Performance of Ionic Liquid-Water Mixtures in an Acetone Cooling Application

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Abstract: In this work, the performance of selected ionic liquid-water mixtures was investigated for an acetone cooling application in a process plant using a double pipe heat exchanger. Cooling media such as river water and ionic liquid-water mixtures were compared, specifically water-saturated quaternary phosphonium-based carboxylate ionic liquids were considered in this work. Ionic liquids generally have high thermal stability, resist degradation, and provide higher temperatures at low vapor pressures and for these reasons, ionic liquids can be a good substitute for conventional heat transfer fluids. At each condition, the performance of the ionic liquid mixture was compared with that of water. For the designated cooling application, the performance of water was much better than the ionic liquid mixtures.

Keywords: performance of ionic liquid mixtures; cooling application; heat transfer fluids; double pipe heat exchanger

1. Introduction

Conventional heat transfer fluids used in cooling applications are water, ethylene glycol and propylene glycol. It must be noted that water, in most cases, is readily available, has a high specific heat, and hence is readily usable as a coolant. However, water can only be used as a coolant over a narrow range of temperatures, and has the potential to freeze at around 0 °C, under ambient pressure. On the other hand, ethylene glycol and propylene glycol when mixed with water in suitable proportions acts as an antifreeze and can be used as a suitable heat transfer fluid. However, both ethylene and propylene glycol are toxic, corrosive and have the potential to degrade after prolonged usage [1].

Likewise, the most popular heat transfer fluid used in heating applications is steam. Steam can be readily produced in boilers and due to its high latent heat, is quite suitable for heating applications. However, the production of steam at high temperatures requires extremely high pressures. This makes the piping system and its associated fittings more complex. Hence, heat transfer fluids such as Dowtherm A, Duratherm, etc. are used in process plants in lieu of steam. Such heat transfer fluids have the capability to produce very high temperatures at much lower pressures.

Ionic liquids have gained interest due to their thermal conductivity, non-flammability, energy storage capacity, near zero volatility etc. In this work, the performance of water-saturated quaternary phosphonium-based carboxylate ionic liquids is compared with water for a specific acetone cooling application.

In this paper, a concentric tube heat exchanger (HX) is considered for cooling acetone from 45 °C to 30 °C. Acetone is commonly used as a solvent for greases and petroleum products. Herein, acetone is treated as a tube side fluid (inner fluid) and coolant is treated as annular (outer fluid). Water and various ionic liquid-water mixtures were considered as coolants. For each coolant, the required HX surface area is computed along with the
required length. The coolant that requires the least HX surface area and length is presumed to provide the best performance. Ionic liquid mixtures with H$_2$O composed of the tetradecyl trihexyl phosphonium ([P$_{14,6,6,6}$][+] cation coupled with butanoate ([ButO]$^-$: C$_4$H$_8$COO$^-$), hexanoate ([HexO]$^-$: n-C$_6$H$_{11}$COO$^-$), octanoate ([OctO]$^-$: n-C$_7$H$_{15}$COO$^-$), and decanoate ([DecO]$^-$: n-C$_9$H$_{19}$COO$^-$) ions were considered in this work.

The synthesis of trihexyl (tetradecyl) phosphonium butanoate, [P$_{14,6,6,6}$][ButO] (CAS: 1393375-56-8), trihexyl (tetradecyl) phosphonium hexanoate, [P$_{14,6,6,6}$][HexO] (CAS: 1393375-57-9), trihexyl (tetradecyl) phosphonium octanoate, [P$_{14,6,6,6}$][OctO] (CAS: 1393375-58-0), and trihexyl (tetradecyl) phosphonium decanoate, [P$_{14,6,6,6}$][DecO] (CAS: 465527-65-5), equipment used, properties of ionic liquid-water mixtures and comparisons with previous work are all described in detail by Oster et al. [1].

A conventional analytical method employing an effectiveness approach, i.e., number of transfer units (ε-NTU), has been used in the heat exchanger calculations. A parallel flow arrangement was considered in the analysis. In addition, the variation of heat exchanger effectiveness with respect to capacity ratio and number of transfer units (NTU) was determined. Since all the parameters in this analysis are expressed in dimensionless basis, these are applicable for any system of units, inlet fluid temperatures, fluid flow rates, materials of construction, HX size, etc.

