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# Heating of Subcooled Refrigerants Flowing inside Porous Tubes Study of Heat Transfer: $CO_2$ Case Study

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## Abstract

Experimental study for the heat transfer characteristics in a porous tube was conducted in this work. The tubes were filled with sand beads. Three different particles mean diameters were used, (dm = 5.15 mm, 3.18 mm, 2.03 mm). Liquid $CO_2$ flowed inside tubes placed in laboratory conditions. The tube out wall temperature in 32 points with longitudinal steps was measured. The pressure at inlet was also measured with the rate of the flow of the gas. Heating tube exit was saturated liquid conditions. Results of these experiments were used to calculate the inside tube heat transfer coefficient.

An analytical investigation besides the experimental work was carried out for the heating of the sub cooled $CO_2$ liquid as a single phase refrigerant. An empirical correlation for the coefficient of convectional heat transfer coefficient was formulated as non dimensional relation:

$$Nu_{De} = 0.19 * (Re_a)^{0.35} (Pr)^{0.4} (\varepsilon)^{-1.39}$$

A comparison between experimental results and that obtained by this developed correlation was carried out. Comparison also was conducted between correlated results and those calculated using the correlation found in Mills 1995, for flow of liquids in porous tubes. The results of this research showed that for heating processes the proposed correlations were proved to possess low deviations. The deviation from the experimental results reached a maximum value of about 4.7 %.

This work can enhance the calculations of heat flux of liquid flowing inside porous media, and can help in the design procedure and practical applications of heat exchangers.

***Keywords:*** *Heating, Porous media, Convection heat transfer, R744*

## 1. Introduction

Considerable emphasis has been placed on the development of various heat transfer surfaces and devices. This can be seen from the exponential increase in world technical literature published in heat transfer augmentation devices, growing patents and hundreds of manufacturers offering products ranging from enhanced tubes to entire thermal systems incorporating enhancement technologies. Energy and material saving considerations, space considerations as well as economic incentives have led to the increased efforts aimed at producing more efficient heat exchanger equipment through the augmentation of heat transfer. Among many techniques investigated for augmentation of heat transfer rates inside circular tubes, a wide range of inserts have been utilized. The utilization of porous inserts has proved to be very promising in heat transfer augmentation. One of the important porous media characteristics is represented by an extensive contact surface between solid and fluid surfaces. The extensive contact surface enhances the internal heat exchange between the surfaces and consequently results in an increased thermal diffusivity. Heat transfer and fluid flow inside porous tubes have many applications such as; heat exchangers, condensers, evaporators and boilers are examples of these applications. Experimental studies conducted for heat transfer of laminar flow in horizontal tubes with/without longitudinal inserts was carried out by Hsieh and Hwang (2003). They reported that enhancement of heat transfer as compared to a conventional bare tube at the same Reynolds number to be a factor of 16. The experimental investigations of Hsieh and Liu (1996) reported that Nusselt numbers were between four and two times the bare values at low Re and at high Re respectively.
Literature also shows many recent studies of heat transfer for single phase flow inside tubes and channels with or without porous media. Examples for that: Bejan (2004), Liou (2005), Incropera (2007); Hammad and Alshqirate (2009) and Alnimer and Alkam (1997).

Following is the common used Whitaker Equation correlation formula for single phase flow inside porous tubes, Mills (1995). This correlation was used for comparison.

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$$Nu_D = (0.5 * Re^{1/2} + 0.2 * Re^{2/3})\, Pr^{1/3} \quad Re<10000 \qquad (1)$$

The analytical method was used to formulate the heat transfer empirical equation. This was conducted for liquid flow inside porous tubes. Parallel to that, experimental work was carried out for the same purpose using sub cooled Carbon Dioxide as liquid to flow inside a 1.73 cm in diameter porous tube.

## 2. EXPERIMENTAL WORK

Figure1 presents a schematic diagram of the experimental test rig. Three main parts are shown: a) Carbon Dioxide supply system, b) Porous test section, and c) Measurement and data acquisition system.

