

Experimental Investigation of Supercritical Fluid Heat Transfer Properties in a Miniature Heat Sink

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ABSTRACT

Currently, efficient heat transmission for compact electronic elements is an essential matter. The objective of this project is to build a test rig to study the heat transfer effect of SCF- CO_2 on the miniature heat sink, which is used in cooling in much small electronic equipment. It needs a heat sink with a liquid cooling scheme that meets these demands as much as feasible. The dimensions of 20 mm x 20 mm and a width of 3 mm were adopted for features of heat transfer as well as the fluid flow of supercritical CO_2 in heat sink copper 360 alloys in this study. A heat sink has 20 homogeneous arrays of fins and 19 tubes, each having a rectangular cross-section and a 1mm hydraulic diameter. Moreover, the adopted pressures, temperatures, and mass velocity ranges were 7.5 to 12 MPa, 35 to 50 °C, and 100 to 500 $\text{Kg m}^{-2} \text{s}^{-1}$, respectively, wherein the CO_2 cooled under these conditions. The results of the experiment found that there is a substantial impact on the properties of liquid inflow and temperature motion by the system pressure, the temperature of CO_2 as well as cluster velocity. Additionally, as the medium temperature of CO_2 in the adjacent significant point area increased, the pressure decreased and the medium temperature movement factor augmented dramatically. It was also noted that the medium temperature movement factor peaked at the pseudo-critical temperature. Using the obtained data, a novel correlation mechanism for limited convection of super-critical CO_2 in regular multi-port micro tubes based on chilling conditions was constructed using the obtained coefficients in this study.

Keywords: Heat Transfer; Supercritical Fluid CO_2 ; Mini Heat Sink

Introduction

The requirement for liquid cooling in much small electronic equipment and other electrical devices has become a key source of concern in recent years in terms of modernization and technical growth. Electronic circuits generate heat that must be dispersed outside of the appliances, thus different cooling systems are required for these devices to function accurately and efficiently. Heat sink cooling systems deems an interesting subject of research in recent years. In this vein, single-phase forced convection in a heat sink that is highly super-cooled is a powerful chilling method with a wide variety of utilization.

Among these applications are magnets high voltage systems, pressurized nuclear fission systems, spacecraft applications such as thermal administration systems, and electronic device industrialization [1]. In the literature, experiments were conducted for the following cases: aligned channels, perpendicular channels, perpendicular packed bed channels, perpendicular annulus, tube-in-tube, and perpendicular normal circulation tube forms, micro-porous media along with one more structure by researchers [2]–[5]. On the other hand, temperature movement was greatly improved in the adjacent region of the critical area, with the greatest temperature movement parameters happening at the analogous pseudo-critical temperature. Such improvement was achieved according to an experimental study of supercritical carbon dioxide flow and heat transfer in multichannel mini ports based on cooling conditions.

It is interesting to note that the temperature movement of liquid acting beneath or nearby supercritical circumstances for different material properties has received much more attention in the literature [7]. Petukhov [8], Hall [9], and Polyakov [10] introduced extensive assessments. In the near-critical zone, Ghajar [11] examined the available empirical methodologies for constrained convective temperature movement. The author discovered that Dittus-Boelter-type correlations could be used to estimate tumultuous constrained convective temperature movement in the adjacent-critical region, whereas the property ratio technique was suggested to estimate substantial differences in physical characteristics. Thereafter, Pitla et al. [12] created a new correlation based on experimental data and numerical calculations to prophesy the temperature movement of supercritical CO₂. The majority of these recent studies made use of considerable or standard channels. Heat transfer measurements were carried out by Liao and Zhao [13] for supercritical CO₂ influx in flat mini/micro circular channels chilled by a stable temperature liquid. The authors discovered a significant disagreement among their outcomes and results that existed in the prior research, which they attributed to the buoyancy impact occurring in regular channels of great tubes dimensions.

In terms of relevant dimensionless parameters, the researchers provided experimental relationships for the axially averaged Nusselt number.

Acosta et al. [16] looked at momentum and mass transport in tight rectangular tubes using diameters of 0.96 and 0.38 mm, respectively, of hydraulic tubes. The authors investigated both laminated flux and turbulence situations. Correlations with greater tubes were stated to be satisfactory, and the heat and mass transfer similarity led to the conclusion that heat transfer correlations should be acceptable as well. However, it was discovered that the walls made of optical materials were required to meet the hydraulic smoothness status.

Furthermore, the surface roughness had a significant impact on both the friction coefficient and the mass movement factor. For mini/microtubes or channels with cooling conditions, the convection heat transfer of fluids at supercritical pressures was also explored. An investigation was conducted by Pettersen [17] to examine the heat transfer average factor of supercritical CO₂ in mini-channels based on a diameter of inner 0.79 mm for chilling conditions. The author discovered that Gnielinski's relationship for the tow-phase temperature movement factor corresponds to experiment data adequately. In regular mini/micro rotational pipes of 0.50, 0.70, 1.1, 1.40, 1.55, and 2.16 mm diameter chilled to a stable temperature, Liao and Zhao [18] studied axially-averaged convection temperature movement to supercritical CO₂. In this vein, the researchers discovered that the buoyancy impact for constrained heat transfer of supercritical CO₂ over regular pipes at maximum Reynolds numbers was still significant, and they improved the relationship for the Nusselt number constrained heat transfer of supercritical CO₂ in regular microtubes. In [19], the researchers studied the influence of mass influx, pressure, and temperature influx on the temperature movement factor and reduction pressure of carbon dioxide at critical compressing based on chilling circumstances for regular chilling pipes with dimensions changing from 1 to 6 mm of diameters and suggests a new Gnielinski formula by suitably choosing the standard heat level. It was noted that the majority of current work on supercritical CO₂ concentrated on averaged properties, with a relatively little study on local temperature movement and compression loss, especially for narrow pipes.

