Journal of Engineering Science and Technology
Vol. 16, No. 4 (2021) 3407 – 3420

CONVECTION HEAT TRANSFER PERFORMANCE FOR THE SCF-CO_2 MEDIA IN MINI-TUBE WITH FINS EXPERIMENTALLY

AMEER ABED JADDOA

Department of Electromechanical Engineering,
University of Technology, Baghdad, Iraq
E-mail: ameer.a.jaddoa@uotechnology.edu.iq

Abstract

In this work, a convection heat transfer has been used in horizontal tube which mixed with CO_2 at super-critical pressures media investigated experimentally. The tube dimensions were 10.8 mm for inner and outer diameter and 40 mm for length. Square, triangle and circular fins were installed in its at different dimensions under pressure ranged from 80 to 120 bar, 0.03 to 0.5 kg/min as flow rate of mass, and 37 to 77 ^{o}C as a temperature. Also, the test situation included on Reynolds numbers that ranged from ranged from 10^3 to 10^7 and Prandtl numbers which ranged from 1 to 12 with and without fins have been tested and analysed by measuring the coefficient of heat transfer, temperatures of fluid bulk and wall. Moreover, this work indicated the influence of mass flow rate, temperature of inlet and heat flux for miniature tube in addition to factor of friction, improvement efficiency, correlation for Nusselt number and criteria of performance evaluation. All these criteria contributed to evaluate the experimental rig in order to enhance the tube heat transfer. Finally, the results revealed that the circular fins superior on other fins shapes (triangle and square) based on heat transfer rates that recorded higher rates compared with square and triangle fins.

Keywords**:** Fins, Horizontal mini-tube, Super-critical carbon dioxide.

1. Introduction

Transferring of the heat in supercritical fluid media nowadays is considering one of the most important technologies in heat transfer field such as power plant applications especially when the fluid is water. When the conductors cooled by fluid to the closed critical points, it has called superconductivity. Many applications are using the supercritical fluids for cooling blade of the gas turbine, cables of power transmission, elements of supercomputer, etc. one of the famous ways to cool the military aircraft and rockets is a fuel as a supercritical fluid. Many complex industrial applications suffer from catalytic breakdown due to the effects of coke as well as side reactions simultaneously. Therefore, the supercritical media being one of the important processes to avoid the failure especially it contains many advantages s such as no tension in the surface, no phase interaction, and no capillary influence. Also, these benefits lead to porous structures penetration as well as to heterogeneous catalytic regeneration and to avoid synthesis problems [1-4]. Moreover, supercritical fluids are considered as a way to increase the roughness of surfaces and analyze the free surface energy of the tissue when be not wet and contact at high water WCO higher from 150^o [5].

Based on pervious works, an experimental heat transfer convection using a supercritical carbon dioxide in a heated miniature tubes in horizontal and vertical has been studied [6]. The results revealed that the Nusselt numbers reduced at all orientation of flow considerably, and the tube diameter reduced lower than 1.0 mm. Also, the outcomes indicated the developing in correlation of averaged Nusselt number to the heat transfer convection horizontally and vertically. Another study dealt with the convection heat transfer in vertical tubes using CO_2 at super-critical pressures. The tubes diameters were 0.948 and 4 mm, while the media of porous particle diameters ranged from 0.2 mm to 0.28 mm [7]. The changing in carbon dioxide thermo physical properties and critical buoyancy greatly affected the heat transfer convection in the mini-tubes vertically and media of porous compared with the empty tube without porous media. A same carbon dioxide as a supercritical fluid used to flow inside channels at supercritical pressures [8].

This paper classified the heat transfer modes into three modes which are normal improved and deteriorated heat transfer. The second mode was larger than the expected coefficient of heat transfer (HTC) for first mode regime, while the third mode was lower than the expected coefficient of heat transfer (HTC) for first mode regime. Small-channeled structures also investigated using supercritical CO_2 flow in order to characterize of the heat transfer [9]. Ten circular channels have been tested experimentally with 1.31 of diameter for multi ports of extruded aluminum. The higher influences have been recorded for CO_2 temperature, velocity of mass and pressure on heat transfer rates increasing especially in critical region. Therefore, the empirical correlation improved the forced convective heat transfer of supercritical CO_2 in horizontal micro/mini channels.

A comparative experiment study has been done for vertical heated pipes using supercritical fluid [10]. The pipe dimensions were 2 m of length at 4.4 and 6 mm of diameters. The system testes at 8.12MPa of pressure when the flux of mass and flux of wall heat were 400 kg/m^2 s, 400 kg/m^2 s, 1200 kg/m^2 s and 30 kW/m^2, 50 kW/m^2, 50 kW/m^2 respectively to be the same percentage for flux of wall heat on flux of mass. Also, ratio of length over diameter remained fixed during the test for normal, improved, and deteriorated heat transfer modes. Moreover, Because of the buoyancy, the flow at a greater diameter, it worked on reducing in heat transfer.

