(1-R) * \ln\left(\frac{2-\mathcal{E}\left(R+1-\sqrt{R+1}\right)}{2-\mathcal{E}\left(R+1+\sqrt{R+1}\right)}\right)} \quad (22)$$

$$corrected\ LMTD = F * LMTD \quad (23)$$

$$NTU = \frac{T_{si} - T_{fi}}{correctedLMTD} \quad (24)$$

$$Ucorrected = \frac{Q}{A * correctedLMTD} \quad (25)$$

Fibres

$$Pr_{fibres}[-] = \frac{c_{p\_fibres} * \mu_f}{\lambda_f} \quad (26)$$

$$Re_{fibres} = \frac{\dot{m}_f * DH}{channel\_area * \mu_f} \quad (27)$$

$$Nu_{fibres} = 3.66 + \frac{0.0668\ Gz}{1 + 0.04\ Gz^{\frac{2}{3}}} \quad (28)$$

$$Gz = \frac{D_i}{L} Re\ Pr \quad (29)$$

$$htc_f = \frac{Nu_{fibres} * \lambda_f * 1000}{DH} \quad (30)$$

Shell

$$Pr_{shell} = \frac{c_{p\_shell} * \mu_s}{\lambda_s} \quad (31)$$

$$Re_{shell}[-] = \frac{\dot{m}_s * DH_{shell}}{channel\_area[* \mu_s} \quad (32)$$

$$Nu_{shell} = C * Re_{shell}^{m} * Pr_{shell}^{\frac{1}{3}} \quad (33)$$

Re_{shell}	C	m
10-100	0.9	0.4
100-1000	0.683	0.466
1000-200000	0.4	0.6

$$htc_s = \frac{Nu_{shell} * \lambda_s * 1000}{DH_{shell}} \quad (34)$$

$$U_{compact\_HE} = \frac{1}{\frac{1}{S\ htc_f} + \frac{d_0}{2\lambda_W} \ln\left(\frac{d_0}{d_i}\right) + \frac{1}{S\ htc_S}} \quad (35)$$

$$A_{required}[m^2] = \frac{Q*1000}{U_{compact\_HE} * LMTD} \quad (36)$$

$$C_s = C_h = \frac{flow_{shell} * c_p}{3600} \quad (37)$$

$$C_f = C_c = \frac{flow_{fibre} * c_p}{3600} \quad (38)$$

$$C_{min} = \min\{C_h; C_c\},\quad C_{max} = \max\{C_h; C_c\} \quad (39)$$

$$NTU = \frac{U_{compact\_HE} * A}{\min\{C_s; C_f\} * 1000} \quad (40)$$

$$Q_{max} = \min\{C_s; C_f\}\left(T_{si} - T_{fi}\right) \quad (41)$$

RESULTS AND DISCUSSION

The thermal performance of a polymeric heat exchangers with parallel, straight and twisted fibres PC – polycarbonate, PA – polyamide, Polypropylene PP using water as working fluids compared

It is assumed that the velocity profile is fully developed at the inlet, and boundary layer development in the entry region is restricted to thermal effects .Such a condition may also be assumed to be a good approximation for a uniform inlet velocity profile since Calculated Prandtl number (Pr) is >>1.

Figures 4 shows variation of differential pressure and heat exchangers effectiveness as fibre flow rate increases. The major losses associated with small diameter fibres and arrangement of the fibre within the shell. Differential pressure is an important factors to minimize pumping power required as part of system design and operating cost analysis, but in the other hand the larger allowed pressure drop the smaller heat exchanger, and higher thermal performance.

