

HEAT TRANSFER TO SUB-COOLED BINARY SOLUTION IN HORIZONTAL TUBE

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ABSTRACT

Predicted heat transfer coefficients (HTC) are widely used for calculations of various heat transfer processes. Various empirical and semi-empirical models of non-dimensional groups are used for calculating the heat transfer coefficients. The common accuracy of the predicted heat transfer coefficient is usually about $\pm 25\%$; however, the accuracy is failing by the inaccurate predictions of the fluid properties.

In the present study the heat transfer coefficient of sub-cooled organic mixture chlorodifluoromethane (R22) - dimethylacetamide (DMAC) were predicted. While the solution's pressure-temperature-concentration, density-temperature-concentration and viscosity-temperature-concentration relations were previously measured in our laboratory and the heat capacity were calculated from the enthalpy of the solution, the only unknown property that was required for predicting the HTC was the mixture thermal conductivity. Various correlations and mixing rules for the mixture thermal conductivity were adopted for predicting the heat transfer coefficient. In order to validate the predicted heat transfer coefficients, an experimental system was designed, built and successfully operated. The heat transfer coefficient was measured experimentally and compared with the predicted one. Based on this comparison the best fitted thermal conductivity for the working fluid (R22-DMAC) is recommended. As a result, the deviations of the predicted heat transfer coefficients, obtained by the well-known Dittus-Boelter equation, from the experimental values were less than 15%.

INTRODUCTION

Utilization of available heat sources for cooling and refrigeration can be implemented by various types of absorption heat pumps, both single and multi-stage. However, the utilization of low potential heat sources for cooling and refrigeration ($< 0^\circ\text{C}$) is limited by the properties of the working fluids and the cycle configuration of the heat pump. For utilization of low potential heat sources ($80\text{-}120^\circ\text{C}$) in cooling and refrigeration to $< 0^\circ\text{C}$, a single-stage absorption heat pump based on organic working fluids is preferable, since conventional working fluids such as ammonia-water or water-lithium bromide are limited to the above described operation conditions. The temperature of the heat source and the cooling

or refrigeration demands are usually the factors that determine the type of working fluid to be used and the type of the absorption heat pump system required-single or multistage.

The commonly used working fluids are ammonia-water or water-lithium bromide. The ammonia-water combination requires a heat source temperature above 120°C for cooling and refrigeration to $< 0^\circ\text{C}$. Such a system is a high-pressure system that requires a rectification column [1]. Ammonia has acceptable thermophysical properties, but it is a flammable and toxic fluid, strongly irritant and is corrosive to copper. The water-lithium bromide solution can be used with a heat source temperature above 70°C for air-conditioning but not for cooling and refrigeration because of the limitation of the evaporator temperature ($> 0^\circ\text{C}$). This system operates under vacuum and does not require a rectification column. The water-lithium bromide solution is highly corrosive and extremely viscous and viscosity-reducing agents are frequently required. The limitations of using common working fluids [2] for utilizing low potential heat sources ($80\text{-}120^\circ\text{C}$) for cooling and refrigeration ($< 0^\circ\text{C}$) are therefore self-evident.

To overcome these limitations, we have chosen working fluids based on hydrofluorocarbon (HFC) and hydrochlorofluorocarbon (HCFC) refrigerants and organic

NOMENCLATURE

| | | |
|-----------|------------------------|---------------------------------|
| A | [m ²] | Area |
| D | [m] | Diameter |
| f | [-] | Friction Factor |
| h | [W/(m ² K)] | Heat transfer coefficient (HTC) |
| H | [kJ/kg] | Enthalpy |
| k | [W/mK] | Thermal conductivity |
| L | [m] | Length |
| \dot{m} | [lit/min] | Mass flow rate |
| Nu | [-] | Nusselt Number |
| Pr | [-] | Prandtl number |
| q | [kW] | Heat Power |
| Re | [-] | Reynolds number |
| T | [K] | Temperature |
| x | [-] | Mole fraction |

Subscripts

| | |
|-----|--------|
| W | Wall |
| L | Liquid |

absorbents [3-6]. The performance of these working fluids in a conventional and triple pressure level (TPL) single-stage absorption cycles were investigated numerically and showed their abilities to overcome the above mentioned limitations.