A horizontal mini-tube as triangular and circular section has been tested numerically by supercritical CO_2 in order to test the laminar mixed convective heat by getting the wall shear stress, velocity, and temperature distributions [11]. The proposed tubes configurator sharply and been equaled to the hydraulic diameters < 1.0 mm the test conditions were under cooling, and without and with gravity. The most important indication was the heat transfer has enhanced due to buoyancy near the pseudo critical point, but not under thermal equilibrium condition for wall of tube and fluid. An analytical examination has been investigated for influence of acceleration and buoyancy on mixed and forced CO_2 convective heat transfer. The examination was at supercritical pressures [12].

As a result, buoyancy and acceleration numbers have improved the correlation of heat-transfer with supercritical carbon dioxide heat-transfer extension. The predicted results of three correlations were accurate about 90 % compared with experimental data within error of ±30%. An experimental and numerical study examined the convection heat transfer of carbon dioxide at supercritical pressure. The particle diameters of sintered porous in vertical tubes ranged from 0.1 mm to 0.12 mm and 0.2 mm to 0.28 mm [13].

CO_2 friction factor results flowing in the heated sintered porous tube differ from those measured in the experiments both for upward and downward flows. The wall temperatures estimated was well agreed with the measured wall temperatures. In heat exchangers, other review paper viewed the experimental works of convection heat transfer correlations in order to obtain the coefficient of heat transfer at flowing of supercritical CO_2 under variant applications [14]. Also, the paper presented varied geometers and dimensions such as inclined, vertical, and horizontal tubes, mini-channels, and circular pipes as closed loop system.

The summary of the study called for more work in this field due to the no global correlation for each application. [15] was study analysis to improving thermosyphon (TPCT) thermal efficiency using nanoparticles/based fluids (water) The decrease in the temperature difference between the condenser and evaporator confirms these enhancements.[16] was study thermo-structural fatigue and lifetime analysis of a heat exchanger as a feedwater heater in power plant, the highest equivalent thermal stresses under these extreme load conditions occur at the joints of the tubes

The current work focused on using different fins in mini-tube by mixing convection heat transfer to supercritical carbon dioxide media. All fins' shapes investigated experimentally and compared with previous work by measuring the coefficient of heat transfer, temperatures of fluid bulk and wall. Moreover, this work indicated the influence of mass flow rate, temperature of inlet and heat flux for miniature tube in addition to factor of friction, improvement efficiency, correlation for Nusselt number and criteria of performance evaluation.

2. Experimental Work

A new experimental rig has been designed and developed in order to verify the convention heat transfer in miniature tube with fins under supercritical in supercritical carbon dioxide media as shown in Fig. 1, the experimental work included on high pressure expansion valve, manometer for high pressure, cylinder of compressed CO_2, liquid of refrigeration, helical heater, CO_2 pump, flow meter, thermocouple type T and Arduino microcontroller.

The specifications of valve are type of 2-Way Ball Material with Size of 3/8, made from chrome-plated steel, while the manometer model of MN10101 and ranged from 0 to 1000 Bar, Diameter of Indicator is 63 mm with 1.6 as accuracy measurement accuracy. The liquid specifications of refrigerator are ready Mixed -30ºC +125 German. The helical heater diameter is 6 mm and 800 ºC as a max. Temperature with 3.7 to 380 V as a supplied voltage that made from Nickel 800 and insulated by powder of magnesium oxide.

On the other hand, the operation pressure of CO_2 pump reaches to 10,000 psi with adaptable rate of flow 10 mL/min). Size of the flow meter 3/8 with 12 m/s as a velocity of fluid the length thermocouple sensor length is 96 cm and can read from -418 to 500 degree. Finally, the main important of Arduino are type of AT mega 328P, with 14 pins for input and output digital and 6 of pins are for PWM technique. Also, the Arduino has 6 of analog input pins and can supplied by 5Vand can provide 3.3V.

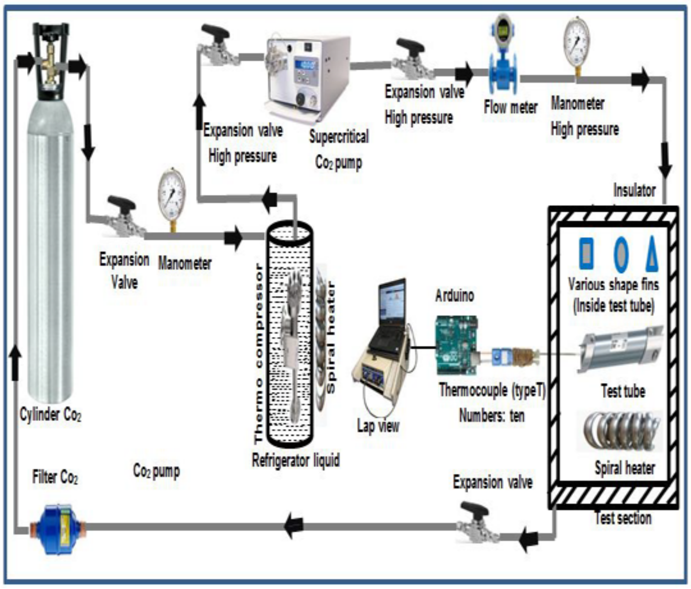

Fig. 1. Experimental work.