Many relationships, notably Dittus-Boelter kind, were created and utilized for constrained heat transfer, and the property proportion and standard heat approach were used to integrate varying-properties impacts. However, because the events are so complex, it is widely accepted that the relations do not demonstrate enough concurrence with trials and should only be employed in certain circumstances. This temperature oscillation causes additional mechanical stress to the soldering and welding joints inside a semiconductor module and may cause a thermal shock. This leads to an even high-temperature oscillation causing thermal shock and fatigue, finally, the device fails. Hence, the junction temperature is the limiting condition, and the maximum junction temperature is not excessive [20]–[23]. This paper describes an experimental examination of supercritical CO₂ heat transfer and

flows features in minor tubes under chilling for a couple of local and averaged through the entire region of the test. A comparison was conducted among the results available in the literature with the experimental results. Consequently, an empirical correlation was developed. The impacts of various factors on heat transfer and pressure drop behavior were also investigated.

Experimental Set-up

Photographically and schematically, Figures 1 and 2 depict the experimental rig, respectively. This study used an experimental setup with integrated measurement devices to investigate the temperature movement behavior of carbon dioxide in a minor heat sink with forwarding fluid flow. The many challenges related to the high-pressure trials need a considerable interval of about 365 days to settle during the test rig's commissioning. Initial exams for system parameters standardization and fault expectation removed leakage and fluctuation operating issues. In every single running process, the test data collection technique was meticulously replicated. The entrance and outlet heat, the entrance pressure, the pressure decline over the experiment parts, the inflow velocity, the voltages across the heater, the current, and the values of impedance were all measured throughout the experiments.

Also, fifteen copper-constantan thermocouples were used to measure the temperature of the nearby tube. The thermocouples were welded to the heat sinks outside surface. The thermocouples were put into mini-channels (0.2 mm deep and wide) cut into the test section surface throughout the tube. Pre and post the test area, mixers were installed to mix the fluid prior to accurate thermal resistors which were used to measure the inlet and output fluid temperatures. The thermocouples and resistors thermal were gauged, utilizing a steady temperature oil bath, prior to installation. The temperature readings were accurate to within 0.1 degrees Celsius. The entrance pressure was computed with a precision of 0.075 present for the entire interval of 12 MPa using a pressure gauge transducer (Model EJA430A). A differential pressure transducer (Model EJA110A) with a precision of 0.075 present in the whole range of 500 kPa was used to measure the pressure decrease in the test section. A Coriolis-type mass flow meter (Model MASS2100/MASS6000, MASSFLO, Danfoss) was used to measure the mass flow rate. The mass flow meter's nominal range was 0–65 kg/h, with 0.1 present accuracies. A vacuum pump was used to empty the test loop before the trials. The test loop was then cleaned three to four times with powerful clear carbon dioxide (99.995 present). Repeatedly, the pipe was emptied and scrubbed with carbon dioxide.

From the compressed CO₂ container, high-pressure carbon dioxide with a clear of 99.995 present was delivered into the system at constant

pressure with 5 MPa. The system pressure was raised to the appropriate super-critical pressure using pressure at maximum level with the carbon dioxide pump. The high-pressure carbon dioxide pump was then switched off, and the sluice connecting the system under investigation to the high-pressure carbon dioxide pump was shut down. The super-critical magnetic pump circulated CO₂ at supercritical pressure. Several experiments revealed that even at considerable pressures, the leakage was relatively modest e.g. 10 MPa. Silicate glass fiber and sponges were used to insulate the system under investigation and the majority of the loop's system. The mass flow rate, input power, and inlet fluid temperature were all kept constant for each test. A data extract device (HP 34970A) and an individual computer were used to connect all the measuring instruments. Post the steady-state conditions were established; heat, mass inflow rate, intake pressure, and pressure decline were observed and documented. In addition, the current-voltage values over the heater were measured. The input and output heat, as well as the inflow rate and input voltage, were used to determine the local bulk medium liquid heat at each testing step. The rise in fluid enthalpy was compared to the electric power input. The heat balancing experiment had a 5% experimental uncertainty. In the studies, the system took a long time to attain a steady-state (50–120 minutes). Temperature, input pressure, and inflow rate fluctuations were continually documented throughout the initial transients. During at least 10 minutes, the system was determined to be in a steady state when the changes in the wall temperatures, inlet and outlet fluid temperatures, and inflow rate and input pressure were set at about 0.1°C. In addition, the flow rate and inlet pressure were all within 0.2 percent. Experimental error in the temperature stability, axial thermal conduction in the system under investigation, erroneously calculated temperature, and the calculation of the temperature movement surface were the main causes of test doubt in the convection heat transfer coefficients. For the small heat sink, the root-mean-square experimental doubt of the convective temperature movement factor was expected to be 11.3 percent. The inlet pressure experimental errors were calculated to be 0.09 percent.