There are numerous references available in the literature and only the most pertinent are discussed. Oster et al. [1] described the synthesis of ionic liquid-water mixtures, investigated properties such as density, specific heat, thermal conductivity, ionic conductivity, viscosity, etc. of pure [P$_{14,6,6,6}$][+] cation together with [ButO]$^-$, [HexO]$^-$, [OctO]$^-$ and [DecO]$^-$ anions. Likewise, properties of [P$_{14,6,6,6}$][ButO] + 16.680% water (w/w), [P$_{14,6,6,6}$][HexO] + 15.825% water (w/w), [P$_{14,6,6,6}$][OctO] + 14.783% water (w/w) and [P$_{14,6,6,6}$][DecO] + 14.150% water (w/w) were also considered in [1]. The apparatus used for experiments and data analysis for the estimation of properties of ionic liquid-water mixtures are described in [1]. The properties of ionic liquid mixtures with water that are considered in this work are presented in Table 1.

### Table 1. Fluid Properties.

| Inner Fluid | Outer (Annular) Fluid—Coolant |
|-------------|--------------------------------|
| Acetone     | [P$_{14,6,6,6}$] [ButO] + 16.680% Water (w/w) |
| Water       | [P$_{14,6,6,6}$] [HexO] + 15.825% Water (w/w) |
| Specific heat $c_p$ (J/kg.K) | 2150.0 | 4190.0 | 3400.0 | 3300.0 | 3100.0 | 2900.0 |
| Density $\rho$ (kg/m$^3$) | 765.4 | 999.1 | 910.0 | 908.0 | 902.0 | 900.0 |
| Dynamic viscosity $\mu$ (Pa-s) | $2.7 \times 10^{-4}$ | $1.2 \times 10^{-3}$ | $6.0 \times 10^{-2}$ | $8.0 \times 10^{-2}$ | $8.2 \times 10^{-2}$ | $9.2 \times 10^{-2}$ |
| Prandtl number Pr | 3.2 | 7.0 | 1073.7 | 1411.8 | 1359.4 | 1434.4 |
| Kinematic viscosity $\nu$ (m$^2$/s) | $3.5 \times 10^{-7}$ | $1.2 \times 10^{-6}$ | $6.6 \times 10^{-5}$ | $8.8 \times 10^{-5}$ | $9.1 \times 10^{-5}$ | $1.0 \times 10^{-4}$ |
| Thermal conductivity $K$ (W/m.K) | 0.180 | 0.575 | 0.190 | 0.187 | 0.187 | 0.186 |

Huminic and Huminic [2] studied two nanomaterials distributed in 1-hexyl-3-methylimidazolium tetrafluoroborate and the pumping power for these ionanofluids were studied. It was seen that these ionanofluids exhibit the best thermal performance under laminar flow conditions. Minea [3] reported the properties of ionic liquids and noted that experimental studies are critical to establish a complete understanding of the behavior of ionic liquids. Zhang et al. [4] considered ionic liquids as a potential absorbent in absorption refrigeration system. Thermal stability and thermal properties were all investigated at
varying mass fractions. Chereches and Minea [5] experimentally evaluated the thermal conductivity of ionanofluids. Bai et al. [6] considered the performance enhancement on the combustion characteristics of coal by the addition of ionic liquids. Meikandan et al. [7] considered ionic liquids for the cooling of solar collectors. Computational fluid dynamics was employed to study the performance of ionic liquids in solar collectors. Jozwiak and Boncel [8] reviewed the previous research work on ionic liquids. Minea and Murshed [9] reviewed the development of ionic liquids and their potential heat transfer applications. The authors describe that the ionic liquids have a great potential for heating applications and conclude that much effort is required to understand the real potential of ionic liquids. Chernikova et al. [10] discussed the various scenarios wherein ionic liquids could be used as heat transfer fluids. The properties between conventional heat transfer fluids and ionic liquids were compared. Chereches et al. [11] employed ANSYS Fluent to study the behavior of ionanofluids. Both laminar and turbulent flow were considered in the study. They concluded that the ionanofluids presents enhanced thermal properties and have a good potential for use in heat transfer applications. Bergman et al. [12] presented the concepts for modelling heat exchangers. The effectiveness—NTU approach used in this work is detailed in [12] as well as the properties of acetone and water.