According to Fig. 2, two different types of circular tubes have been used one is made from stainless steel type 1Cr18N9T with 9 and 11 mm for inner and outer diameters respectively, while the second one made from copper with 8 and10 mm for inner and outer diameters respectively. Both tubes were empty and circular, triangle and square fins have located inside it with 2 mm of dimension. 10 of thermocouples have been located uniformly in order to measure the tube temperature surface among of 40 mm as a test heated length.

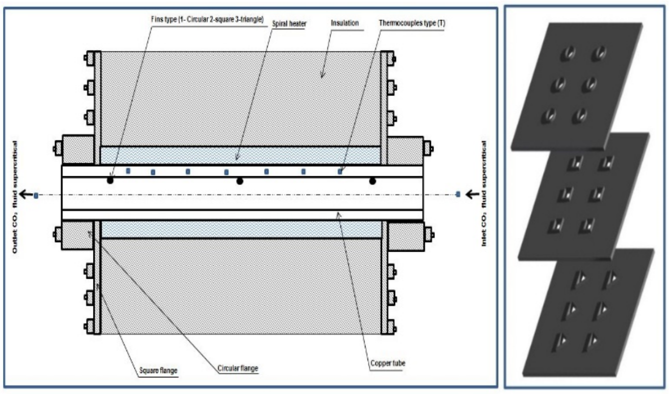

Fig. 2. Test section.

According to the steady state condition and balance of energy, the rate of heat transfer (Q^*) is calculation between the wall of tube and to the CO_2 as shown in equation below [17]:

$$Q^* = m^* \, (x_{out} - x_{in}) \tag{1}$$

where (m^*) is referring to the rate of mass flow, while ($x_{out} - x_{in}$) refers to the CO_2 enthalpy for the inlet and outlet test section. On the other hand, the equation below used to calculate the difference of (T^{**}) as:

$$T^{**} = [(t_w - t_{in}) - (t_w - t_{out})]/ln \, [(t - t_{in})/(t_w - t_{out})] \tag{2}$$

In the active heated length, coefficient of average heat transfer (h) as:

$$h = Q^*/AT^{**} \tag{3}$$

where *A* is refereeing to the tube inner surface and Nusselt number can be obtained by:

$$\mathrm{Nu}_b = hd/k_b \tag{4}$$

where (d) indicates to the diameter of tube, (k)indicates to the CO_2 thermal conductivity and (b) indicates to the CO_2 bulk mean temperature (T_b) which can be expressed as:

$$T_b = (T_{in} + T_{out})/2 \tag{5}$$

where, (T_{in}) is referring to the inlet temperature, while (T_{out}) is indicates to the outlet temperature.

Also, Reynolds number can be calculated by:

$$\mathrm{Re}_b = ud/v \tag{6}$$

where (u) and (u) are referring to kinematic viscosity and velocity, respectively for CO_2.

The factor of friction (C_f) can be expressed by:

$$C_f = \Delta p/[(l/d)(\rho u^2/2)] \tag{7}$$

where(Δp) and (l) are referring to inlet and outlet Pressure drop difference and test section heated length respectively.

When the Reynolds number is constant (CR), coefficient of heat transfer ratio of the fitted fins tube (h_i) over the tub plain (h_j) is called the enhancement efficiency (E_e) as shown in Eq. (8) [18] .

$$E_e = h_i/h_j \vdots CR \tag{8}$$

The influence of secondary flow that induced by buoyancy assumed negligible. Therefore, the Grashof number can be estimated by [19] and [18]:

$$\mathrm{Gr} = [gd^3(\rho_w - \rho_b)]/[(\nu_b^2\rho_b) \tag{9}$$

where $(\rho_w - \rho_b)$ are the CO_2 density at the wall and bulk temperature mean, (ν_b) kinetic and dynamic viscosities at the bulk mean temperature.

In order to calculate the mean specific heat and improve the correlations in heat transfer, the effect of temperature change or flux of heat $(T_w - T_b)$ between wall and bulk mean temperatures have been considered (ρ_b/ρ_w) , $c_{ba}/c_{bb})$ on the rate of heat transfer rate to be:

$$c_b = (X_w - X_b)/(T_w - T_b) \tag{10}$$

where (J_w) is referring to CO_2 enthalpy at wall mean, while the (J_b) is referring to CO_2 enthalpy at temperature bulk mean.

