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Towards defining the optimal design parameters for a test setup studying heat transfer with carbon dioxide at supercritical conditions

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ABSTRACT

As the demand for environmentally friendly refrigeration technologies continues to grow, the use of natural refrigerants for the thermal management of detectors for High Energy Physics becomes more and more relevant. Whereas the use of boiling CO_2 flows is well established at CERN for applications requiring cold conditions, this study focuses on the investigation of the heat transfer capabilities of carbon dioxide in supercritical conditions. In this regard, high thermal capacities and low values of density and viscosity are some of the main characteristics of fluids above their critical point. Furthermore, thanks to their inherent single-phase-like nature, supercritical fluids could allow for relatively simple fluid management in multibranched circuits. These features, combined with its critical temperature of 31 °C, make of supercritical carbon dioxide (sCO_2) an exceptionally well-suited candidate for all thermal management applications where the electronics can be operated at temperatures above 32 °C. However, several points on this subject remain open in the available literature, and this calls for the need for additional accurate experimental observations. This work is centered around the development use of a dedicated test rig designed to explore the thermodynamic performance and efficiency of sCO_2-based systems.

Keywords: Carbon Dioxide, Refrigeration, Supercritical, Heat Transfer, Pressure Drop

1 INTRODUCTION

Investigation of heat transfer at supercritical conditions dates back to the 1930s with the study of free-convection heat transfer to fluids at near-critical point (Pioro, 2019). In the 1950s, the use of water at supercritical conditions (SCW) seemed like an attractive application for the Rankine cycle to increase the thermal efficiency of coal-fired thermal power plants. At the end of the 1950s and beginning of the 1960s, early studies were carried out to evaluate the use of SCW in nuclear reactors, and several were developed in many countries. Since the year 2000, there has been a renewed interest in developing SCWR (SuperCritical Water Reactor) to improve the coming generation of nuclear systems (Wang et al., 2018), which are expected to be implemented in the coming years (Duffey and Pioro, 2005).

Currently, the most widely used supercritical fluids (SCW) are water and carbon dioxide. The latest general phenomenological review on supercritical carbon dioxide cooling dates to 2019, while another review published in 2020 by Xie et al. (Xie et al., 2020) provides a detailed survey of the topic of heat transfer deterioration in sCO_2 in vertical flows. In the latter, authors remark the limited number of investigations at tube sizes below 2 mm. Research on sCO_2 has been increasing significantly, as indicated by a growing number of publications. A ScienceDirect search with keywords "supercritical," "carbon dioxide," and "heat transfer coefficient" shows over 11,000 results by February 2024, with more than 4,000 articles published since 2019. This trend underscores the growing interest in sCO_2 technology for various applications. Despite this increasing interest, there is a critical need for reliable data, especially for small diameters, to develop accurate models and optimal designs for precise control or efficiency in systems operating with sCO_2. This article is led towards finding the optimal design of a test setup to study heat transfer of sCO_2, which can have the applications listed below.

1. **Thermal-power cycles**, where the two main cycles are the supercritical and trans-critical cycles. Power cycles based on super-critical carbon dioxide as the working fluid have the potential to yield higher thermal efficiencies due to several key factors like higher operating pressures, the reduced compression work (due to the high density of sCO_2 and subsequent lower specific volume), among others. Thermal-power cycles operating with sCO2 show a high potential in future power generation systems with applications including fossil fuel, nuclear power and waste recovery (White et al., 2021). Some notable examples of sCO_2 plants include the STEP facility (10 MW capacity) and the Sandia National Laboratories (Marion et al., 2022).
2. **Direct cooling with carbon dioxide** is foreseen as an appealing technology for future detectors operating at warm temperatures, as specified in the ECFA Detectors R&D Roadmap Process Group (ECFA Detectors R&D Roadmap Process Group, 2021). A more general application in cooling of electronics was never considered until very recently, where the increase in power density and size reduction of electronic packages shifted the interest from air-cooling to advanced heat transfer technology (Dehdashti Akhavan et al., 2023). Indeed, while cooling of electronic components has been limited almost exclusively to conventional single-phase or flow boiling systems, sCO_2 has shown very interesting properties such as high heat transfer coefficients and low viscosity, together with the minimal footprint of carbon dioxide when compared to other commercial refrigerants (Da Rosa et al., 2019). With regards to two-phase cooling, supercritical cooling has the advantage of showing significantly smaller pressure drops and avoiding instabilities commonly found in phase change heat transfer while showing acceptable values of heat transfer coefficient near the pseudo-critical point.

2 THE SUPERCRITICAL CONDITION

The critical point of carbon dioxide is 74 bar and 31 °C. At temperatures below the critical value, the variation of pressure with volume along an isotherm exhibits discontinuities where it intersects the saturation line; where phase change takes place - see Figure 1. The horizontal, constant pressure segments represent the presence of two distinct phases. Above the critical point, such discontinuities are not found. Macroscopically, there is a continuous variation from a liquid-like condition to a gas-like one. Above the critical pressure, at temperatures lower than the critical, the liquid-like (higher density) fluid is found. At temperatures higher than that, the gas-like fluid is found.