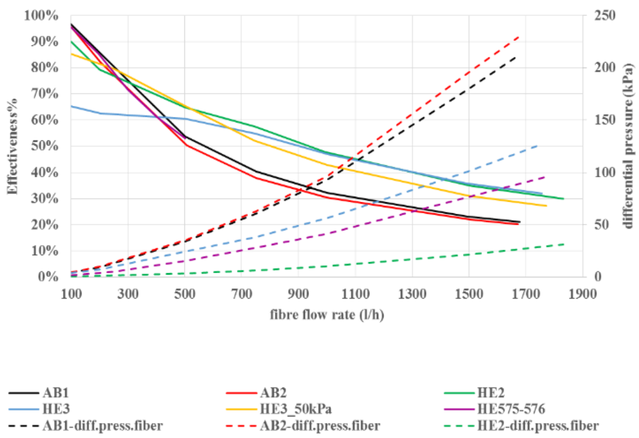

Figure.4. Energy efficiency of the heat exchangers under varying flow conditions

Figure 5 shows fibre side heat transfer characteristic of the AB1, AB2, HE2, HE3, H7 and H575-576 heat exchangers in laminar flow region. The j factor vs. Re data for fibre tube shows significant nonlinearity over the Reynolds number range $50 \leq Re \leq 12, 00$. The j factor decreases with Re number increases, the rate of decrease is higher for low Reynolds numbers and lower for high Reynolds number.

However, the ratio of increase is predominant for low Re (below 500) and it is not significant for high Re (above 1000).

This is indicating that, the value of convective heat transfer coefficient increases with respect to increase in Reynolds number. This is due to effect of increase in differential pressure, water velocity that lowers the Colburn j factor.

Calculated values of Nu number is 4 in all cases.

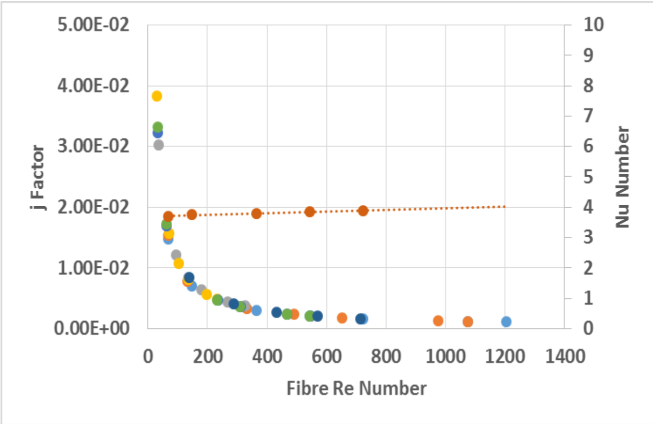

Figure.5.Fibre side Heat transfer characteristic

Thermal performance of the heat exchangers at 50 and 100 kPa pressure drop compared in figure 6 and 7

Fibre flow rate restricted due to random twisted fibre which resulted lower heat load and overall heat transfer coefficient for HE575-576

HE2 and HE3 are the same dimensions, with different numbers and type of the fibres. HE2 has 2000 fibres with outside diameter of 1.3 mm, inside diameter of 1.05 mm, 1.96 m2 of heat transfer area. HE3 has 4240 fibres with outside diameter of 0.8 mm, inside diameter of 0.65 mm, 2.56 m2 of heat transfer area. The large amount of fibres in HE3 with larger area ration resulted a large pressure drop on shell side which led to heat load reduction.

The diameter of polymeric hollow fibre has significant influence on overall heat transfer coefficients on both the inside and outside surfaces of fibres. The HE2 with maximum 36 kW of heat load and overall heat transfer coefficient of 600 w/m2k at 50kPa has better thermal performance compared with HE3, with higher heat transfer area.

