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Study of Heat Transfer for Superheated Refrigerants Flow Inside Micropipe Heat Exchanger.

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Abstract: In this work, the heat transfer characteristics of super heated (single phase) refrigerants boiled inside micropipe heat exchangers were studied experimentally and analytically. Carbon dioxide gas (natural refrigerant) was used as a refrigerant. Three micropipe heat exchangers with different internal diameters were tested experimentally. The boiling and superheating processes of the refrigerant were conducted in still room temperature air. An empirical correlation for the convection internal heat transfer coefficient was formulated Correlation was validated against some experimental results and compared to all experimental results and other literature correlation; an agreement of more than 90 % was noticed. This work can enhance the calculations of heat flux of gases flow inside mini and micro tubes and it can also help in the design procedure and practical applications of heat exchangers.

Key words: Superheating, Micro pipe, Convection heat transfer, R744.

INTRODUCTION

Energy and materials saving considerations, space considerations as well as economic incentives have led to the increased efforts aimed at producing more efficient heat exchanger equipment. One of these techniques which are used for augmentation of heat transfer rates is using of micropipe heat exchangers. Due to several environmental problems, there is an increasing trend towards the use of natural working fluids in various power, heating and cooling applications.

Kumar and Gopal, (2009) made a comparative study between carbon dioxide and conventional secondary fluids that can be used in natural circulation loops for various refrigeration and air conditioning applications. Results are presented for both laminar and turbulent flow conditions. They observed that due to its excellent thermo-physical and transport properties and near critical point operation, carbon dioxide gives rise to highly compact systems when compared to most of the conventional secondary fluids.

Experimental and analytical studies conducted by Hammad and Alshqirate, (2009) for heat transfer and pressure drop of laminar flow in horizontal micropipes heat exchangers. They studied cooling of flow inside mini and micro tubes. Tarawneh *et al*. (2011) studied experimentally the heat transfer characteristics in a porous tube, the tubes were filled with sand beads and CO_2 was used as working fluid.

Olson, (1999) reported measurements of heat transfer coefficients of flowing supercritical carbon dioxide in a heated horizontal tube. He found that at lower pressures, conditions of high mass flow and low heat flow enhanced the heat transfer, while conditions of low mass flow and high heat flow degraded the heat transfer

Park and Hrnjak, (2007) studied CO_2 and R410A flow boiling heat transfer, pressure drop, and flow pattern at low temperatures in a horizontal smooth tube.

Thome and Ribatski, (2005) introduced state-of-the-art of two-phase flow and flow boiling heat transfer and pressure drop of CO_2 in macro- and micro-Channels.

Alshqirate (2012) examined the effect of heat exchanger type on two phase heat transfer coefficient and pressure drop by designing three different types of heat exchangers and tested them experimentally to investigate two-phase heat transfer coefficient and pressure drop during the condensation process.

Literature also shows many recent studies of heat transfer for single phase flow inside micro and macro tubes and channels, for example: Bejan, (2004), Liou, M. (2000), Incropera *et al*. (2007).

Following is the known published Colburn Correlation. This correlation is applicable for single phase flow inside tubes. It was used in this work for the purpose of comparison issues, Incropera *et al*., (2007):

$$Nu_D = 0.023\,(Re_D)^{4/5}\,(Pr)^{1/3} \tag{1}$$

The experimental work was carried out to find the convection heat transfer coefficient of super heated CO_2 gas using evaporated carbon dioxide gas flow inside micropipe heat exchanger. Analytical method was used to formulate the heat transfer empirical equation in terms of Nusselt number. A comparison between experimental and analytical results was carried out for validation purposes. Comparison with literature correlations was also shown.