This work aims to make a comparative study on the performance of ionic liquid-water mixtures in a particular cooling application.

2. Governing Equations

This project considers the cooling acetone under a set of desired conditions. Acetone has the ability to dissolve gases and petroleum products and in the process under study, it needs to be pumped to a machine for bottling that can fill 9000 sixteen-ounce bottles per hour. Acetone is stored in tanks and is maintained at 45 °C. For better operating conditions, it is desired to maintain acetone at 30 °C.

For cooling, abundant city water (from a nearby river), and various ionic liquid-water mixtures are assumed to be available at 15 °C. Likewise, the temperature rise of coolant in the heat exchanger is limited to 1 °C.

The dimensions of the double pipe heat exchanger to cool acetone using water and ionic liquid mixtures are determined. Acetone will flow through the internal pipe and the coolant will flow in the annular region, between the internal and outer pipe. The properties of acetone and various coolants are listed in the table below. The properties of ionic liquid-water mixtures were obtained from [1].

The assumptions employed in this study are as follows:
- No heat transfer between heat exchanger and the surroundings. The heat exchanger is assumed to be well insulated.
- For simplicity, conduction resistance of the inner pipe wall is assumed negligible.
- Fouling effects are not considered in this study.

The rate of heat transfer in the concentric tube heat exchanger is calculated as follows [12]:

\[ Q_{acetone} = \dot{m}_{acetone} \times C_{p \, acetone} \times \Delta T_{acetone} \]  

(1)

As there is no heat transfer between the heat exchanger and the surroundings, it can be assumed that [12]:

\[ Q_{acetone} = \dot{Q}_{coolant} \]  

(2)

The required mass flow rate of coolant can be determined from the above relationship (1).

The next step in the design process is to determine optimum diameter of inner (acetone) and outer (coolant) pipes. Choosing an appropriate diameter is critical in fluid mechanics applications. An undersized pipe will certainly undergo excessive frictional pressure drop that would add to the operational cost of the heat exchanger. Likewise, an oversized pipe will add to the capital cost of the heat exchanger. In common with industry standards, the fluid velocity is limited to 6 m/s. Since the mass flow rate of acetone and coolant are known, the diameter of inner and outer pipe can be assessed. A 1/2” schedule (SCH)
40 carbon steel (CS) pipe is chosen as the inner pipe and 2” SCH 40 CS pipe is chosen as the outer pipe. Coolant flows through the annular region and the concept of hydraulic diameter must be used for understanding flow in the annular region. The hydraulic diameter may be expressed as:

\[ D_H = D_3 - D_2 \]  

(3)

where \( D_3 \) refers to the inner diameter of the outer pipe and \( D_2 \) refers to the outer diameter of the inner pipe. The internal fluid in this analysis is acetone. The Reynolds number of acetone can be determined as follows:

\[ Re = \frac{V \cdot D_1}{\nu} \]  

(4)

where \( D_1 \) refers to the inner diameter of the inner pipe. The Nusselt number may be determined using Dittus-Boelter equation [12]:

\[ Nu_D = 0.023Re^{0.8}Pr^n \]  

(5)

where \( n = 0.3 \) for fluid cooling and \( n = 0.4 \) for fluid heating.

The Nusselt number is related to heat transfer coefficient through the following relationship [12]:

\[ Nu_D = \frac{h_i \cdot D_1}{\kappa_{acetone}} \]  

(6)

Therefore, the internal heat transfer coefficient, \( h_i \), can be determined from the above equation. The annular fluid is the coolant. The Reynolds number of coolant flow can be determined from Equation (4) by employing hydraulic diameter \( D_H \) instead of \( D_1 \). The Nusselt number and external heat transfer coefficient may be determined using Dittus-Boelter equation [12] as described above. Neglecting fouling resistances, and conduction resistance through the wall thickness, the overall heat transfer coefficient may be expressed as [12]:

\[ \frac{1}{U_o} = \frac{1}{h_i} + \frac{1}{h_o} \]  

(7)

The required temperatures and mass flow rates for acetone and coolant are known. This enables the HX effectiveness, \( \varepsilon \), and capacity rate ratio, \( Cr \) to be calculated. The capacity rate ratio may be given as:

\[ Cr = \frac{C_{\text{min}}}{C_{\text{max}}} \]  