3. Validation of Plain Tube Data

Figure 3 shows the coefficient variant of heat transfer with Carbon dioxide temperature bulk mean for the current experimental work compared with previous work by Liao and Zhao [6]. Both works were for supercritical carbon dioxide media of mini-plain-tube. The results indicate that both of works present almost identical results to predict the results.

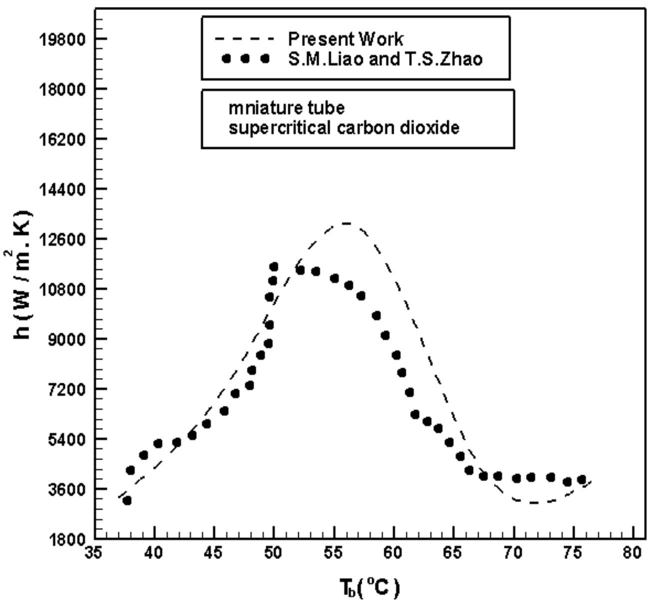

Fig. 3. Coefficient of heat transfer for present and previous work [6].

An experimental data for current work and predicted pervious works can be seen in Figs. 4 and 5. The well agreement between two works can be notable about ± 6.2% for Nusselt number, while the factor of friction was within ±3.4%.

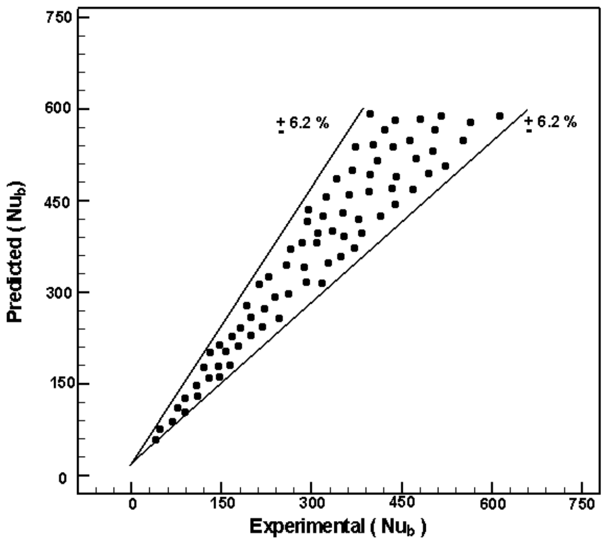

Fig. 4. Predicted Nusselt number and experimental.

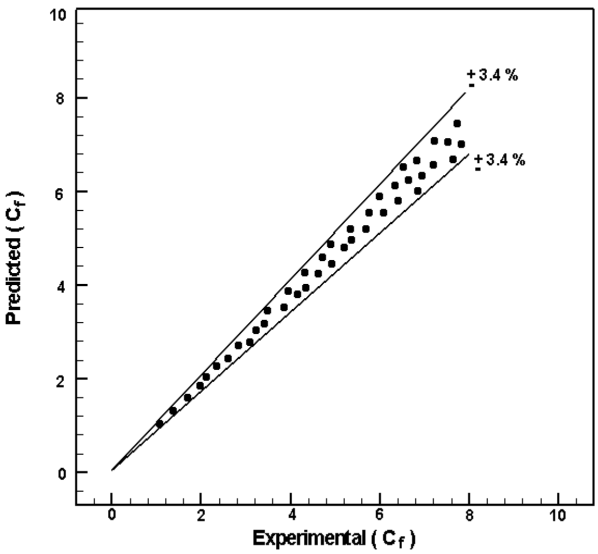

Fig. 5. Predicted Friction factor and experimental.

4. Results and Discussion

Based on variant rate of mass flow and static pressure, Fig. 6 presents the coefficient variant of heat transfer with CO_2 temperature bulk mean in horizontal tube. The main indication is the heat transfer peak value. Also, the enhancement in heat transfer can be noticeable near the corresponding pseudocritical temperature (T_b = 57.3 ºC) for each pressure.

Moreover, the specific heat behavior was variant in same style for the same region. Other notice was the heat transfer coefficient peak value has reduced when the static pressure raised from 80 to 120 bar due to decreasing the specific heat decreases and increasing in pressure. Finally, at 80 bar of pressure and 0.03 of mass flow rate, the beat value of heat transfer has been recorded. Decreeing in the turbulence that resultant by flow acceleration was and at Pressure 80 bar and mass flow rate 0.03, reduced turbulence as a result of flow acceleration because of sound heating that considers one of the main ways to the heat transfer due to the buoyancy.