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Experimental work and Procedures:

The test apparatus and its main components are shown schematically in Figure 1. The experimental set-up consists basically of: 1- The pressurized CO_2 cylinder as a main source of carbon dioxide gas. 2- High pressure regulating valve with built-in gas cylinder pressure gauges. 3- Chest freezer used for cooling the carbon dioxide gas. 4- The micropipe heat exchanger (condenser). 5- High pressure cutoff and isolating valves. 6- Pressure gauges. 7- Sight glasses. 8- The micropipe heat exchanger (evaporation and super heating). 9- Data Acquisition System (*DAS*). 10- Volume flow meter for measuring the mass flow rate of the gas.

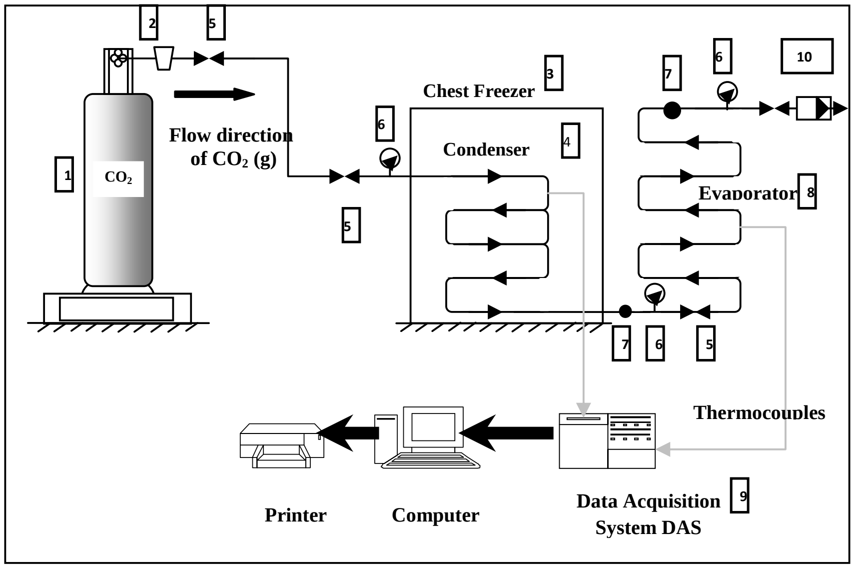

Fig. 1: Schematic diagram of the experimental test rig.

This schematic diagram was used to achieve two parts; first part conducted inside a chest freezer: cooling, condensation and sub cooling. Second part conducted in still room temperature air out side the chest freezer: heating, evaporation and superheating.

K-type thermocouples were used in order to measure the outside wall surface temperature of the micropipe heat exchanger during evaporation and super heating processes and the ambient temperature. Those thermocouples was connected to a module of 32 channels, which is in turn plugged in the data acquisition system of model SCXI-1000, The well-known (LAB VIEW) soft ware is used for the processing of the signals of the thermocouples.

For each experimental test run, the variations of the temperature with time were monitored until the steady state conditions are achieved, and then the temperatures were recorded and plotted against the micropipe heat exchanger length as shown in Figure 2.

The pressure was recorded in three positions at steady state conditions; they were at bottle exit, after the cooling section and at exit of the heating section.

Volumetric rate of flow in m^3s^{-1} was read at the end outlet flow by a gas flow meter calibrated for CO_2.

Experiments were conducted for different pipe diameters, at different pressures, and at settings of different rates of flow.

The different experimental conditions used in this study are listed in Table 1.

Table 1: Experimental conditions.

Test sections	Three Micropipe heat exchanger
Process	Super heating in still room temperature air
Working fluid	CO_2
Micropipe internal diameters, D_i [mm]	0.6, 1.0,1.6
Test section total length [m]	16
Test section inlet pressures, [kPa]	3350, 3600, 4000, 4500
Saturation temperatures, [^oC]	-1.5, 1.23, 5.30, 9.98
Volume flow rates, [$l\ min^{-1}$]	1, 2, 3, 4