(8)

The capacity rate \( C \) in Equation (8) is given as:

\[ C = m_c p \]  

(9)

Since there are two fluids in the HX, the fluid that has the higher value of \( C \) is \( C_{\text{max}} \) and the other fluid is \( C_{\text{min}} \). The Heat Exchanger effectiveness can be determined through the following relationship

\[ \varepsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{\dot{Q}_{\text{acetone (or)}} - \dot{Q}_{\text{coolant}}}{C_{\text{min}}(T_{\text{acetone,inlet}} - T_{\text{coolant,inlet}})} \]  

(10)

For a concentric tube heat exchanger subjected to parallel flow, the heat exchanger effectiveness may also be given as [12]:

\[ \varepsilon = \frac{1 - \exp[-NTU(1 + C_r)]}{1 + C_r} \]  

(11)

The number of transfer units (NTU) is a dimensionless parameter that is widely used by process engineers and heat exchanger designers. NTU is a substantial dimensionless parameter as it includes material characteristics, fluid characteristics, flow characteristics,
heat exchanger size, fouling, etc. The effectiveness of heat exchanger, $\epsilon$, in Equation (11) is plotted by varying NTU between 0.1 and 10, and capacity rate ratio, $Cr$, between 0 and 1. NTU = 0, is meaningless, as it refers to a HX having zero surface area.

Using Equations (8) and (10) and Figure 1/Table 2, the required NTU for HX can be determined. The HX dimensions can then be determined from NTU. The NTU can be defined as [12]:

$$NTU = \frac{U_0 A}{C_{\min}}$$

(12)

Figure 1. Heat exchanger effectiveness vs. number of transfer units (NTU) for a double pipe heat exchanger (HX) operating in parallel flow configuration.

Table 2. HX effectiveness as a function of NTU and Cr for double pipe HX in parallel flow.

| NTU | $\epsilon$ | NTU | $\epsilon$ | NTU | $\epsilon$ | NTU | $\epsilon$ | NTU | $\epsilon$ |
|-----|------------|-----|------------|-----|------------|-----|------------|-----|------------|
| Cr = 0  | 0.0        | Cr = 0.25 | 0.0    | Cr = 0.5  | 0.0    | Cr = 0.75 | 0.0    | Cr = 1.0  | 0.0    |
| 0.5    | 0.39       | 0.5 | 0.37       | 0.5 | 0.35       | 0.5 | 0.33       | 0.5 | 0.32       |
| 1.0    | 0.63       | 1.0 | 0.57       | 1.0 | 0.52       | 1.0 | 0.47       | 1.0 | 0.43       |
| 1.5    | 0.78       | 1.5 | 0.68       | 1.5 | 0.60       | 1.5 | 0.53       | 1.5 | 0.48       |
| 2.0    | 0.86       | 2.0 | 0.73       | 2.0 | 0.63       | 2.0 | 0.55       | 2.0 | 0.49       |
| 2.5    | 0.92       | 2.5 | 0.76       | 2.5 | 0.65       | 2.5 | 0.56       | 2.5 | 0.50       |
| 3.0    | 0.95       | 3.0 | 0.78       | 3.0 | 0.66       | 3.0 | 0.57       | 3.0 | 0.50       |
| 3.5    | 0.97       | 3.5 | 0.79       | 3.5 | 0.66       | 3.5 | 0.57       | 3.5 | 0.50       |
| 4.0    | 0.98       | 4.0 | 0.79       | 4.0 | 0.67       | 4.0 | 0.57       | 4.0 | 0.50       |
| 4.5    | 0.99       | 4.5 | 0.80       | 4.5 | 0.67       | 4.5 | 0.57       | 4.5 | 0.50       |
| 5.0    | 0.99       | 5.0 | 0.80       | 5.0 | 0.67       | 5.0 | 0.57       | 5.0 | 0.50       |
Table 2. Cont.