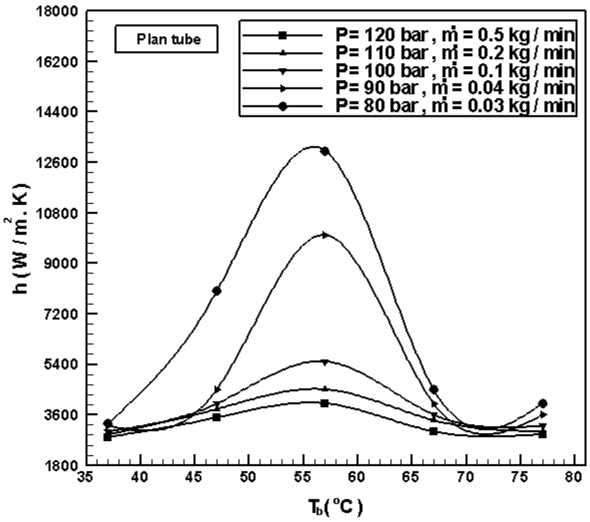

Fig. 6. Different static pressure and mass flow rate variation with coefficient of heat transfer for bulk mean temperature.

Figure 7 presents the coefficient variant of heat transfer with CO_2 bulk mean temperature in horizontal tube at 80 bar and 0.03 for pressure and mass flow rate respectively, with difference fins. The results revealed that the process of heat transfer has been enhanced due to fins. Also, in plain tube, the coefficient behaviour of heat transfer with fins superior on tube without fins generally, in addition to increasing the Nusselt number. It was also noticed that the effect of the fins (especially for circular fins) worked to swirl flow and distributed the pressure in the radial direction based on experimental data and made the boundary layer thinner throughout the wall of tube and caused to increasing the fluid heat flow that resulted due to turbulent flow and enhanced the convection heat transfer.

For horizontal flow, the influence of Nusselt numbers has been cleared in Fig. 8 at 80 bar of pressure, 103 to 107 for Reynolds number period and 1 to 12 for Prandtl number period with changing of fluid bulk mean temperature. The bulk mean temperature is dimensionless and equalled to T_{bc} which is views the CO2 pseudocritical temperature. At $0.8 < T_b / T_{bc} < 1.6$, the value of the Nusselt numbers with fins has been decreased for the triangular fin shape for supercritical CO_2 which led to the use of the circular shape fins.

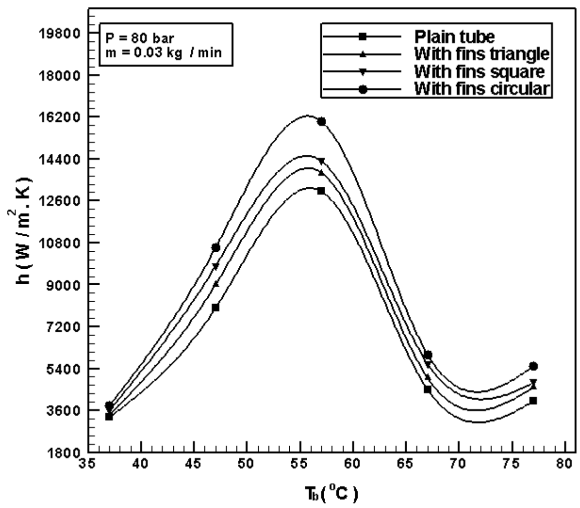

Fig. 7. coefficient of with heat transfer and bulk mean temperature variations for different fins shapes.

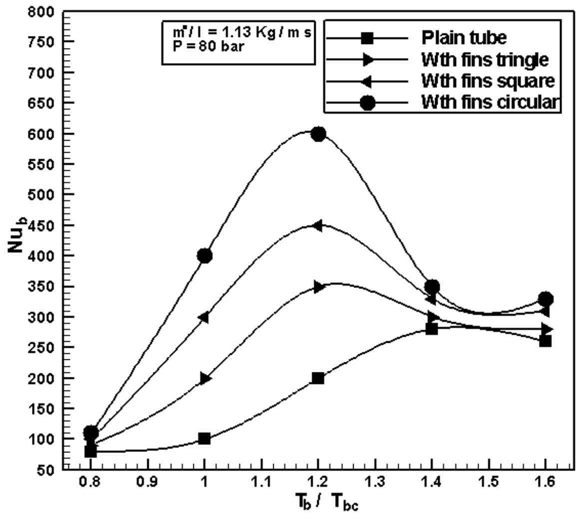

Fig. 8. Shapes of fins and effects on the Nusselt number.