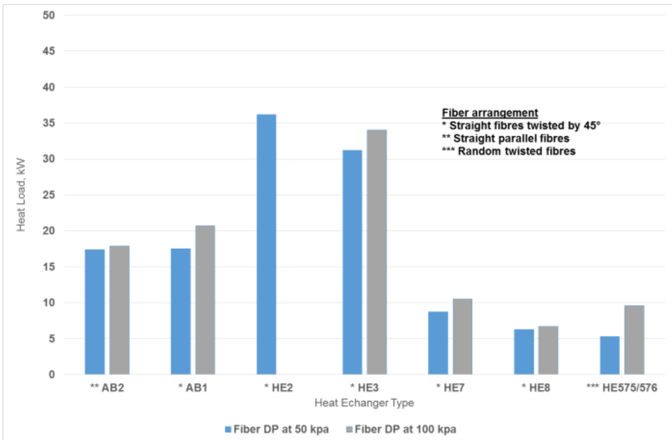

Figure.6. Heat Load, at 50 & 100 kpa, fibre differential pressure

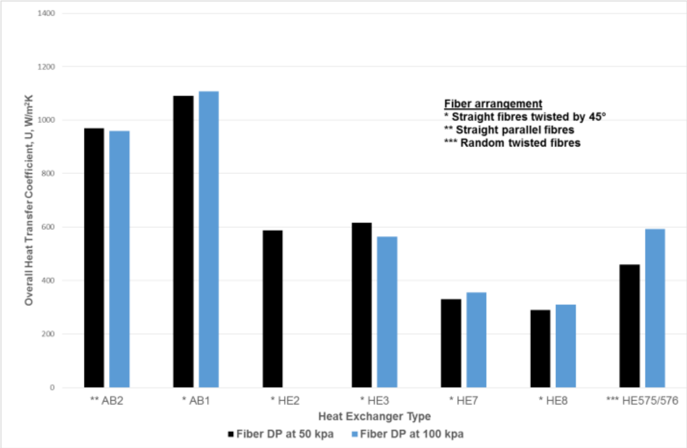

Figure.7. Overall Heat Transfer coefficient at 50 & 100 kpa differential pressure

There is no significant difference between the results for AB1 and AB2. heat exchangers. Higher heat transfer coefficients were calculated for the heat exchangers, but the maximum fibre flow rate did not exceed 600 l/h at 50 kPa and 1000 l/h at 100 kPa and heat load were limited to 18 kW and 22 kW respectively

Calculated heat load for HE2 was higher compared with AB1 and AB2, but its dimensions and weight are larger. AB1 and AB2 could be used as effective units for various applications.

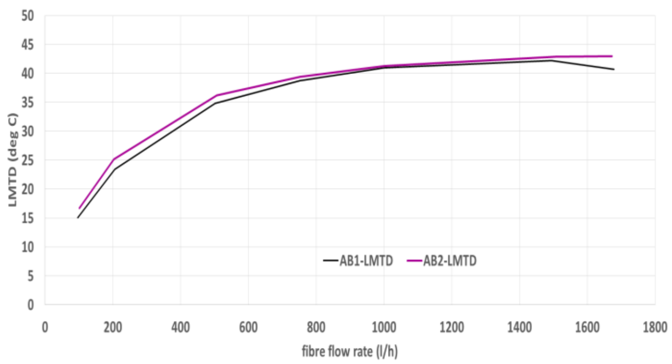

Figure.8. AB1 and AB2 heat exchangers LMTD

Figure.7. shoes variation of LMTD with fibre flow rate under present test condition for AB1 and AB2 heat exchangers. These heat exchanger achieved larger LMTD compared with other heat exchangers which indicating the heat is transferred more easily for the required heat load.

CONCLUSION

This paper presents the experimental results of polymeric hollow fibre heat exchangers for liquid to liquid application. Number of heat exchangers with differ fibre lengths, and diameter manufactured with various fibres materials and tested.

Heat exchangers with 45 degrees twisted hollow micro fibre (OD=0.8mm, ID=0.65 mm), achieved a good performance due to increase fibre's surface area. These heat exchangers are able to achieve high values of overall heat transfer coefficients and efficiencies. The low thermal conductivity of polymers are not an obstacle to manufacture the heat exchangers using polymer hollow fibres for highly corrosive industrial applications

Most importantly, the fibre materials are made from PP, PC, PA11, which are cheaper to produce, environmentally friendly, and flexible for recycling.

The highest overall heat transfer calculated under test conditions is about 1100 W/m2K which compared with typical metallic shell and tube heat exchanger

The experimental results demonstrated that such proposed polymer hollow fibre shell and tube heat exchanger have similar performance to metallic shell and tube heat exchanger. The advantages of cheaper material, flexibility of recycling, low weight for assembly and low carbon emission during the manufacturing process makes it very competitive in the heat exchanger world market.

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