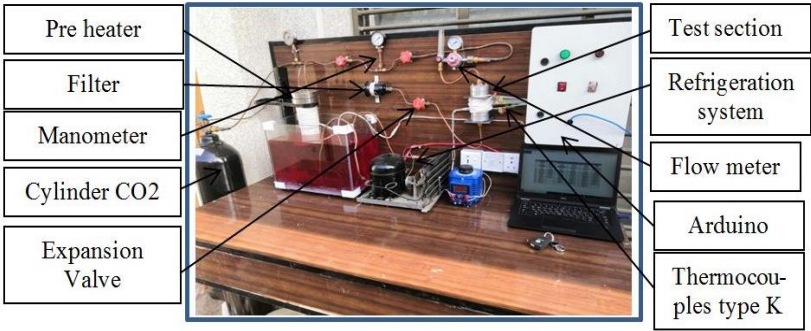


Figure 1: Supercritical carbon dioxide experiment set-up

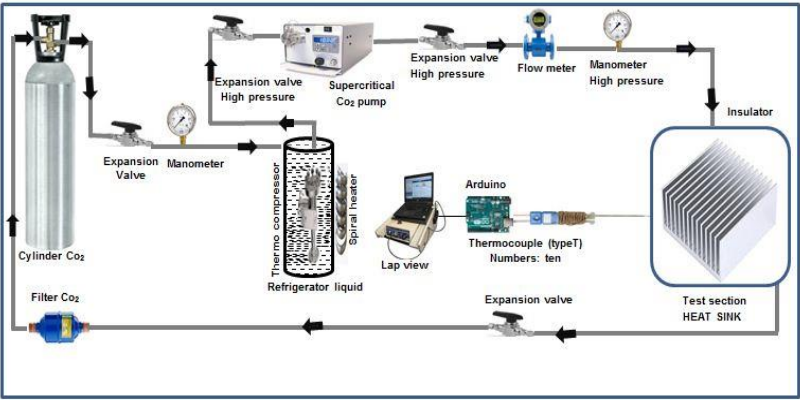


Figure 2: Schematic of a test rig in action

Test section (heat sink)

The heat sink in Figure 3(a) was made of copper 360 alloys with dimensions of 20 mm × 20 mm and a thickness of 3 mm. A heat sink is made up of 20 fins and 19 channels, each with a rectangular cross-section and a hydraulic diameter of 1mm. Figure 3(b) depicts the channel width, height, and thickness, as well as all other measurements.

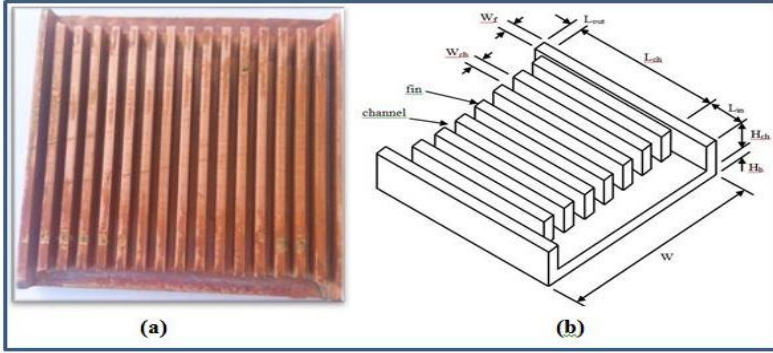


Figure3: (a) Configuration of heat sink, (b) Heat flux cooling application, a schematic depiction of the heat sink geometry is shown

When the mini heat sink was heated instantly with a low voltage AC current, the temperature transfer in the pipe can be presumed to be single dimension material with an implicit heat generation. The outer surface was isolated, and thermocouples were used to measure the mini heat sink surface temperatures, $T_{wo}(x)$. $T_{wi}(x)$, the surface's regional temperatures, was computed as:

$$T_{wi} = T_{wo} + \left[\left(\frac{q}{16k} \right) (l_o - w_i)^2 \right] + \ln \left(\frac{l_o}{w_i} \right) \left[\left(\frac{q}{8k} d_i^2 \right) \right] \quad (1)$$

The pipes were winded by the cable to heat the small heat sink. Utilizing the calculated heat value, $T_w(x)$, of the plat, temperatures of the temperature movement surface were estimated.

$$T_{wi}(x) = T_{w\psi}(x) - \ln \frac{l+2\psi}{l} \frac{qw}{2ks} \quad (2)$$

At each axial point, the domestic temperature movement factor, h_x , and the Nusselt value, Nu_x , were determined as follows:

$$h_x = \frac{q}{(T_{wi} - T_f)}(x), Nu_x = \frac{h \times l}{K_{f,b}}(x) \quad (3)$$

Using the local bulk liquid formation enthalpy, $h_{f,b}(x)$, the domestic bulk liquid heat, $T_{f,b}(x)$, was estimated.

$$h_{f,b} = h_{f,o} + \frac{\pi q_x l x}{m} \quad (4)$$

In the small heat sink, the mean temperature movement factor, h , and the mean Nusselt value, Nu_m , were computed as:

$$h = \frac{q}{T_{wi} - T_f}, Nu = \frac{hl}{K_f} \quad (5)$$

The average of entire domestic wall heat was used to calculate the mean temperature of the heat transfer surface.

$$T_m = \sum_{i=1}^n (T_{wi} \Delta x) / L \quad (6)$$

The average of the measured and output liquid heat was used to get the mean liquid temperature, $T_{f,m}$. The Reynolds number is calculated as follows:

$$Re = \frac{4m}{l\mu\pi} \quad (7)$$

Result and Discussion

By regularly altering the pressure, heat flux of the surface, the interior temperature, and the CO₂ mass quickness, a sequence of experiments were carried out. Figure 4 shows the pressure drop from the test section's intake to exit CO₂ average temperature at a particular pressure of 9.0 MPa beside different accelerations ranges of 100 - 500 kg/m²s, respectively. It's worth noting that the pressure reduction in Figure 4 was measured in kPa, which is substantially lower compared with the total working pressure of the tests, which is 8.0 MPa. Additionally, in the case of the average temperature of carbon dioxide being lower/greater than the pseudo-critical heat, the pressure reduction augmented progressively with augmenting the average temperature of carbon dioxide. In the same context, when the average heat of CO₂ is near the pseudo-critical temperature, the pressure reduction increases dramatically owing to the extreme change in its material characteristics. It might be conclusively proven that mass acceleration has a meaningful impact on pressure reduction. In a brief statement, the greater the pressure reduction, the greater the mass velocity.