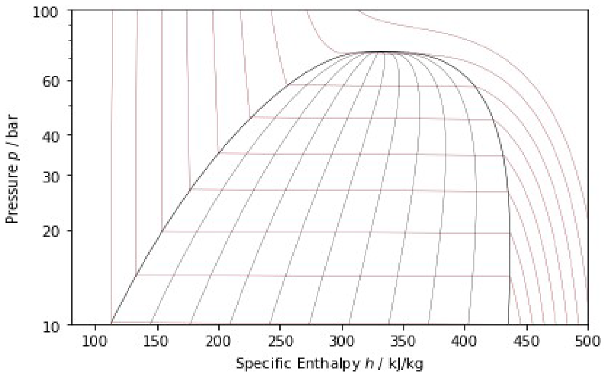
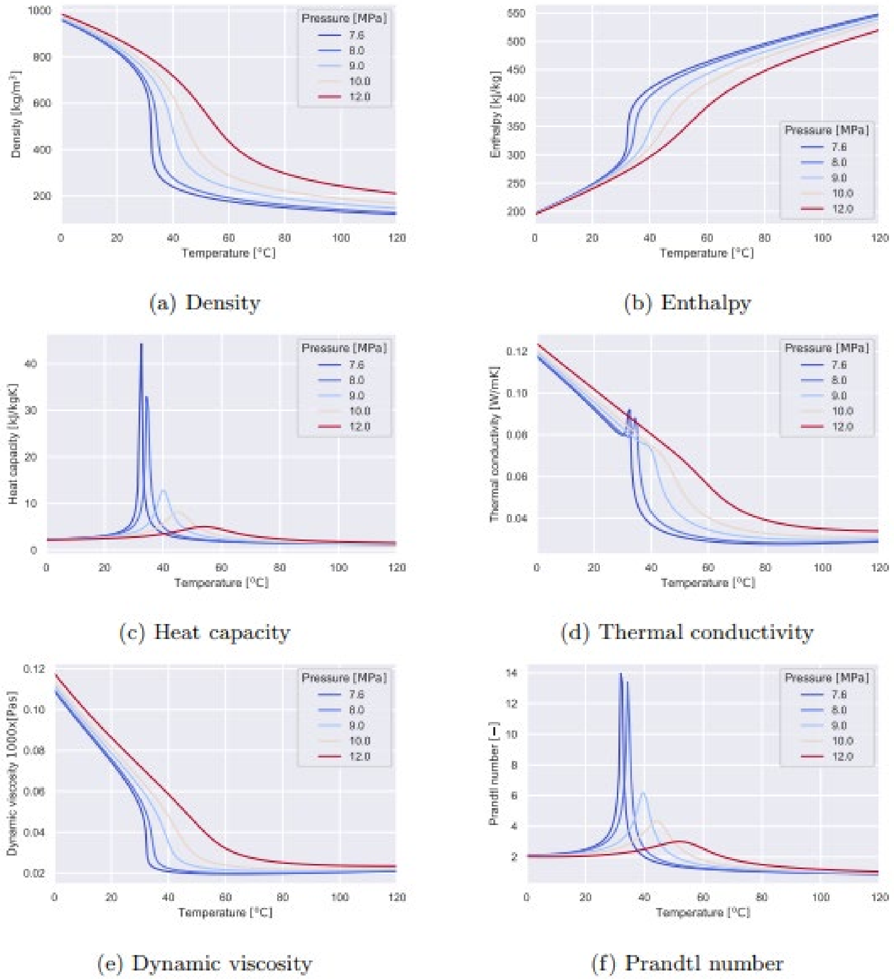


Figure 1. Pressure-enthalpy diagram of carbon dioxide

Figure 2 illustrates how supercritical CO_2 properties vary with temperature at different pressures. Above the critical pressure, specific heat and thermal conductivity peak near the pseudo-critical point, while density and viscosity drop sharply. The extreme dependence of fluid properties on temperature and non-uniformity of density can significantly affect the mean flow and turbulence fields as well as on the heat transfer effectiveness (Jackson, 2013). Variation in density can impact turbulence production due to flow acceleration from thermal expansion or buoyancy itself. Combined with strong variations in heat capacity and thermal conductivity, these factors may have important consequences in heat transfer effectiveness (Yoo, 2013). Therefore, understanding these property variations is essential when designing a cooling system with such fluid, since a very narrow range of possible operating conditions would not be feasible. In contrary, a thorough analysis of the possible (and feasible) operating parameters is paramount in the first stage of design. A thorough analysis of feasible operating parameters is vital in the initial design stage, requiring experimental determination and systematic study of heat transfer coefficients and pressure drops to identify optimal conditions.