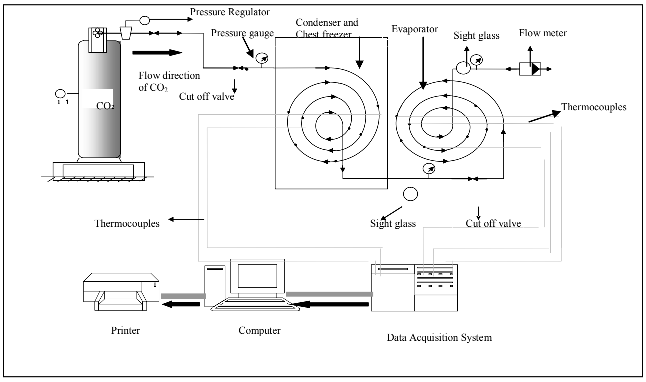


Fig.1: Schematic diagram of the experimental test rig

Thirty two K-type thermo couples were installed on the out side surface of the tube wall. These thermo couples were used to measure the out side wall surface temperature and the ambient temperature during heating and evaporation processes. They connected to a module of 32 channels, which is in turn plugged in the data acquisition system of model SCXI-1000, manufactured by National Instruments Company. The well-known LAB VIEW soft ware was used for the processing of the signals into temperature readings.

For each experimental test run, the variation of the temperature with time can be monitored. The temperature readings were recorded repeatedly for a certain period until the steady state conditions were achieved. The temperature readings were plotted against the length of the porous tube as shown in Figure 2.


Fig. 4: Gravel samples.

The grain size, porosity, density, permeability and thermal conductivity of the gravel were determined experimentally in the labs of the faculty of engineering and technology in the University of Jordan. Table 2 shows the sand media configurations.

Table 2: sand media configurations

| ε (%) | $V_T$ ($m^3$) *$10^{-4}$ | Mass (kg) | $\rho_b$ (kg/ $m^3$) | $\rho_s$ (kg/ $m^3$) | $d_m$ (m)*$10^{-3}$ |
|---|---|---|---|---|---|
| 39.8 | 714 | 1.13 | 1583 | 2630 | 5.15 |
| 43 | 714 | 1.076 | 1507 | 2630 | 2.03 |
| 44.5 | 714 | 1.047 | 1466.8 | 2630 | 3.18 |

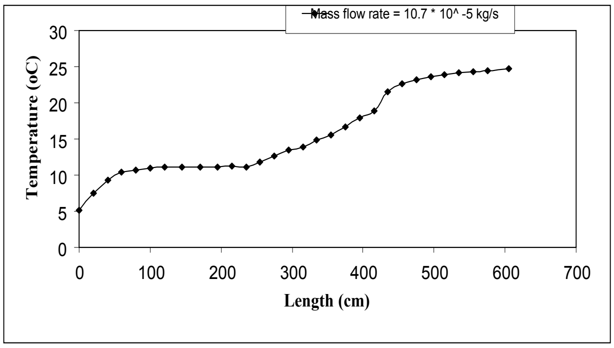


Fig. 2: Out side wall surface temperature in $^oC$ versus test section length for $P_i$ = 4300 kPa, porosity = 39.8% and mass flow rate = $10.7*10^{-5}$ kg /s for evaporation process.

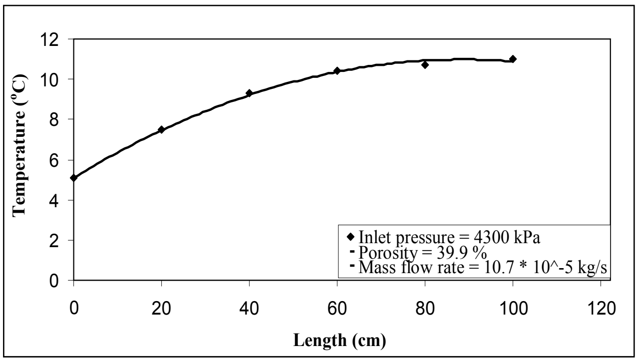


Fig.3: Out side wall surface temperatures vs. porous tube test section length during liquid heating process.

The experimental conditions used in this study covered a domain shown in Table 1.