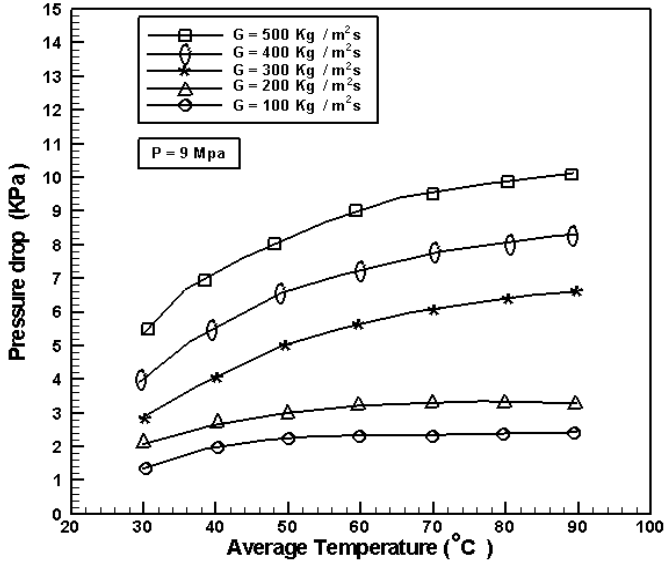


Figure 4: CO₂ average temperature vs. pressure decrease at various mass velocities

Figure 5 displays the performance for various running pressures range 7.5 - 12 MPa and mass velocities of 100 - 600 kg/m² s. Through this study, it can clearly be seen that during the interval of the average heat of carbon dioxide was lower or equal to the pseudo-critical heat ($T_{pc} = 32$ °C at $P = 7.5$ MPa and $T_{pc} = 36$ °C at $P = 12$ MPa), see Figure 6, the pressure decline continuously augmented as the average heat of carbon dioxide was increased. Owing to the substantial difference in CO₂'s physical properties, the pressure decline augmented drastically as the average heat increased near the pseudo-critical point. The mass velocity, as expected, has a major impact on the pressure drop. In addition, at constant operating pressure, the pressure reduction increases with mass velocity; that is to say the bigger the pressure drop, the higher the mass acceleration. In this respect, because the difference in physical characteristics grows smaller as the running pressure increases, the pressure drop reduces for a fixed mass velocity.

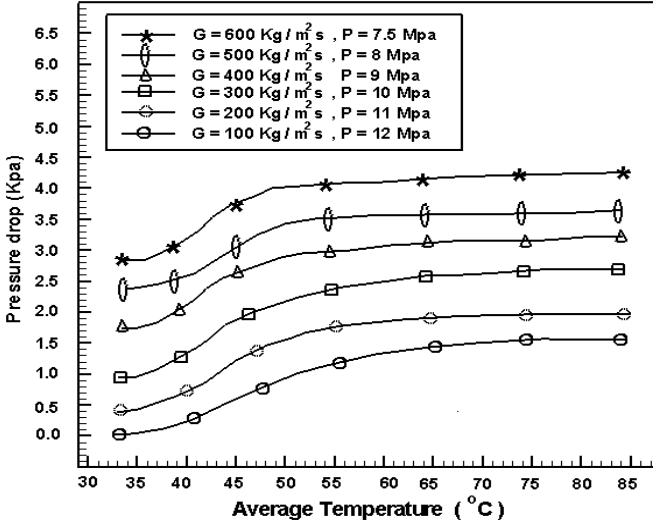


Figure 5: CO₂ average temperature vs. pressure drop for various mass velocities and pressures

For system parameters $P = 9$ MPa, $G = 288 \text{ kg m}^{-2}\text{s}^{-1}$, and $T_{\text{in}} = 33$ °C, Figure 6 shows standard alteration in the domestic pressure P , the pseudo-critical heat T_{pc} , the domestic bulk mean heat of carbon dioxide T_{bm} , and the domestic interior wall heat T_{wi} over the experiment system under investigation, besides the calculated input and output heat of carbon dioxide T_{bm} . Because the pressure decrease over the experiment segment is so modest, the related pseudo-critical heat remains nearly constant. In the current cooling condition, the heat value was dropped over the inflow direction of carbon dioxide. The estimated carbon dioxide temperatures at the input and outflow were matched the test data quite well.

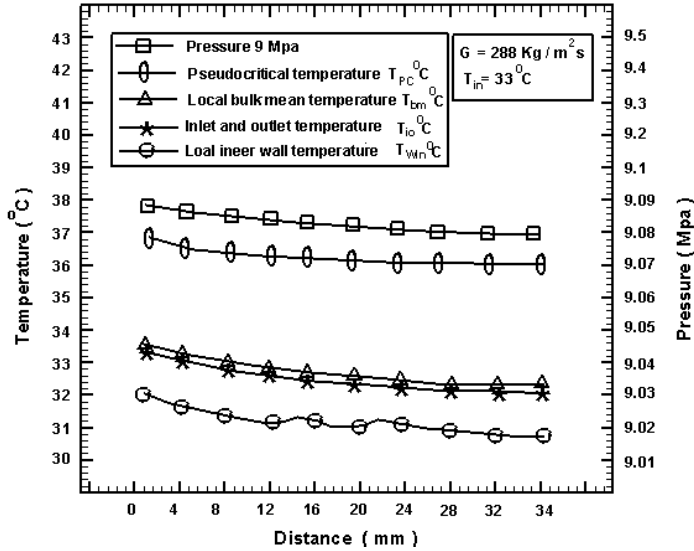


Figure 6: Performance of pressure and temperature against distance

Along the test segment, Figure 7 depicts the related fluctuations in the local temperature influx and the local temperature movement factor. Both the inner wall temperature and the temperature movement factor show some oscillations. A consequence of this is that owing to a lack of ideal correlation between the cooled copper blocks and the exam sample through the testing. However, the measured parameters remain indicating evidence of the general tendency of fluctuation in the temperature flow and temperature movement factor: the temperature influx and temperature movement factor dropped as the temperature of CO_2 approached the pseudo-critical temperature along the test segment.