To evaluate the figures of merit and performance indicators of these cycles one needs to use data related to the heat transfer coefficients of the working fluids. Predicted heat transfer coefficients are widely used for calculations of various heat and mass transfer processes. Various empirical and semi-empirical models of non-dimensional groups are used for calculating the heat transfer coefficients. These groups depend on the fluid and flow properties. The common accuracy of the predicted heat transfer coefficient is usually about $\pm 25\%$ [7]. Therefore many works, which can be found in the literature, aimed to achieve better accuracy; however, the accuracy is failing by the inaccurate predictions of the fluid properties. When the fluid is a mixture of miscible fluids the predictions of the fluid properties are very rough and therefore the prediction of the heat transfer coefficient is more complicated.

The Dittus-Boelter correlation [8] is widely used for calculating the HTC for fully developed turbulent flow in a pipe. Additional correlations for the Nusselt number, pretended to be more accurate than the widely used Dittus-Boelter equation, were suggested by Gnielinski [9] for heating in smooth and non-smooth pipes. In order to calculate the heat transfer to a test tube, one should know the flow conditions at the inlet and the outlet of the test section and the solution's density, viscosity, heat capacity and thermal conductivity. Since the solution's pressure-temperature-concentration, density-temperature-concentration and viscosity-temperature-concentration relations were previously measured in our laboratory [10], the heat capacity were calculated from the enthalpy of the solution and the fluid volumetric flow rate was measured in this study, the only unknown property that was required for calculating the HTC was the thermal conductivity of the mixture.

Thermal conductivity of solutions is one of the properties that are difficult to predict. Most of the existing techniques are empirical. The most popular measuring technique for the thermal conductivity is based on transient measurement of hot wire [11]. Although this technique is widely used for pure fluids, it is yet unreliable for mixtures, since the properties of the mixtures are not well defined; the local concentration of the volatile phase may change together with the fluid local temperature and the various time scales of the heat and mass transfer.

Several correlations for prediction of thermal conductivity of pure liquids were developed [12-24]. In order to estimate the thermal conductivity of liquid mixtures, various mixing rules for the solution can be used [25-29].

PRESENT STUDY

In the present study, 65 combinations of different correlations of thermal conductivity for the two pure component of the mixture (R22-DMAC) and different mixing rules were used to assess the heat transfer coefficient (HTC).

The pure component correlations that were tested were the Missenard correlation [12-15], the boiling point method correlation [16], the Latini method correlation [17-23] and the Sastri Method correlation [24]. The mixing rules that were tested were the Li Method [25], the Jemeison Method [26], the Phillipov Method [27, 28], the Power Low Method [29-32] and the Latini and Baroncini method [19-22], which is a combination of correlation of pure components and mixing rules.

The experimental value of the HTC under the following experimental conditions: nominal volumetric flow rate of 3 lit/min, operating pressure 3 bars and heat input of 600 Watt, is presented in Fig. 1 as a horizontal dashed line. The predicted HTC values at the same conditions are also presented in Fig. 1 as a function of the thermal conductivity. The predicted HTC were obtained by using the Dittus-Boelter correlation with the 65 combinations for estimating the thermal conductivity of the mixture. The predicted HTC demonstrates a very large scattering.

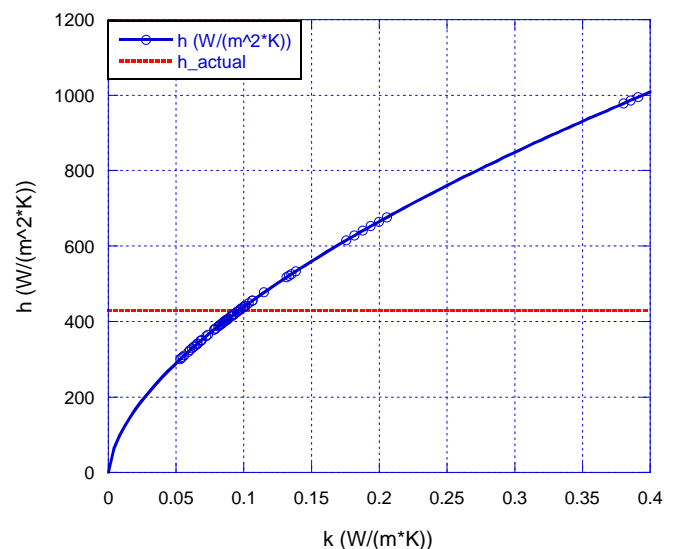


Figure 1 - HTC vs. thermal conductivity for all 65 options

Since the only unknown property that was used for calculating the HTC was the thermal conductivity of the mixture, it was decided to develop an experimental system for determine the HTC directly and compare it with the predicted values. The question was how to choose the best correlations for the thermal conductivity and what kind of mixing rule should be used: Should it be the correlation with the lowest deviation from the experimental data, or the correlations that fits the pure components with the most appropriate mixing rule?