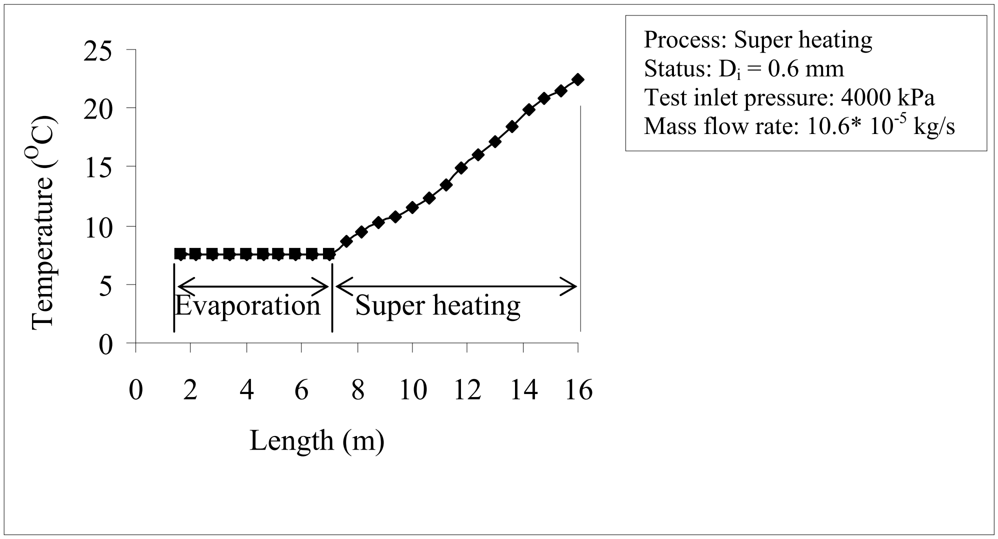

Fig. 2: Outside wall surface temperatures in (^{o}C) versus micropipe test section length during evaporation and super heating processes in still room temperature air.

Calculations of Experimental Heat Transfer:

In order to calculate the heat absorbed by the gas while super heating, different experiments were carried out. The experimental independent variables were: the micropipe internal diameter, D_i, (three different values), the test section inlet pressure, Pin, (four different values), and the rate of flow, $\dot{V}$, (four different values). The tube outside surface temperatures at 25 points along the whole test sections of about 16 m, were measured by means of K-type thermocouples fixed on the outer surface at longitudinal locations.

The heat flow formed a radial in ward heat flux. Free convection and conduction heat transfer occurred. Heat balance for the heat transfer in still room temperature air was modeled by the following equations:

$$\dot{Q}_{CO_2} = h_o\, A_o\, \Delta T_{lmo} \tag{2}$$

Where the outer logarithmic mean temperature difference equals:

$$\Delta T_{lmo} = [(T_1 - T_a) - (T_2 - T_a)] / \ln[(T_1 - T_a) / (T_2 - T_a)] \tag{3}$$

$$\dot{Q}_{CO_2} = \dot{m}_{CO_2}\, C_p\, (T_2^{/} - T_1^{/}) \tag{4}$$

$$\dot{Q}_{CO_2} = h_i\, A_i\, \Delta T_{lmi} \tag{5}$$

And the inner logarithmic mean temperature difference equals:

$$\Delta T_{lmi} = [(T_1^{/} - T_1) - (T_2^{/} - T_2)] / \ln[(T_1^{/} - T_1) / (T_2^{/} - T_2)] \tag{6}$$

Where: T_1 and T_2 are the first and last temperatures of the wall outside surface, $T_1^{/}$ and $T_2^{/}$ are the gas inlet and outlet mean temperatures and T_a is ambient temperature. A_o and A_i are the external surface and the internal surface micropipe area, respectively. The external and internal convection heat transfer coefficients are h_o and h_i respectively. $\dot{m}_{CO_2}$ is the gas mass rate of flow.