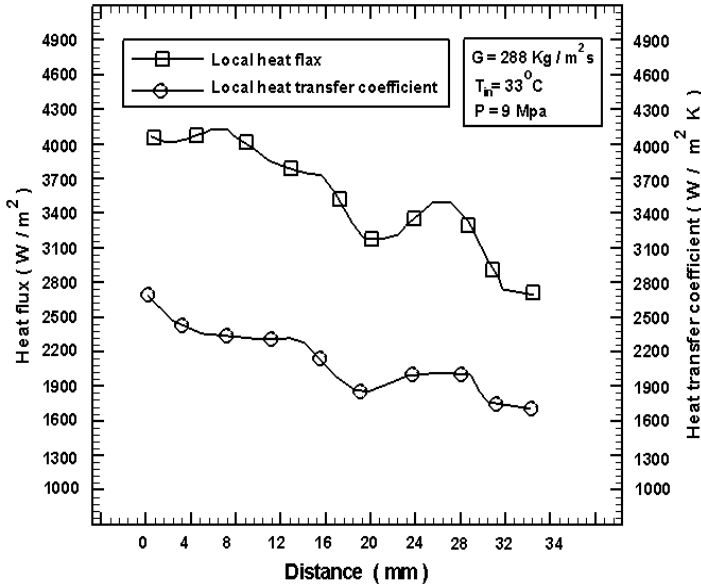


Figure7: Performance of heat influx and temperature movement factor against distance

At varying inlet temperatures of CO₂ T_{in} from 35 to 50 °C, Figures 8 and 9 show the fluctuation of domestic pseudo-critical heat T_{pc} , domestic bulk average heat of carbon dioxide T_{bm} , and domestic temperature factor α over the exam segment. The relevant working conditions were adopted in this work as $G = 210 \text{ kg/m}^2 \text{ s}$ and $P = 9.5 \text{ MPa}$. The situation with $T_{in} = 48 \text{ °C}$, which is the nearest to the pseudo-critical heat of carbon dioxide, 39 °C, yields the highest temperature movement factor, as seen in Figures 8 and 9. Considering the particular temperature achieves a maximum magnitude at the pseudo-critical level, a maximum temperature movement factor is obtained as the heat of carbon dioxide is near to the pseudo-critical heat [5, 6]. This finding shows that CO₂ temperature has a considerable impact on temperature movement around the pivotal point and that the maximum temperature movement average appears in this zone.

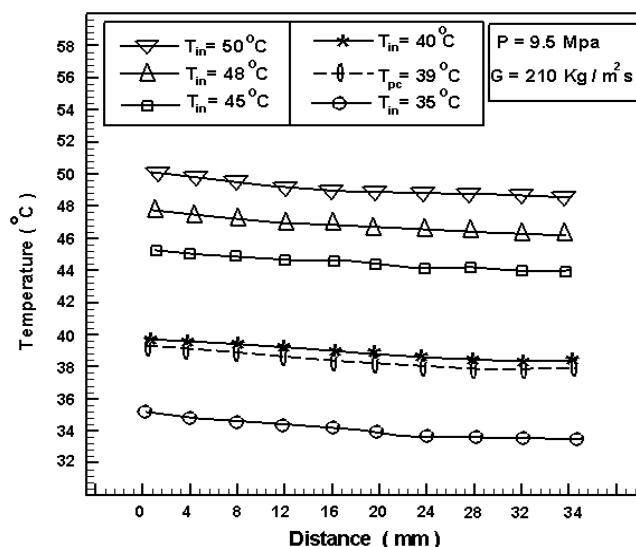


Figure 8: Local temperature fluctuations along the test segment

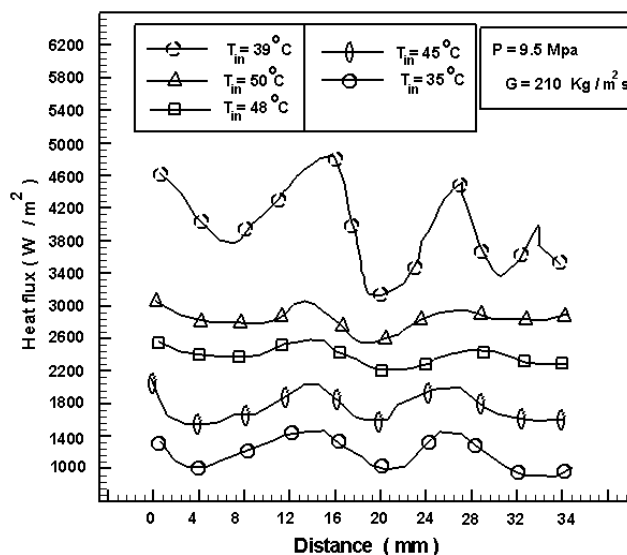


Figure 9: Differences in domestic temperature movement factor accompanying the rig-test segment

The significant shift of material characteristics for instance density, specified temperature, viscosity, and electric accessibility is among the most essential aspects of supercritical liquids in the close-critical zone. Figure 10 depicts typical CO₂-specific heat fluctuations in terms of heat based on different pressures (7.5, 8.0,10,11, and 12 MPa). At the close of the pseudo-critical heat zone, which matches the maxim of specified temperature at the working pressure, a drastic change may easily be seen [1-2]. CO₂ temperature movement and inflow properties may vary significantly compared to steady characteristic liquids as a result of these changes in fluid property.