During this study, the best thermal conductivity correlations and mixing rules were defined as such that revealed the best assessment of the heat transfer coefficient (HTC). The best assessment was obtained by comparing the experimental values of the HTC with a known HTC correlation. First the best correlation for each pure component was chosen, and than the most appropriate mixing rule was chosen.

EXPERIMENTAL SET UP

In order to investigate the HTC and the thermal conductivity of the solution, an experimental system for analyzing heat transfer of binary solutions was designed, built and successfully operated. Schematic illustration of the experimental setup is shown in Fig. 2. The experimental system includes test section, separation chamber, heat exchangers, solution pump and absorber. In addition there are two peripheral systems: a cooling water cycle, which enables cooling the absorber and the solution before the entrance to the solution pump, and a heating thermal oil cycle, which enables heating the solution before the entrance to the test section (3/4" copper tube). Two site glass chambers (3/4") fitted at the inlet and outlet of the test section for observing and photography the solution at these locations.

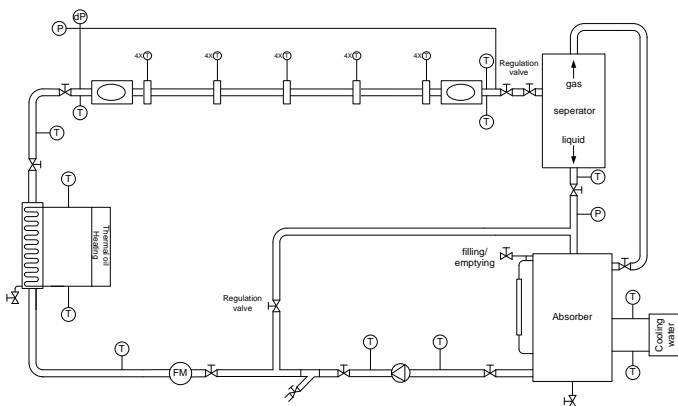


Figure 2 - The experimental set up

A rich R22-DMAC solution, at equilibrium or sub-cooled condition, pumped from the absorber through a volumetric flow meter and a thermal oil heat exchanger for pre-heating toward the test tube at elevated pressure. The purpose of the pre-heating is getting equilibrium conditions at the exit of the test section. Equilibrium condition was defined when the volatile fluid, R22 in this cases, started to be desorbed from the solution and a vapor bubble was observed at the tube outlet site glass. Generating the first bubbles at the exit of the test section ensure an equilibrium condition. By measuring both the solution temperature and pressure at the tube outlet, the solution composition at equilibrium can be found. Electric wire was wrapped around the test section and used as the heating source. This ensured a uniform heat flux through pipe walls. The solution flows through a regulation valve which controls the solution flow rate and act as a pressure reducer valve. The pressure reduction of the solution in the separation vassal causes to flashing phenomenon, the gas phase (mostly the refrigerant vapor) flows up and the liquid (poor solution) flows down. The two streams flow into the absorber, where the poor solution absorbs again the refrigerant to get a rich solution at equilibrium or at sub-cooled condition at the absorber outlet.

Thermal oil is heated and circulated by a Haake-N3-B heating bath. It was used to pre-heat the solution at the heat exchanger, and cooling water bath was used to cool the solution

at the absorber. Pressure and temperature were measured at several locations along the system (see Fig. 2). The experiments were performed at several nominal values of flow rate, pressure and heating input to the test tube. Due to the limitations of the pre-heating cycle (3KW), not all combinations of flow rate, heat input and pressure could be tested, e.g., equilibrium conditions can't be obtained for high pressure, high flow rates and low heat input. The experiments begun at equilibrium conditions, and operated until it reached a steady state steady flow conditions.