| NTU | ε | NTU | ε | NTU | ε | NTU | ε | NTU | ε |
|-----|---|-----|---|-----|---|-----|---|-----|---|
| Cr = 0 | Cr = 0.25 | Cr = 0.5 | Cr = 0.75 | Cr = 1.0 |
| 5.5 | 1.00 | 5.5 | 0.80 | 5.5 | 0.67 | 5.5 | 0.57 | 5.5 | 0.50 |
| 6.0 | 1.00 | 6.0 | 0.80 | 6.0 | 0.67 | 6.0 | 0.57 | 6.0 | 0.50 |
| 6.5 | 1.00 | 6.5 | 0.80 | 6.5 | 0.67 | 6.5 | 0.57 | 6.5 | 0.50 |
| 7.0 | 1.00 | 7.0 | 0.80 | 7.0 | 0.67 | 7.0 | 0.57 | 7.0 | 0.50 |
| 7.5 | 1.00 | 7.5 | 0.80 | 7.5 | 0.67 | 7.5 | 0.57 | 7.5 | 0.50 |
| 8.0 | 1.00 | 8.0 | 0.80 | 8.0 | 0.67 | 8.0 | 0.57 | 8.0 | 0.50 |
| 8.5 | 1.00 | 8.5 | 0.80 | 8.5 | 0.67 | 8.5 | 0.57 | 8.5 | 0.50 |
| 9.0 | 1.00 | 9.0 | 0.80 | 9.0 | 0.67 | 9.0 | 0.57 | 9.0 | 0.50 |
| 9.5 | 1.00 | 9.5 | 0.80 | 9.5 | 0.67 | 9.5 | 0.57 | 9.5 | 0.50 |
| 10.0 | 1.00 | 10.0 | 0.80 | 10.0 | 0.67 | 10.0 | 0.57 | 10.0 | 0.50 |

From Figure 1 and Table 2 it is shown that increasing NTU over a certain limit is not advantageous, as the heat exchanger effectiveness tends to be constant beyond the threshold limit. Since NTU is directly proportional to the HX surface area, higher NTU value corresponds to larger surface area. This means that adding surface area to the HX does not necessarily enhance the heat transfer at all times. This is in contrast with the generalization that increasing area enhances heat transfer. Figure 1 and Table 2 presented herein will guide engineers to select the appropriate NTU for the heat exchanger, thereby optimizing the HX area and capital cost.

The overall heat transfer coefficient and minimum capacity rate \( C_{\text{min}} \) are known as well. Therefore, using Equation (12), surface area of HX can be determined. From surface area, \( A \), the length of the heat exchanger can be readily obtained as well. The surface area, \( A \), and HX length, \( L \), are coupled by the following relationship:

\[
A = \pi \times D^2 \times L
\]

3. Results & Discussion

Table 3 describes the cooling of acetone in a double pipe HX using different coolants. The ionic liquid mixtures considered are \([P_{14,6,6,6}] [\text{ButO}] + 16.680\% \text{ water (w/w)}\), \([P_{14,6,6,6}] [\text{HexO}] + 15.825\% \text{ water (w/w)}\), \([P_{14,6,6,6}] [\text{OctO}] + 14.783\% \text{ water (w/w)}\) and \([P_{14,6,6,6}] [\text{DecO}] + 14.150\% \text{ water}\). The properties of water and ionic liquid-water mixtures at operating conditions are described in Table 1.

Based on the equations presented, a MATLAB model was established to understand the cooling performance of acetone. The results presented are directly from the analytical HX model using \( \varepsilon \)-NTU approach. For simplicity, only parallel flow configuration was considered in this work. Similar results are also expected for counter flow configuration.

Based on the analysis, water provides the best cooling performance. The results from the MATLAB model are described in Tables 3 and 4, and Figure 2. Water, under ambient conditions, possesses a high specific heat, lower viscosity and high thermal conductivity. These are the main reasons for which water is popular as a heat transfer fluid and used as a secondary refrigerant in commercial air conditioning applications. In the current analysis, from Table 4 and Figure 2, it can be seen that the HX area is significantly reduced while using water as the cooling medium. This is because water possesses higher specific heat and lower viscosity as compared with the ionic liquid-water mixtures. It must be noted that the usage of ionic liquid-water mixtures as heat transfer fluid is still under examination due to the cost of ionic liquids. Pure ionic liquids have higher dynamic viscosities and lower densities, and these make the flow in the heat exchanger in laminar regime. The properties of pure ionic liquids can certainly be enhanced by dilution with water. Water has much
lower viscosity, higher specific heat and hence is the best choice. The aspect of diluting pure ionic liquid also lowers the cost of ionic liquid mixtures.