Based on measured values of Nub form Fig. 8, Gr/Re^2 range will be examined. The value of Gr/Re^2 is upper than 10^{-3} at $T_b/T_{bc} < 0.8$, and at $T_b/T_{bc} > 1.6$ become lower than 10^{-3} for the rectangular and triangular fins as shown in Fig. 9. These values indicate that the buoyancy influences importance at low $T_b/T_{bc} < 1.6$, while the buoyancy influences importance reduced at $T_b/T_{bc} > 1.6$.

The convection correlations have been found in horizontal flow for mini-heated tube at supercritical CO_2 in for constant temperature depend on least square which the experimental data fit of 50 at different fins for the circular tubes.

$$\mathrm{Nu}_b/\mathrm{Nu}_{bc} = 5.38(\mathrm{Gr}/\mathrm{Re}_b^2)^{0.254}(c_b/c_{bc})^{0.319}(\rho_w/\rho_b)^{0.825} \quad (11)$$

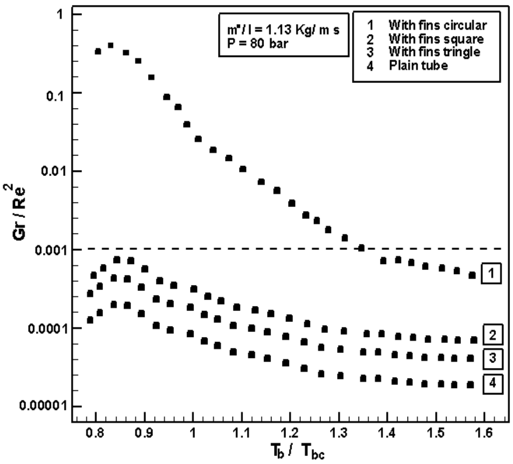

Fig. 9. Relationship between Gr/Re^2 and T_b/T_{bc}.

Figure 10 presents the comparison of heat flux influence with the Nusselt number, and the results appeared that the Nusselt number at different fins decreased under heat flux values of 4.5×10^4 and 1×10^5 W/m^2 with remaining the other conditions due to the importance of heat flux influence on the rate of heat transfer and the profiles of different temperatures and velocities. The variation in flux of heat is compatible with variation in the wall mean temperature and bulk mean.

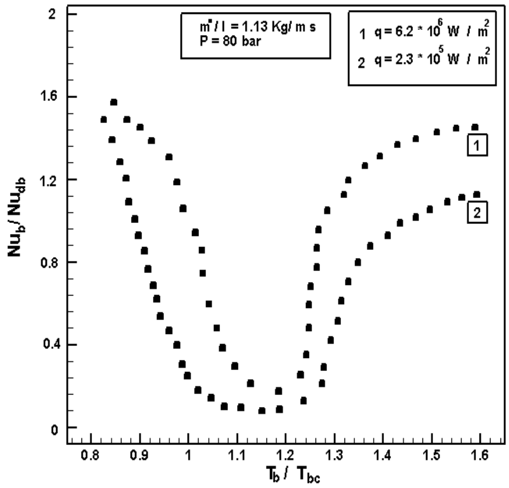

Fig. 10. heat flux influence on Nusselt number.

According to Fig. 11, it can see that the value of mean Nusselt number for tube with fins compared without fins tube was higher. Moreover, the rate of average Nusselt number at circular fins was high with increasing of the Reynolds number. The reason behind that is the heat transfer area has increased due to fins in addition to the reversed flow worked improves the convection of heat transfer. The holes make the flow pass through it and influences on wall tube at high velocities in order to increase the wall cool as a result or it can say, the fins work on recirculate the flow in order to increase the heat exchanging for boundary layer, and it is increased with increasing of the fins numbers which consider one of the best ways for flow mixing. The form of correlations derivative for Nusselt number with fins is:

$$\mathrm{Nu}_b = 1.422\ (L/D)^{0.025}\mathrm{Re}^{0.831} \tag{12}$$

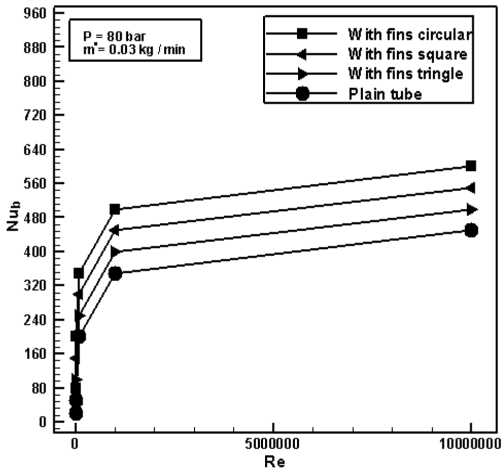

Fig. 11. Reynolds and Nusselt numbers for different fins shapes.