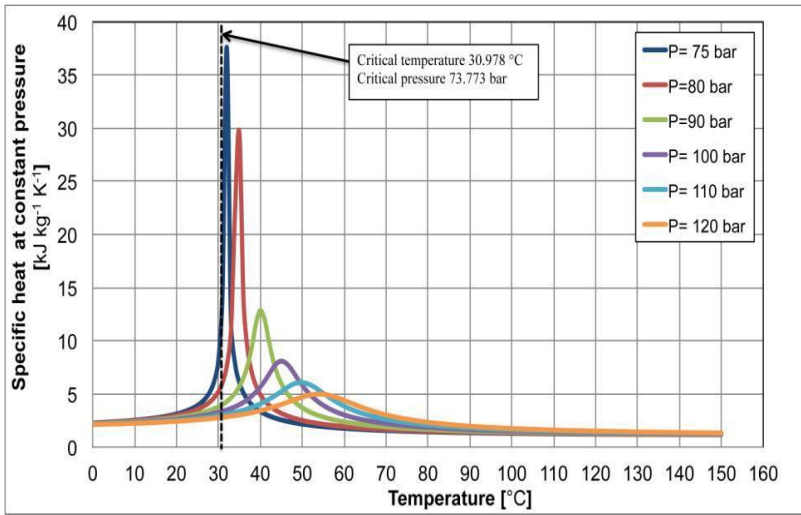


Figure10: Heat specific of carbon dioxide at pressure (7.5, 8.0 ,9,10,11, 12 MPa)

Figure 11 shows how the rate temperature movement factor varies with CO₂ temperature rate for a specific system pressure of $P = 9.5$ MPa and mass velocities range $100 - 500 \text{ kg m}^{-2}\text{s}^{-1}$. At each mass velocity, the rate of temperature movement factor augments rapidly in the close-critical zone, peaking around the identical pseudo-critical heat $T_{pc} = 39$ °C at $P = 9.5$ MPa. As illustrated in Figure 10, the exceptionally increase in temperature movement rate nearby the pseudo-critical heat zone is because specific heat varies similarly throughout the region near the critical point. Furthermore, the higher temperature movement factor grows individually with mass acceleration for the given operating pressure of 9.5 MPa, implying that mass acceleration has a considerable impact on temperature movement performance [3-4].

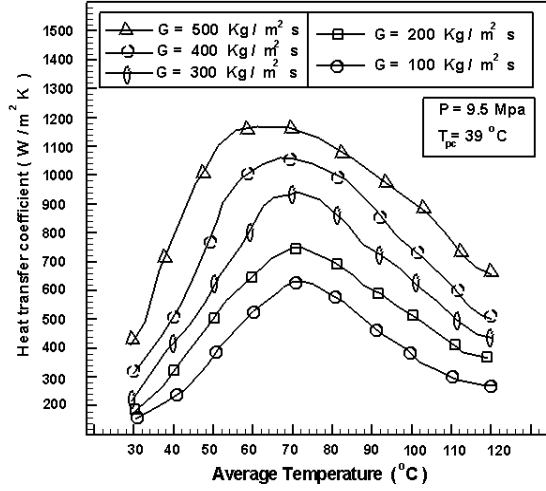


Figure 11: CO₂ average temperature vs. heat transfer coefficient average

Figure 12 illustrates the impact of the adopted system pressure on the temperature movement factor rate; the adopted parameters of this work were pressure 7.5 - 12 MPa, and mass velocities 250 kg/m²s. Figure 12 demonstrates a comparable difference in the temperature movement rate in degree with the heat rate of carbon dioxide presented in Figure 11. That is to say, a direct relationship between increasing the CO₂ temperature and the average heat transfer degree. Wherein, it is to reach the maximum level rapidly at the adopted pseudo-critical heat. T_{pc} = 32 °C at Pressure 7.5 MPa, Pseddocrical temperature 34 °C at Pressure = 8.0 MPa, Pseddocrical temperature 36 °C at Pressure 9 MPa, Pseddocrical temperature 38 °C at Pressure 10 MPa, Pseddocrical temperature 40 °C at Pressure 11 MPa, Pseddocrical temperature 42°C at Pressure 12 MPa. According to the same specific heat difference at the precise pressure in which the obtained results in Figure 10 were achieved, the maximum level of heat transfer average factor increases as the pressure decreases at 10-12 MPa.

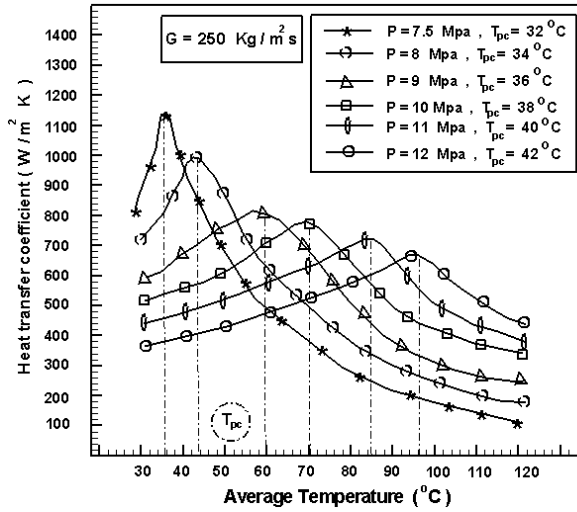


Figure 12: Differences in CO₂ average temperature vs. heat transfer factor average

Dimensional numbers, many relationships between the Nu and Re, and others are suggested in the literature. The results of the current experiment are compared to the 5 most significant correlations suggested by Huai-Koyama, Di-Boelter, Gnielinski, Pe-Popov, Pitla, and Li-Zhao. Figures 13 and 14 show a comparison predicted by 5 correlations in the literature for local Nusselt number along with the test section a with experimental data; Figure 13 for the case of pressure 8 MPa, mass velocities = 350 kg/m²s, and temperature inlet 31 °C, and Figure 14 for the case of Pressure 8 MPa, mass velocities 350 kg/m²s, and temperature inlet 55 °C, respectively. A comparison of the cases experimentally investigated for local Nu over the test segment with those estimated in the literature for 5 relationships. All correlations have indicated a smooth change in the local Nu, but the present experimental data show some fluctuation in the local Nu along the test section. The Pe-Popov correlation, in particular, is much overestimated; since the relationship was formed using data from larger diameter tubes and is therefore unsuitable for the current small diameter tubes. Hu– Koyama, Di-Boelter, Gnielinski, and Pitlas other three correlations all indicate a proper order of magnitude for the local Nu.