Fluid properties

For analyzing the experimental data the solution thermophysical properties have to be known. These are depending on pressure, temperature and solution composition. Experimental measurements and evaluation of equilibrium thermodynamic properties of the mixture R22-DMAC were conducted in our laboratory [33-34]. The refrigerant concentrations in the liquid and the vapor phases for each measured point as a function of the pressure and temperature, the enthalpy of the solution as a function of temperature and concentration, and experimental measurements of the viscosity and the density of the solutions at equilibrium as a function of temperature and pressure were carried out. The only unknown parameter that needs for the calculations was the thermal conductivity, which is estimated in this work.

Measurement Devices

Temperatures at various locations (as illustrated in Fig. 2) were measured by copper-constantan (type T) thermocouples with a maximum uncertainty of $\pm 0.3^{\circ}\text{C}$. All the thermocouples were connected differentially to the same reference point, thus the difference between measured temperatures would be more accurate. The high and the low pressure of the system were measured by STS pressure transducers, ATM model, measuring range of 0-20 bar with a maximum uncertainty of $\pm 0.1\%$ FS, located at the inlet to the test tube and the exit of the separation chamber, respectively. The high pressure was controlled electronically by a Grundfos CRNE2 multi-stage centrifugal pump. The volumetric flow rate of the solution at the exit of the pump was measured by 1/2" Trimec Turbopulse Turbine flow meter, measuring range of 1.83-18.33 lit/min, with a maximum uncertainty of $\pm 0.5\%$ FS. All the thermocouples and measuring devices were connected to a personal computer via an Agilent electronic data logging system with sampling rate of 10 channels per sec. Data logging application was written using National Instruments Lab VIEW.

Analysis of the Experimental Data

Each data set, at steady state steady flow conditions, includes above 100 measured points per each measurement device that were recorded for particular flow rate, pressure and heat input, was analyzed in the following way. The average value of each thermocouple, pressure and flow rate at the steady state steady flow was calculated. The liquid temperature (bulk temperature, T_L) profile assumed to be linear along the heating section and its averaged value was calculated as the

average of the inlet and outlet temperatures. The wall temperature (T_w) was measured at 5 locations along the heating section (see Fig. 2). 4 thermocouples were mounted in each location, one at the bottom, one at the top and two on the side of the pipe cross section. These were used to obtain the averaged wall temperatures along the pipe. Since the wall temperature at the two ends of the test section were influenced by the heat transfer to the non heated parts of the system, it was decided to eliminate the sides effects and to investigate the heat transfer coefficient only at the 2 middle sections of the heated pipe. Therefore to avoid the ends effects, the inlet and outlet wall temperatures were not taken into account to obtain the linear wall temperature profile. Thus the experimental HTC was calculated by dividing the wall heat flux by the averaged temperature difference between the liquid and the pipe wall.

$$h = q/A \cdot (\overline{T_w} - T_L) \quad (1)$$

The heat flux, q/A , was calculated by an energy balance of the fluid.

$$q = \dot{m}(H_{out} - H_{in})/\pi D_i L \quad (2)$$

where D_i , L , H_{in} and H_{out} are the inner tube diameter, the tube length and the enthalpies of the solution at the inlet and the outlet of the investigated test section, respectively. By knowing the solution temperature and concentration (that was constant during a specific test), the enthalpy of the solution was obtained. As mentioned above the HTC can also be predicted by adopting various empirical correlations from the literature. Thus the HTC can be calculated from Nusselt number:

$$h = Nu \cdot k/D_i \quad (3)$$

The Dittus-Boelter equation [8] for Nusselt number is widely used:

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (4)$$

This correlation is for fully develop turbulent flow, i.e., $Re \geq 2500$ [8]. The Reynolds numbers of the flows which were investigated during this study were in the range between 2200 and 8500.

Two other Nusselt correlations, pretended to be more accurate than the widely used Dittus-Boelter equation, were also examined. These correlations were suggested by Gnielinski [9]. The first correlation considered the wall friction factor, and the second correlation was developed for a smooth pipe. Gnielinski correlation [3] for heat transfer in non-smooth pipe

$$Nu = \frac{(f/8)(Re_p - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \quad (5)$$

This correlation is valid for $0.5 < Pr < 2000$ and $3000 < Re_D < 5 \cdot 10^6$.

The friction factor was calculated by [25]:

$$f = (0.790 \ln Re_p - 1.64)^{-2} \quad (6)$$

Gnielinski correlation [3] for heat transfer in smooth pipe:

$$Nu = 0.012(Re_D^{0.87} - 280)Pr^{0.4} \quad (7)$$

and is valid for $1.5 < Pr < 500$ and $3000 < Re_D < 5 \cdot 10^6$.