Equation 2 will be used to calculate the heat quantity using Churchill and Chue formula to calculate h_o, the formula is: (Incopera, *et al*, 2007).

$$Nu_D = \{0.60 + ((0.387\, Ra_D^{1/6}) / [1 + (0.559/Pr)^{9/16}]^{8/27})\}^2 \tag{7}$$

Equation 4 will be used to calculate the mean gas flow temperature at exit, ($T_2^{/}$), as the gas temperature at inlet is known to equal saturation temperature at the measured pressure.

Then equation 5 will be used to calculate the convection heat transfer coefficient of CO_2 at the inner surface flow of the tube. In this step conduction heat transfer through the tube wall was neglected.

These results will be calculated and tabulated as the experimental convection heat transfer coefficient, h_{exp} and the Nusselt number as Nu_{exp} for super heating gaseous CO_2 flow inside micro tubes.

Analytical Work:

In literature Colburn equation was used as a basic equation to calculate the convectional heat transfer coefficient as mentioned previously. So, the proposed correlation can be formulated as follows:

$$Nu_D = C\, Re_D^{m}\ \ Pr^{n} \tag{8}$$

Where: C is a constant; (m and n) are exponent constants.

The experimental data was correlated over the range of the calculated Re_D and Pr values considered within this work domain.

As a result, the values of the constants: C; m; and n were evaluated to be: 0.022; 0.73; and 0.48 respectively. The correlation for heat transfer relation between Nu_D, Re_D and Pr for CO_2 super heated gas was formulated in the form:

$$Nu_D = 0.022\, Re_D^{0.73}\, Pr^{0.48} \tag{9}$$

The values of convection internal heat transfer coefficient resulted out of this correlation were tabulated as $Nu_{corr.}$

RESULTS AND DISCUSIONS

1- Figure 3 compares the experimental heat transfer coefficient results with the correlated results of this work in equation 9. From this figure it's found that the values of the heat transfer correlation agrees with the experimental results with an average standard deviation (ASD) of a bout 2.2 %.

2- Figures 4 and 5 compare the experimental heat transfer coefficient with two other correlated values: this work correlation in equation 9 and the Colburn correlation in equation 1. From these figures, it can be noticed that the values of this work correlation are more closely to the experimental results than that resulted from the literature correlation. Comparing both correlations with each other, difference was noticed. The difference was calculated to equal 1.4 : 1.0, the Colburn Nu to this work Nu respectively. The difference noticed to be due to the exponent of Pr value which raised from 0.33 in Colburn’s to 0.49 in this work. Prandtl number is a physical property, not affected by flow parameter.

3-Figures 6 and 7 show the effect of the Reynolds number and the Prandtl number on the convection heat transfer coefficient. The heat transfer coefficient increases as the two numbers increase. Low kinematic viscosity of the gas and the turbulence effect due to high velocity cause the increase in the Reynolds number and Prandtl number which in turn enhance the heat transfer.

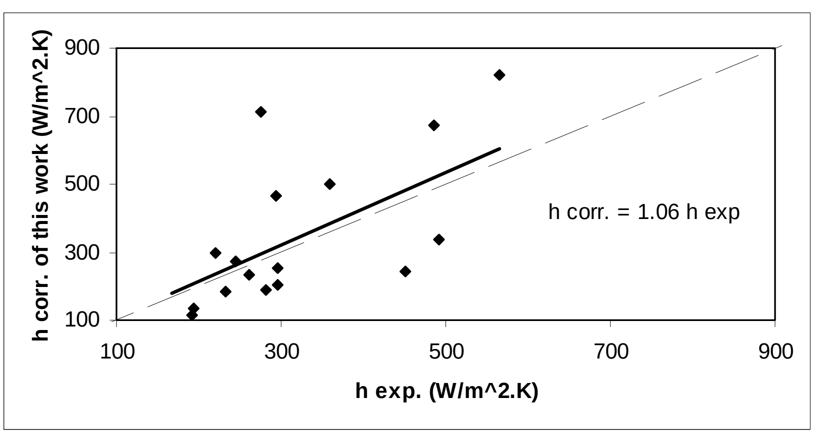

Fig. 3: Experimental heat transfer coefficient, $h_{exp.}$ Vs correlated values of this work.