On the other hand, Fig. 12 shows that when Reynolds number increasing, the factor of friction decreasing. Also, the friction factor comparison between the tubes with and without fins indicates that the existing of fins increases the friction factor. Moreover, the space between fins leads to increase the it when it small due to friction factor for fins is higher than the plain tube and the smaller space length between the fins leads to dynamic pressure fluid dissipation. The form of correlations derivative for friction factor with fins is:

$$C_{fb} = 12(L/D)^{-0.024}\ \mathrm{Re}^{-0.328} \tag{13}$$

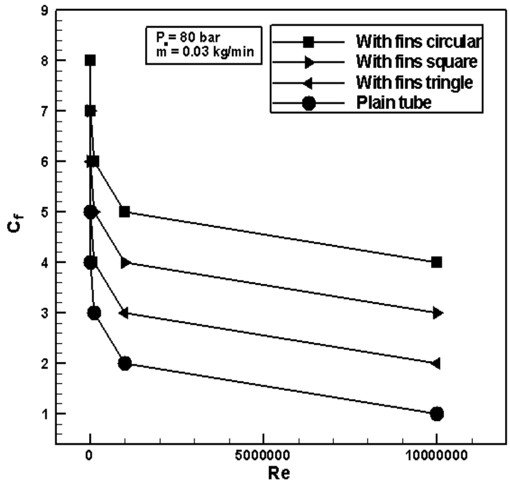

Fig. 12. Reynolds number and Friction factor for different fins shapes.

As shown in Fig. 13, the ratio of performance for tube with circular fins improved and increased at low Reynolds number. It can say at high Reynolds number that the turbulence flows with fins not useful to save the energy. The form of correlations derivative for improvement efficiency with fins is:

$$E_e = 7(L/D)^{-0.022}\ \mathrm{Re}^{-0.468} \tag{14}$$

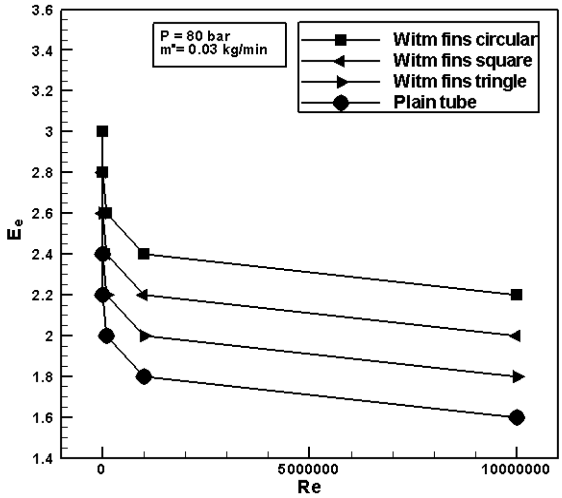

Fig. 13. Reynolds number and improvement efficiency for different fins shapes.

5. Conclusions

This work investigated of convection heat transfer in horizontal tube which mixed with CO_2 at super-critical pressures media experimentally. The main conclusions can be summarized as; firstly, a new experimental investigation has been performed for mini-tube with inserting of circular fin in in order to enhance the rate of heat transfer with supercritical carbon dioxide media. Secondly, the best fins shape was the circular when it inserted in mini-tube and worked on enhanced the heat transfer area in addition to reverse the flow. As a result, transfer of convention heat has been improved. Thirdly, at 80 bar and 0.03 of pressure and rate of mass flow respectively, the finest transfer of heat has been recorded. One of the ways can decrease the heat transfer is decreasing the turbulence because of buoyancy. Fourthly, at $0.8 < T_b / T_{bc} < 1.6$, the values of Nusselt numbers are depending on fins form for supercritical CO_2. Finally, using of circular fins presented new correlations for each of performance ration, factor of friction and Nusselt numbers that helped to provide the real evaluation of proposed method.

Nomenclatures

A	Area surface of the test section, m^2
C_f	Factor of friction
E_e	Enhancement efficiency
g	Acceleration of gravity, m/s^2
h	Coefficient of average heat transfer ,W/m^{2}K

k	Thermal conductivity ,W/m.K
l	Test section heated length, mm
m^*	The rate of mass flow, kg/s
Q^*	Transfer rate heat, W
Re_b	Reynolds number
T^{**}	Logarithmic temperature, ºC
u	Velocity flow of CO_2, m/s
x_in, x_out	CO_2 enthalpy for the inlet and outlet, kJ
Greek Symbols	
Δp	Pressure drop difference of CO_2, bar
μ	Dynamic viscosity, kg/m s
v	Kinematic viscosity of CO_2, m^2/s
ρ	Density of CO_2, kg/m^3
Subscripts	
b	Bulk
i ,j	Ratio of the fitted fins tube
in, out	Inlet and outlet CO_2 of the test section
w	Wall
v	Kinematic viscosity of CO_2, m^2/s