Table 1: Experimental conditions

| | |
|---|---|
| Process | Heating outside at the ambient temperature (24-26 $C^o$) until saturation. |
| Working fluid | $CO_2$ |
| Mean diameter of the particles of the sand($d_m$), (mm) | 5.15, 2.03, 3.18 |
| Permeability(κ), ($m^2$) | $4.65*10^{-12}$, $5.7*10^{-12}$, $8.2*10^{-12}$ |
| Porosity of sand | 39.8 %, 43 %, 44.5 % |
| Test section total length, (m) | 4 |
| Test section inlet pressure, (kPa) | 3700, 4000, 4300, 4600 |
| Volume flow rate, (Liter / min) | 4, 5, 6, 7 |

Local gravel that consists of 98 % silica was bought and cleaned in the lab in order to be used as a porous media. This material is readily available in the markets. From randomly selected samples, three different grain size ranges were obtained and used, Figure 4 shows these samples.

## 3. CALCULATIONS

### 3.1. Apparent properties

Apparent physical properties are preferred to be used in dealing with flow in porous media such as: apparent thermal conductivity, ($K_a$), apparent heat capacity, ($C_{pa}$) and apparent viscosity, ($\mu_a$).
The apparent thermal conductivity and heat capacity of the porous bed comes as a result of the effect of solid material and that of the fluid. Both can be given by the following equations:

$$K_a = \varepsilon K_m + (1-\varepsilon) K_s \tag{2}$$

$$C_{pa} = \varepsilon C_{pm} + (1-\varepsilon) C_{ps} \tag{3}$$

While the apparent viscosity which was enlarged due to the increase in its effect by the increase of the contact area between the flowing fluid and the stationary porous media. It can be calculated by the following relation:

$$\mu_a = \gamma \mu \tag{4}$$

Where $\gamma > 1$ and equals $[1+ (1-\varepsilon) D/d]$, this can be called viscosity increase factor.
Thus the apparent Reynolds number, Prandtl number, and Nusselt number can be calculated as follows:

$$Re_a = \rho_L * u_L * d_m * \varepsilon / \mu_a (1-\varepsilon) \tag{5}$$

$$Pr_a = C_{pa} \ \mu_a / K_a \tag{6}$$

$$\text{Where } D_e = d_m \varepsilon / (1-\varepsilon) \tag{7}$$

$$\text{And, } Nu_a = h D_e / K_a \tag{8}$$

### 3.2. Heat Transfer

The tube outside surface temperatures at points along the whole test sections (about 4 m), were measured by means of K-type thermocouples fixed on the outer surface at longitudinal locations. These temperatures were tabulated along with the test section length. Different experiments were carried out changing independent variables: the porosity, ε, (three different values), the test section inlet pressure, $P_{in}$, (four different values), and the rate of flow, V, (four different values).
The following two equations (10 and 11) will be used to calculate the experimental mean heat transfer coefficient of $CO_2$ at the inner surface flow of the tube.

$$Q_{CO2} = \dot{m} * Cp_a * (T_1 - T_2) \tag{9}$$

$$Q_{CO2} = h_i * A_i * \Delta T_{lmi} \tag{10}$$

Where $\dot{m}$ is mass rate of flow of liquid, $T_1$ and $T_2$ are the liquid inlet and outlet mean temperatures, $\Delta T_{lmi}$ is the inner logarithmic mean temperature difference, $A_i$ is the inner surface tube area, $h_i$ is the inner experimental mean heat

## 4. ANALYTICAL WORK

### 4.1. Convection Heat Transfer

The proposed correlation for convectional heat transfer coefficient based on the apparent properties can be formulated for laminar flow in porous tubes as follows:

$$Nu_a = C (Re_a)^m (Pr_a)^n (\varepsilon)^l \tag{11}$$

Where C is a constant and m, n and l are exponent constants of the equation.
The experimental data was correlated over the range of the $Re_a$ and $Pr_a$ values considered within this work domain, and the following correlation was formulated:

$$Nu_a = 0.19 (Re_a)^{0.35} (Pr_a)^{0.4} (\varepsilon)^{-1.39} \tag{12}$$

## 5. RESULTS DISCUSION

Figure 5 compares the experimental heat transfer coefficient with the correlated heat transfer coefficient of this work and the correlated literature heat transfer coefficient of Mills (1995). It can be noticed from this figure for the values of this work heat transfer correlation agrees with both the experimental results and those calculated using literature correlation of Whitaker, stated in Mills (1995) with an average standard deviation of a bout 4.7%.