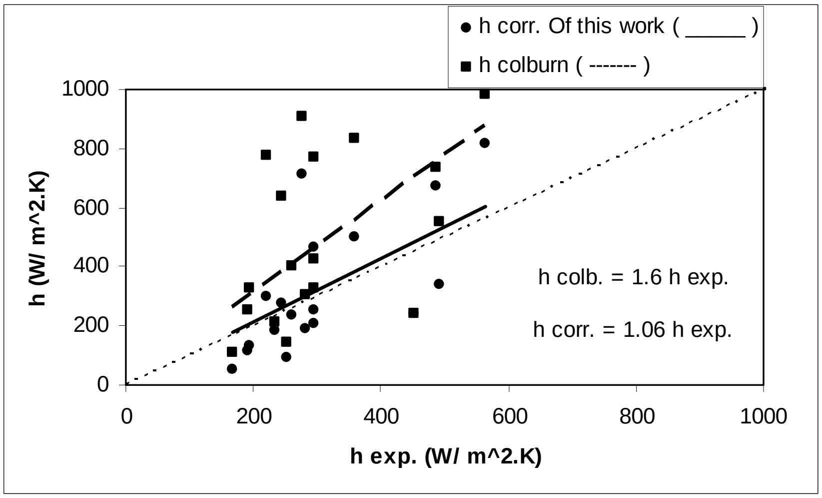

Fig. 4: Experimental heat transfer coefficient Vs two correlated values.

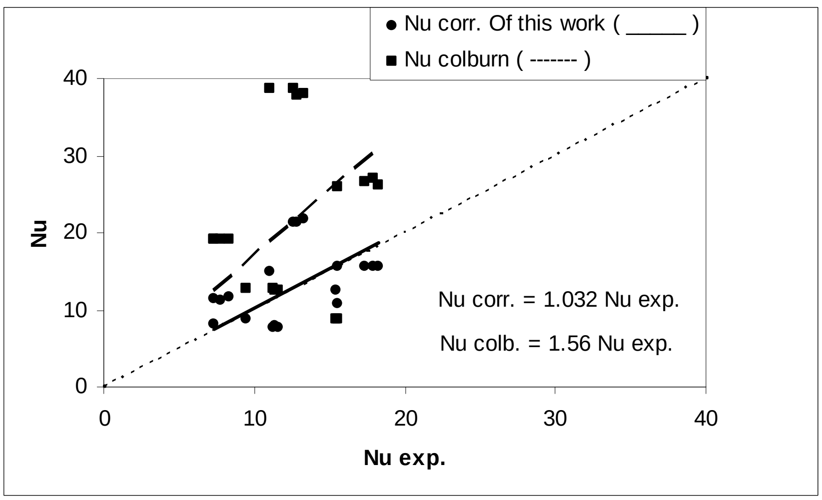

Fig. 5: Experimental Nusselt numbers Vs two correlated values.

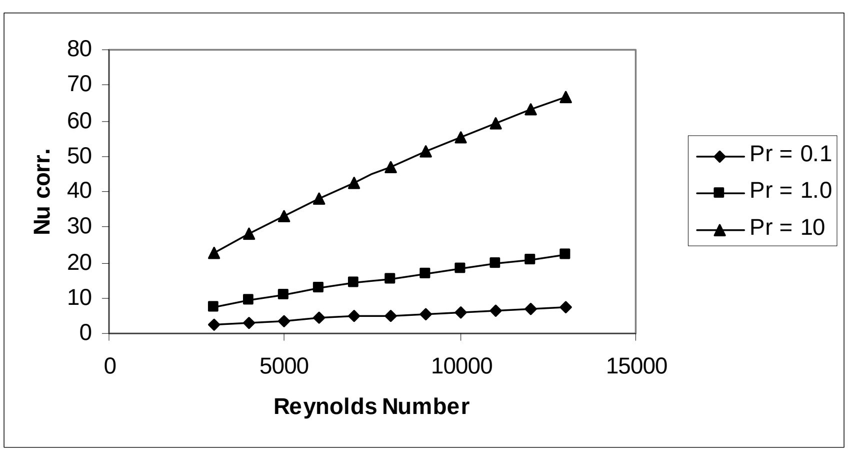

Fig. 6: Reynolds number Vs. correlated Nusselt number at different Prandtl numbers.