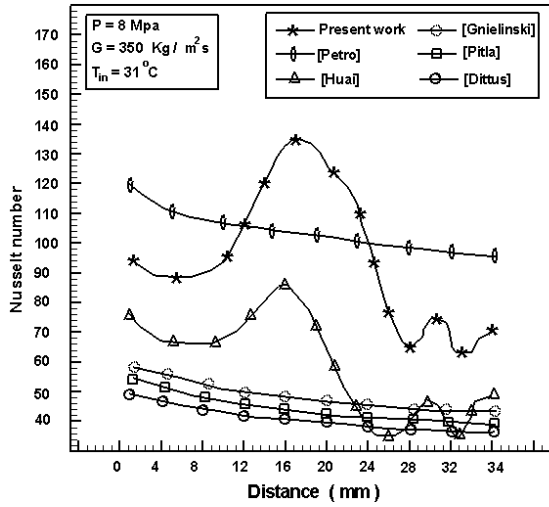


Figure 13: Comparison of experimental data for the case of pressure 8 MPa, mass velocities 350 kg/m²s, and temperature inlet = 31 °C

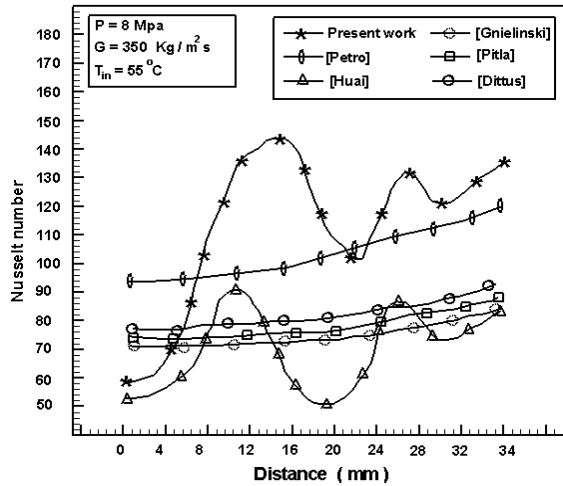


Figure 14: Experimental data comparison for the case of pressure 8 MPa, mass velocities 350 kg/m²s, and temperature inlet 55 °C

All correlations have indicated a smooth change in the local Nusselt number, but the current experimental data show some fluctuation in the local Nusselt number along the test section. The Petrov-Popov correlation, in particular, is much overestimated; since the relationship was formed using

data from larger diameter tubes and is therefore unsuitable for the current small diameter tubes. Huai – Koyama, Dittus-Boelter, Gnielinski, and Pitla's other three correlations all indicate a proper order of magnitude for the local Nu. The average Nu achieved in this study in terms of CO₂ average temperature is compared to estimation utilizing Di-Boelter, Gnielinski, and Liao-hypothesized Zhao's correlations. Figure 15 shows the results under the conditions of Pressure 8.0 MPa and mass velocities of 310 kg/m²s. It can be demonstrated that all 3 correlations estimated the highest Nu close to the relevant pseudocritical temperature of 35 °C at a Pressure of 8 MPa and also provide the local Nu with the right order of magnitude.

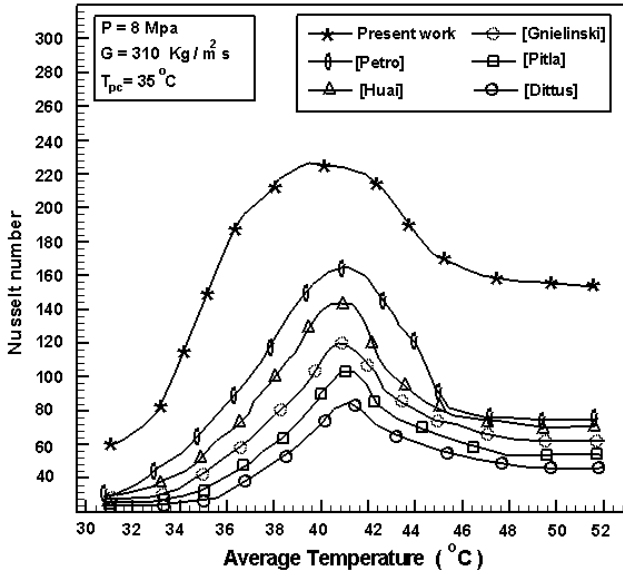


Figure 15: Correlations in the literature with the measured Nu

The next experimental relation for constrained convection temperature movement of SFCO₂ in a regular small heat sink was achieved under a large quantity of experimental data.

$$Nu = 3.34 \times 10^{-1.965} Pr^{0.4} Re^{0.9} \left(\frac{average\ C_p}{C_{pw}} \right)^{0.099} \left(\frac{\rho_r}{\rho_w} \right)^{-2.23} \quad (8)$$

where C_{pw} and ρ_w represent the expected according to wall temperature. ρ_r is denoted CO₂ bulk temperature expected value. Average Cp is determined as $\frac{h_f - h_w}{T_r - T_w}$.