The Dittus-Boelter correlation fitted better for a wider range of HTC. The two Gnielinski's correlations are less accurate for low HTC values.

EXPERIMENTAL RESULTS AND DISCUSSION

Using all possible combinations of correlations and mixing rules (65 options) for the thermal conductivity of the two components, the heat transfer coefficient was predicted and compared with the experimental data. The maximum errors which were obtained for each combination are between 10% and 65%. Only 32 options are within an error less than 20%. In order to choose the best correlation and mixing rule, each combination was tested, and the most appropriate combination was obtained. The best combination that fits the experimental results is the Boiling Point correlation for the absorbent, the Latini correlation for the refrigerant and the power low mixing rule (see Fig. 3). This combination fits the experimental result the most; but it is not necessarily the right one to choose, since the thermal conductivity correlation that was used for each component is not necessarily the most appropriate correlation for predicting the pure component thermal conductivity. Therefore, it was decided to look for the best correlation for the absorbent, then for the refrigerant and finally choosing the best mixing rule. This will be presented in the following subsections.

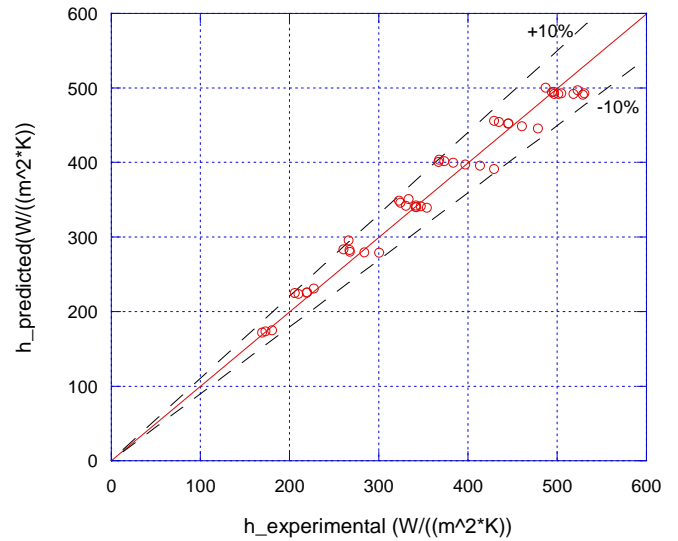


Figure 3 - Comparison between the experimental HTC vs. the predicted HTC for the best combination: Boiling Point correlation for the absorbent, the Latini correlation for the refrigerant and the power low mixing rule.

The Absorbent Correlation

The Missenard, Boiling point, Latini and Sastri correlations were used to predict the thermal conductivity of the absorbent. From the comparison between the predicted and the literature [35] values of the DMAC thermal conductivities, the

Sastri model underestimated thermal conductivity of the absorbent; the boiling point and the Latini correlations have better results, and the Missenard correlation fits the best with an error less than 10%. Therefore it was concluded that the Missenard correlation is the best option for DMAC. The four correlations were also tested for predicting the thermal conductivities of other absorbents (DMETEG and NMP). A comparison between the predicted and the literature [36] values of the absorbents DMETEG and NMP thermal conductivities was also made. Although the fitting correlations have different behavior with these two absorbents, it seems that the Missenard correlation is yet the best choice for predicting the absorbents thermal conductivities.

The Refrigerant Correlation

The Missenard, Boiling point, Latini and Sastri correlations were used to predict the thermal conductivity of the refrigerant. Comparison between the predicted and the literature [37] values of the refrigerant R-22 thermal conductivities was made. The Missenard correlation was far from describing the thermal conductivity of the refrigerant. The other three were close to each other, and the error range was about 15%. The boiling point and the Latini correlations had similar results, and were better for the higher values of thermal conductivity. The Sastri correlation was better for the lower values of thermal conductivity. In this work the values of thermal conductivity are low (at high temperatures), so the correlation of Sastri is the best option for R22. This correlation was also tested for other refrigerants: R32, R124, R125, R134a and R152a. The results were compared to literature values [37]. The Sastri correlation was not as good for the refrigerants R32 and R134a as for R22. The results for the refrigerants R125, R134a and R152a were very close with an error range of 25%. It should be noted that the Latini correlation fits much better than the Sastri correlation for all the refrigerants for low temperatures, however, since the Sastri correlation gives better predictions for R22 at higher temperatures (low thermal conductivities), as is the case in this study, we adopted the Sastri correlation for further analysis.