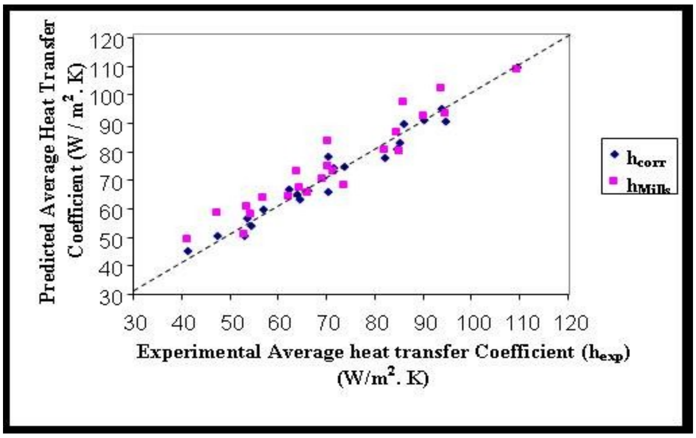


Fig. 5: Experimental heat transfer coefficient, $h_{exp}$ Vs two correlations calculated values

Figure 6 compares the predicted heat transfer coefficient of this work and the correlated literature heat transfer coefficient of Mills (1995). It can be noticed from figure 6 that values of this work correlation agree with the literature correlation with an average standard deviation of a bout 4.1%.

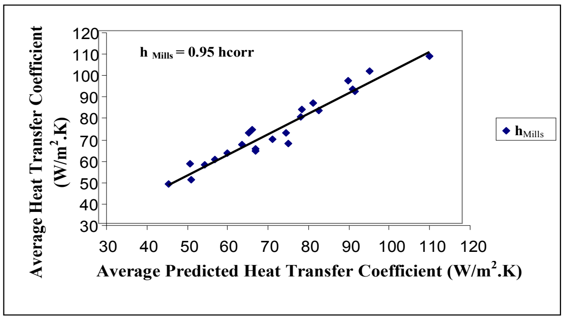


Fig. 6: Average Predicted heat transfer coefficient, $h_{corr}$ Vs Mills heat transfer coefficient

## 6. CONCLUDING REMARKS

This work experimentally explores the heating of sub cooled refrigerants and the heat transfer mechanism in a porous tube taking Carbon Dioxide as a case study. The average heat transfer coefficient and Nusselt number distributions at various positions were obtained for various particles Reynolds numbers. Finally, an empirical heat transfer correlation equation under various conditions was presented. The following conclusions are drawn.

1. Empirical correlation between the average Nusselt number and the Reynolds number based on the apparent properties of Carbon Dioxide and sand bead was developed which can help in the design of the related devices, including compact heat exchangers.

2. It is clear from the results for both heat transfer coefficient and Nusselt number, that the values of this work correlation agree with both the experimental results and those ,calculated using literature correlation of Whitaker (Mills, 2006), with an average standard deviation of a bout

### Nomenclature

| | |
|---|---|
| $Re_D$ | Reynolds Number,( ρ V D/μ) |
| L/D | length/diameter for tubes |
| Pr | Prandtl number, ($C_p$ μ/K) |
| Nu | Nusselt number,( hd/k) |
| T | Temperature (K or $^oC$) |
| P | Pressure (kPa) |
| m | Mass flow rate (kg/s) |
| h | Heat transfer coefficient ( kJ/$m^2$K) |
| κ | Permeability |
| Pe | Pecklet number, Re Pr |
| d | Particle diameter (mm) |
| V | Volume flow rate (l / min) |
| A | Cross sectional area ($m^2$) |
| *Latin*: | |
| Δ | Delta |
| γ | viscosity increase factor |
| μ | dynamic viscosity (m .s) |
| ε | Porosity |
| *Subscripts*: | |
| lmtd | log. mean temp. differ |
| i | Inner |
| o | Outer |
| a | Apparent |
| m | mean value |
| T | Total |
| s | Solid |

## ACKNOWLEDGMENTS

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