Concluding Remarks:

Super heating process over gases was studied experimentally taking Carbon Dioxide as a case study. The convection heat transfer coefficient was correlated for flow in a micro tube heat exchanger under various conditions. The following remarks were concluded:

* Simple empirical correlation was formulated for the Nusselt number of convection heat transfer coefficient calculations during super heating of gases flow inside micro tubes. This can help in the design of the related devices, including compact heat exchangers.

* The agreement of this work correlation with experimental results in convection heat transfer coefficient and Nusselt number is about 33 % closer than that of Colburn.

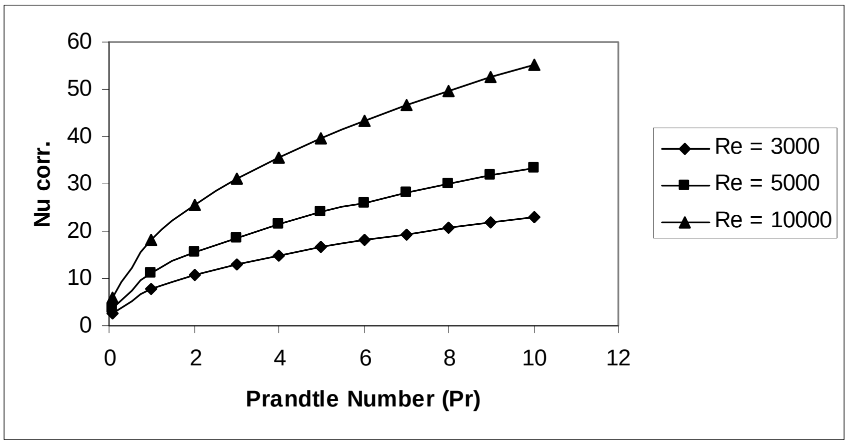

Fig. 7: Prandtl number Vs. correlated Nusselt number at different Reynolds numbers.

Nomenclature			
ASD	Average Standard Deviation	*Latin symbols*	
A_i	Micropipe internal surface area [m^2]	ΔT_{lm}	Logarithmic mean temperature difference [^oC]
A_o	Micropipe external surface area [m^2]	$\dot{Q}_{CO_2}$	The heat transfer released from the gas [W]
CO_2	Carbon dioxide	$\dot{V}$	Volume flow rate [l min^{-1}]
C	Constant	$\dot{m}_{CO_2}$	The gas mass rate of flow [kg s^{-1}]
DAS	Data Acquisition System	*Subscripts*	
D_i	Micropipe internal diameter [m]	*D*	diameter
h_i	Convection internal heat transfer coefficient [W m^{-2} K^{-1}]	*a*	ambient
h_o	Convection external heat transfer coefficient [W m^{-2} K^{-1}]	*corr*	Correlated (predicted)
$h_{corr.}$	Correlated heat transfer coefficient[W m^{-2} K^{-1}]	*exp*	Experimental
h_{exp}	Experimental heat transfer coefficient[W m^{-2} K^{-1}]	*g*	Gas
Nu_D	Nusselt number	*i*	Internal
$Nu_{exp.}$	Experimental Nusselt number	*in*	Inlet
$Nu_{corr.}$	Correlated (predicted) Nusselt number	*lm*	Logarithmic
P_{in}	Test section inlet pressure [kPa]	*m*	Mean
Pr	Prandtl number	*o*	external
Ra_D	Rayleigh number	*Superscipts*	
Re_D	Reynolds number	m, n	Power constants

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