The Mixing Rule

After finding the best correlation for each pure component (Missenard correlation for the absorbent and Sastri correlation for the refrigerant), the four mixing rules were tested: The power low method, the Jemeison method, the Fillipov method and the Li method. The Latini & Beroncini prediction method was also tested. Using the Li method as the mixing rule, results the largest error, 30%, but it is yet an acceptable error at heat transfer problems. The maximum error of the HTC reduced to 20% while using the Jameison method error and than to 15% while using the Fillipov method and the power low method. The averages of the relative errors were 13.3%, 9.4% and 7.8%, respectively with standard deviations of 5.5%, 5.3% and 4.9%, respectively. Therefore, it was concluded that the best way to predict the thermal conductivity of the mixture R22–DMAC is by using the Power Low Method as the

mixing rule together with the Missenard correlation for the absorbent and the Sastri correlation for the refrigerant.

In order to improve the prediction of the mixture's thermal conductivity, we modified the Fillipov mixing rule. The parameter α , changed from 0.72 to 0.8 and instead the concentrations product, x_1x_2 , we used its square root, $\sqrt{x_1x_2}$:

$$k_m = x_1k_1 + x_2k_2 - 0.8\sqrt{x_1x_2}(k_2 - k_1) \quad (8)$$

Fig. 4 presents a comparison between the predicted HTC, while using the Power Low Method or the Modified Fillipov Method, and the experimental HTC. As can be seen, the error range of the HTC while using the modified Fillipov method, as the mixing rule, results in an error range of 10%, with an average relative error of 5.3% and standard deviation of 3.6% which is better than those obtained with the Power Low Method. Note that the power low method is a pure predicted method, while the modified Fillipov method is based on the experimental data.

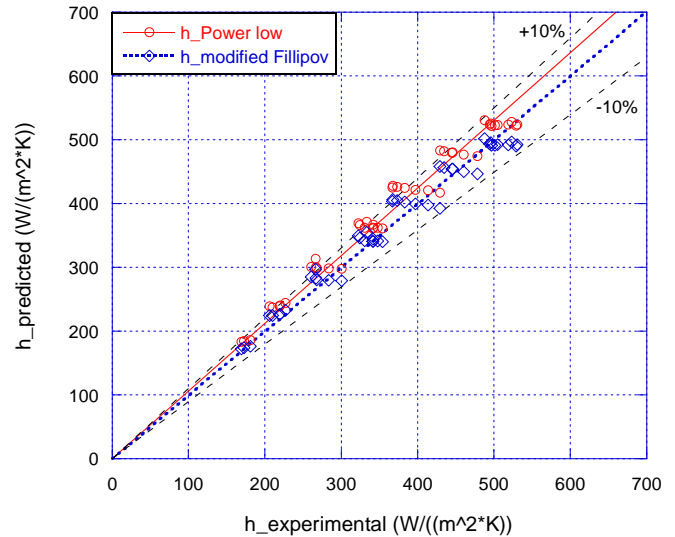


Figure 4 - Comparison between the experimental HTC vs. the predicted HTC, while using Missenard correlation for the absorbent, the Sastri correlation for the refrigerant with the power low and modified Fillipov mixing rules.

SUMMARY AND CONCLUSIONS

In this paper the experimental HTC of the binary mixture of R22-DMAC was compared to the predicted HTC were the only unknown parameter was the thermal conductivity of the mixture. The thermal conductivity of the binary mixture of R22-DMAC was estimated. Several thermal conductivity correlations for pure component and mixing rules were tested. First, all options of pure component thermal conductivity correlations and mixing rules were tested for the investigated solution (R22 - DMAC). Then the best correlations for predicting the pure components thermal conductivity were chosen, by comparing its predicted values with literature values. The

best thermal conductivity correlation for DMAC found to be the Missenard correlation. The best thermal conductivity correlation for R22 found to be the Sastri correlation. Then all mixing rules were tested and the Power Low Method found to be the best with an error range of 15%. A modified Fillipov mixing rule was suggested (Eq. 8). The predicted HTC calculated by the modified Fillipov mixing rule decreases the maximum error to about 10%.

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