Table 3. Double pipe HX model for cooling acetone using water and ionic liquid mixtures.

| Item                              | Inner Fluid Hot Fluid | Outer (Annular) Fluid—Coolant |
|-----------------------------------|-----------------------|------------------------------|
|                                   | Acetone               | [P<sub>14,6,6,6</sub>] [ButO] + 16.680% Water (w/w) | [P<sub>14,6,6,6</sub>] [HexO] + 15.825% Water (w/w) | [P<sub>14,6,6,6</sub>] [OctO] + 14.783% Water (w/w) | [P<sub>14,6,6,6</sub>] [DecO] + 14.150% Water (w/w) |
| Specific heat c<sub>p</sub> (J/kg·K) | 2150.0                | 3400.0                       | 3300.0                      | 3100.0                        | 2900.0                      |
| Density ρ (kg/m³)                 | 765.4                 | 910.0                        | 908.0                       | 902.0                         | 900.0                       |
| Dynamic viscosity µ (Pa·s)        | 2.7 × 10⁻⁴            | 1.2 × 10⁻³                   | 6.0 × 10⁻²                  | 8.2 × 10⁻²                    | 9.2 × 10⁻²                  |
| Prandtl number Pr                 | 3.2                   | 7.0                          | 1073.7                      | 1411.8                        | 1359.4                      | 1434.4                      |
| Kinematic viscosity ν (m²/s)      | 3.5 × 10⁻⁷            | 1.2 × 10⁻⁶                   | 6.6 × 10⁻⁵                  | 8.8 × 10⁻⁵                    | 9.1 × 10⁻⁵                  | 1.0 × 10⁻⁴                  |
| Thermal conductivity K (W/m·K)    | 0.180                 | 0.575                        | 0.190                       | 0.187                         | 0.187                       | 0.186                       |
| mass flow rate (kg/s)             | 0.9                   | 7.0                          | 8.6                         | 8.9                           | 9.5                         | 10.1                        |
| Inlet temp T<sub>in</sub> (°C)    | 45.0                  | 15.0                         | 15.0                        | 15.0                          | 15.0                        | 15.0                        |
| Outlet temp T<sub>out</sub> (°C)  | 30.0                  | 16.0                         | 16.0                        | 16.0                          | 16.0                        | 16.0                        |
| Rate of heat transfer (W)         | 29,399.1              | 29,399.1                     | 29,399.1                    | 29,399.1                      | 29,399.1                    | 29,399.1                    |
| Flow rate (m³/s)                  | 1.2 × 10⁻³            | 7.0 × 10⁻³                   | 9.5 × 10⁻³                  | 9.8 × 10⁻³                    | 1.1 × 10⁻²                  | 1.1 × 10⁻²                  |
| Diameter d₁ (inner diameter of inner pipe) m | 1.58 × 10⁻² | 2.13 × 10⁻² | 2.13 × 10⁻² | 2.13 × 10⁻² | 2.13 × 10⁻² | 2.13 × 10⁻² |
| Diameter d₂ (outer diameter of inner pipe) m | 5.25 × 10⁻² | 5.25 × 10⁻² | 5.25 × 10⁻² | 5.25 × 10⁻² | 5.25 × 10⁻² | 5.25 × 10⁻² |
| Diameter d₃ (inner diameter of outer pipe) m | 6.03 × 10⁻² | 6.03 × 10⁻² | 6.03 × 10⁻² | 6.03 × 10⁻² | 6.03 × 10⁻² | 6.03 × 10⁻² |
| Diameter d₄ (outer diameter of outer pipe) m | 3.12 × 10⁻² | 3.12 × 10⁻² | 3.12 × 10⁻² | 3.12 × 10⁻² | 3.12 × 10⁻² | 3.12 × 10⁻² |
| Hydraulic diameter Dₕ m           | 6.1                   | 3.9                          | 5.3                         | 5.4                           | 5.8                         | 6.2                         |
| Velocity (m/s)                    | 275,757               | 104,848                      | 2485                        | 1920                          | 1994                        | 1900                        |
| Reynolds number                    | 0.3                   | 0.4                          | 0.4                         | 0.4                           | 0.4                         | 0.4                         |
| Nusselt number                    | 732.6                 | 520.8                        | 195.1                       | 177.1                         | 179.8                       | 176.8                       |
| heat transfer coefficient (W/m²K)  | 8346.9                | 9609.5                       | 1189.7                      | 1062.9                        | 1079.1                      | 1055.0                      |
| C (W/K)                            | 1959.9                | 29,399.1                     | 29,399.1                    | 29,399.1                      | 29,399.1                    | 29,399.1                    |
| C<sub>min</sub>/C<sub>max</sub>    | 0.1                   | 0.1                          | 0.1                         | 0.1                           | 0.1                         | 0.1                         |
| Effectiveness                     | 0.5                   | 0.5                          | 0.5                         | 0.5                           | 0.5                         | 0.5                         |
| Required NTU<sub>parallel flow</sub> | 0.8                  | 0.8                          | 0.8                         | 0.8                           | 0.8                         | 0.8                         |
| Universal heat transfer coefficient (W/m²K) | 4466.9               | 1041.3                       | 942.8                       | 955.6                         | 936.6                       |
| Required HX area A₀ (m²)          | 0.3                   | 1.4                          | 1.6                         | 1.5                           | 1.5                         | 1.6                         |
| Required HX length L (m)          | 4.9                   | 21.1                         | 23.3                        | 23.0                          | 23.4                        | 23.4                        |
Figure 2. Required double pipe HX area and length—Cooling acetone using water and ionic liquid mixtures.