References

1. Zakharov, A.; Jaddoa, A.A.; Timur, B.; and Gumerov, F.M. (2014). Synthesis of the palladium catalyst with the supercritical CO_2-Impregnation method realized in the static mode. *International Journal of Analytical Mass Spectrometry and Chromatography*, 2(4),113-122.
2. Jaddoa, A.A.; Timur, B.; Gumerov, F.M.; Gabitov F.R.; and Neindre B.I. (2015). Regeneration of nickel-molybdenum catalysts DN-3531 and criterion 514 used in Kerosene and gas oil hydrotreating by supercritical Carbon Dioxide extraction. *International Journal of Analytical Mass Spectrometry and Chromatography*, 3(3), 37-46.
3. Jaddoa, A.A.; Bilalov T.R.; Gumerov, F.M.; Gabitov, F.R.; Zaripov, Z.I.; Yarullin, R.S.; Pimerzin, A.A.; and Nikulshin, P.A. (2017). Supercritical fluid CO_2-extraction regeneration of nickel–molybdenum catalyst for hydrotreatment. *Catalysis in Industry*, 9(1), 31-38.
4. Jaddoa, A.A.; Zakharov, A.A.; Bilalov, T.R.; Nakipov, R.R.; Gabitov, I.R.; Zaripov, Z.I.; and Gumerov F.M. (2016). Some thermodynamic processes of anthracite–carbon dioxide mixture in supercritical fluid state. *Russian Journal of Physical Chemistry B*, 10(8), 1180-1190.
5. Bilalov T.R.; Zakharov A.A.; Jaddoa,A.A.; Gumerov F.M.; and Neindre, B. I. (2017). Treatment of different types of cotton fabrics by ammonium palmitate in a supercritical CO_2 environment. *Journal of Supercritical Fluids*, 130, 47-55.
6. Liao, S.M.; and Zhao, T.S. (2002). An experimental investigation of convection heat transfer to supercritical carbon dioxide in miniature tubes. *International Journal of Heat and Mass Transfer*, 45(25), 5025-5034.
7. Jiang; P.X.; Xu, Y.J.; Lv, J.; Shi R.F.; He, S.; and Jackson, J.D. (2004). Experimental investigation of convection heat transfer of CO_2 at super-critical

pressures in vertical mini-tubes and in porous media. *Applied Thermal Engineering*, 24(8-9), 1255-1270.

8. Kruizenga, A.; Li, H.; Anderson, M.; and Corradini, M. (2012). Supercritical carbon dioxide heat transfer in horizontal semicircular channels. *Journal of Heat Transfer*,134(8): 081802.
9. Huai, X.; and Koyama, S. (2007). Heat transfer characteristics of supercritical CO_2 flow in small-channeled structures. *Experimental Heat Transfer*, 20(1), 19-33.
10. Song, J.H.; Kim, H.Y.; and Bae, Y.Y. (2008). Heat transfer characteristics of a supercritical fluid flow in a vertical pipe. *Journal of Supercritical Fluids*, 44(2), 164-171.
11. Cao, X.L.; Rao, Z.H.; and Liao, S.M. (2011). Laminar convective heat transfer of supercritical CO_2 in horizontal miniature circular and triangular tubes. *Applied Thermal Engineering*, 31(14-15), 2374-2384.
12. Shenghui, L.; Huang, Y.; Liu, G.; Wang, J.; and Leung L.K.H. (2017). Improvement of buoyancy and acceleration parameters for forced and mixed convective heat transfer to supercritical fluids flowing in vertical tubes. *International Journal of Heat and Mass Transfer*, 106, 1144-1156.
13. Jiang; P.X.; Shi, R.F.; Zhao, C.R.; and Xu Y.J. (2008). Experimental and numerical study of convection heat transfer of CO_2 at supercritical pressures in vertical porous tubes. *International Journal of Heat and Mass Transfer*, 51(25-26),6283-6293.
14. Cabeza, L.F.; De Gracia, A.; Fernandez, A.I.; and Farid, M.M. (2017). Supercritical CO_2 as heat transfer fluid: A review. *Applied Thermal Engineering*.125, 799-810.
15. Hoseinzadeh, S.; Sahebi, S.A.R.; Ghasemiasl, V.; and Majidian, A. (2017). Experimental analysis to improving thermosyphon (TPCT) thermal efficiency using nanoparticles/based fluids water. *European Physical Journal Plus*, 132(5), 197.
16. Hoseinzadeh, S.; and Heyns, P.S. (2020). Thermo-structural fatigue and lifetime analysis of a heat exchanger as a feedwater heater in power plant. *Engineering Failure Analysis*, 113, 104548.
17. Adams, T.M.; Abdel-Khalik, S.I.; Jeter, S.M.; and Qureshi, Z.H. (1998). An experimental investigation of single-phase forced convection in microchannels. *Applied Thermal Engineering*, 41(6-70), 851-857.
18. Yakut, K.; Sahin, B.; and Canbazoglu, S. (2004). Performance and flow-induced vibration characteristics for conical-ring turbulators. *Applied Energy*, 79(1), 65-76.
19. Kakac, S.; Shah, R.K.; and Aung, W. (1987). *Handbook of single-phase convective heat transfer*. USA: John Wiley and Sons.