The bulk average heat of carbon dioxide is used to determine all physical parameters in Nu, Re, and Pr. Equation 8 ranges of application are $7 \leq \text{Pressure} \leq 12$, $20 \leq \text{bulk mean temperature} \leq 53$, $100 \leq \text{mass velocities} \leq 600$, and $1 \leq \text{heat flux} \leq 25$. In this scenario as illustrated in Figure 12 pressure 7.5, 8.0, 8.5 MPa, mass velocities 209.3 kg/m²s, Figure 16 illustrates a comparison of the measured Nu value with those computed utilizing the correlation, Equation 8. Clearly, the novel correlation accurately predicts the experimental findings.

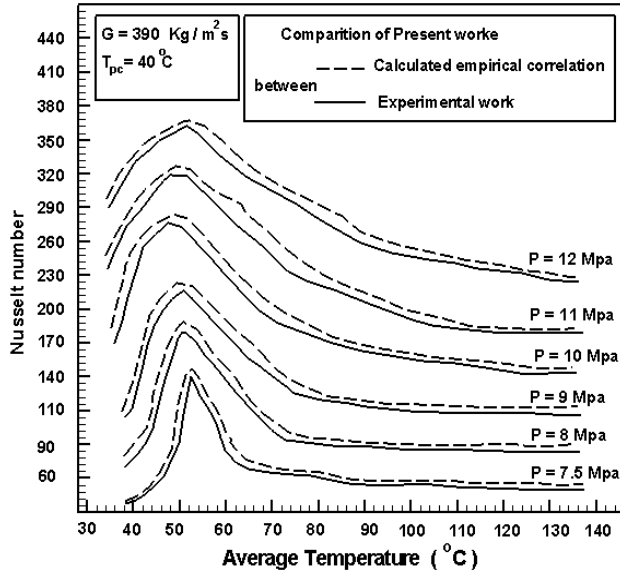


Figure 16: Equation 8 and the experimental Nusselt number are compared

The obtained results by Equation 8 based on Nusselt value were compared with calculated using Dittus-Boelter and Gnielinski correlations. See Figure 17. In this regard, the comparable adopted limitations of this research were Pressure 8.0 MPa and mass velocities = 300 kg/m²s. Nu was predicted by three correlations to be in the appropriate order of magnitude, with a maximum variation of roughly 18 percent.

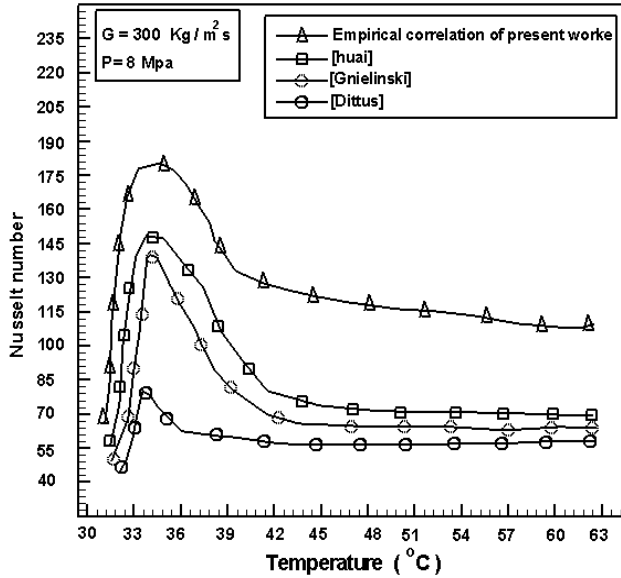


Figure 17: Equation (8) is compared to correlations found in the literature

Conclusions

1. Heat transfer performance is influenced by test section pressure, mass acceleration, and CO₂ heat, notably in the close-critical zone. Meanwhile, as the test section pressure rises, the maximum heat transfer coefficient drops. Also, it was noted that the heat transfer coefficient increases as the mass velocity increases.
2. According to the experimental results, the pressure reduction increases considerably as the average temperature of carbon dioxide in the zone close to the pseudo-critical heat increases.
3. The pressure declined was significantly affected by mass acceleration and adopted pressure. Even more, for a fixed operating pressure, the pressure reduction augmented with mass acceleration, however, for a specific mass acceleration, the pressure reduction declined with adopted pressure. Furthermore, the results of the experiments also showed that CO₂ temperature, operating pressure, and mass velocity all play a vital role in the overall system performance.
4. At the nearby-critical zone, temperature movement is substantially improved, at the matching pseudo-critical heat, with the greatest temperature movement factor.

5. For constrained convective temperature movement of supercritical CO₂ in a regular heat sink in the chilling situation, a Dittus-Boelter correlation category was obtained based on a significant quantity of experimental data besides the novel correlation accurately matched the experimental results.

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Nomenclature

A Area surface of the test section, m²
w_f Width of test section mm
g acceleration of gravity, m/s²
h coefficient of average heat transfer, W/m²K
k thermal conductivity, W/mK
l_f test section heated length, mm
T_{wi} Local temperature along the test section °C
T_{fb} The local bulk fluid temperature °C
T_{pc} Pseddocritical temperature °C
p Pressure of CO₂ Mpa
T_m Mean temperature °C
G Mass velocities kg m⁻²s⁻¹
Nu_x Nusselt number $\frac{\rho u x}{\mu}$
u Velocity flow of CO₂, m/s

Re_b Reynolds number ud/ν
 Δp Pressure drop difference of CO₂, bar

Greek symbols

ν Kinematic viscosity of CO₂, m²/s
 ρ Density of CO₂, kg/m³
 μ dynamic viscosity, kg/ms

Subscripts

w Wall
in, out inlet and outlet CO₂ of the test section
b bulk
f fluid
x axis

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