| Item          | H₂O | [P₁₄,₆,₆,₆] [ButO] | [P₁₄,₆,₆,₆] [HexO] | [P₁₄,₆,₆,₆] [OctO] | [P₁₄,₆,₆,₆] [DecO] |
|---------------|-----|---------------------|---------------------|---------------------|---------------------|
| Required HX area A₀ (m²) | 0.3 | 1.4                 | 1.6                 | 1.5                 | 1.6                 |
| Required HX length L (m)    | 4.9 | 21.1                | 23.3                | 23.0                | 23.4                |

From Table 3, it can be clearly seen that the flow of ionic liquid-water mixtures are in the transition region. Generally, for heat transfer applications, it is required to have flows in the turbulent regime for better mixing and heat transfer characteristics. Inverters and mixers installed in pipes can artificially create turbulent flows, and these might be necessary while dealing with ionic liquids. Nevertheless, from [1] it can be inferred that ionic liquid-water mixtures have other characteristics such as lower vapor pressures, lower freezing point, potential for energy storage, etc. The ability of ionic liquid-water mixtures to lead to higher temperatures at lower vapor pressures is quite advantageous and can serve as a potential alternative for conventional heat transfer fluids. Likewise, the capability of ionic liquid-water mixtures to produce a lower freezing point adds immense value to industrial refrigeration systems. However, further studies are required to assess the possibility of using ionic liquid-water mixtures in other heat transfer applications.

4. Conclusions

It must be noted that for the chosen sensible cooling application, water is the best choice for coolant selection. The thermo-physical properties of water under ambient conditions are highly suitable for cooling applications. However, the results from this study cannot be generalized as the work considered herein is for a specific cooling application. Characteristics such as corrosion, thermal stability, degradation, fouling, etc. were not
considered in this study. A detailed study on these characteristics must be explored with the help of experimentation. Likewise, as an extension of this work, the suitability of ionic liquid-water mixtures for higher temperature applications must be analyzed. The benefits of their integral characteristics such as thermal stability, energy storage, capability to produce higher temperatures at lower vapor pressures, etc. may then become more evident.

With regard to HX, to be competitive in the market, it is imperative to design HX equipment such that it is small, less heavy and low cost and yet delivers the required heat transfer. The parameters described in Figure 1/Table 2 are in dimensionless basis and hence are appropriate for either the metric or British system of units. Figure 1/Table 2 will help the engineers in selecting the precise NTU for a double pipe HX operating in parallel flow configuration. Since NTU is a function of HX area, increasing NTU increases HX area. Moreover, additional HX area implies more material, extra weight, and more capital cost. It can be clearly seen from Figure 1 and Table 2 that increasing the NTU (HX area) beyond 3.0 serves no purpose. This is contrary to the general belief that increasing the surface area increases heat transfer. Therefore, the established information in this work will benefit engineers in erecting a cost effective double pipe heat exchanger for any given thermo-fluid application.

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