(a) Density (b) Enthalpy

(c) Heat capacity (d) Thermal conductivity

(e) Dynamic viscosity (f) Prandtl number

Figure 2: Variation of physical properties of supercritical CO_2 at different pressures with respect to temperature

Finally, in the region where the specific heat reaches a maximum value, a considerable change in a very important parameter occurs: the relative work of expansion, defined as $E_q = pdV/dQ)_p = p\beta/\rho Cp$. While this value has values typical of droplet-like liquids (around 10^{-2}) below the pseudo-critical value, when the temperature approaches it, the relative work of expansion rises to rather high values of 0.2-0.4, characteristic of gases (Kurganov et al., 2012). Kurganov et al. used that parameter to determine the boundaries of what they called the pseudo-phase transition region. They defined hm_0 as the lower boundary for the enthalpy that satisfied the condition $E_q \leq 0.02 - 0.03$ and hm_1 as the enthalpy that yielded a E_q typical of carbon dioxide in an ideal gas state. They found that the difference between hm_1 and hm_0 yielded the latent heat of evaporation of the two fluids studied (water and carbon dioxide) at the same reduced pressure ($p/p_{cr} = 0.27$). Conceiving this region as a pseudo-phase transition region provides insights on why the heat transfer coefficient at supercritical pressures is comparable to cases at subcritical pressures, as well as why applying a well-established single-phase correlation does not yield valid results. This indicates that, although there is no phase change, one can (in a justified manner) conceptualize the issue as a pseudo-phase transition.

3 CURRENT STATUS OF RESEARCH

As specified in Section 1, the latest general review in heat transfer with supercritical fluids was published by Pioro, I.(Pioro, 2019) and provided an overview of the current status of heat transfer research in forced convection of fluids at supercritical pressures, however it was mostly oriented towards nuclear reactor applications. The majority of studies deal with heat transfer and hydraulic resistance of working fluids, where the main fluids are water and carbon dioxide. Pioro highlighted that satisfactory analytical methods for practical prediction of forced-convection heat transfer at supercritical pressures have not been developed due to the difficulty in dealing with steep property variations, especially at turbulent flows and high heat fluxes. However, this review was rather informative and mostly focused on prediction methods, which still showed uncertainties of ±25% for HTC values and ±15% for calculated wall temperatures.

A handful of test rigs have been built and aimed to study different aspects of heat transfer with supercritical carbon dioxide. In (Illyés et al., 2023), a list of experimental facilities built for sCO_2 is given, although most of them are focused on studying supercritical heat exchangers in order to develop thermal-power cycles. In contrast, test benches that focus on the measurement of the heat transfer coefficient of carbon dioxide in supercritical conditions are scarcer, although Ehsan et al. (Ehsan et al., 2018) provided a detailed analysis of the existing studies back in 2018. One can also divide the heat transfer studies in two: heating and cooling. Cooling is more related to gas cooler applications and therefore to the first application described in this report. In this case, for electronics cooling, one would have to look at the heating studies. Additionally, the common belief that sCO_2 heat transfer is an established and well-known field is rather false, considering that many studies reporting heat transfer coefficients report values that differ significantly in magnitude at similar operating conditions. In addition to that, sometimes the authors fail to specify all the operating parameters in their studies, as can be seen in Ehsan et al.'s review paper.

The main issues studied when investigating the heat transfer performance of sCO_2 are the understanding of the parametric trends and the development of prediction methods to estimate the heat transfer coefficient and pressure drop (in much less magnitude) at supercritical conditions. The parameters commonly investigated are mass flux (flow velocity), heat flux, operating pressure, inlet temperature, pipe dimensions and flow direction. Most articles that aim to provide a parametric study of the effects of different variables in the heat transfer coefficient and pressure drop present challenges in interpretation and do not provide a clear trend in all the parameters evaluated. This is caused by a lack of consensus in critical aspects of data interpretation, with emphasis in the analysis and study of heat transfer deterioration and how to quantify it. In this regard, the effect of pressure has been the most well understood, since its main effect is the dampening of the peak in the specific heat capacity (as well as the variation of other physical properties), thus reducing the maximum heat transfer coefficient attainable keeping other process conditions constant. The other parameters have been more difficult to address, since they play a role on the main force responsible for convection, thus adding an extra layer of complexity to the process. The effect of different parameters in the heat transfer coefficient is further discussed in Section 4.

In what respects the predictive methods, a considerable number of correlations have been developed to predict the heat transfer coefficient of supercritical fluids (Cabeza et al., 2017). However, none of them has managed to provide a unique universal correlation applicable for a given geometry (Guo et al., 2020). So far, the review works on heat transfer correlations of sCO_2 have been conducted focusing specially on the comparison of the existing correlations with an experimental dataset. A detailed analysis of the heat transfer mechanism and fluid flow behaviour of sCO_2 has not yet been conducted (Ehsan et al., 2018).

The correlations used for sCO_2 can be grouped into Dittus-Boelter type and Petukhov-Kirillov (Gnielinski) type. Researchers in the field of sCO_2 have modified both general correlations by means of correction terms to account for the drastic variations in thermophysical properties, buoyancy, and acceleration effects. In many cases, correlations have been applied blindly, without consideration of the ranges of applicability, which has led to an overpopulation of correlations trying to correct the already existing ones. It has not been uncommon that the correlations evaluated have not been relevant to the experimental dataset they have been applied to (Ehsan et al., 2018). Additionally, Pioro et al. (L. Pioro, 2020) and Kurganov et al. (Kurganov et al., 2012) wrote that a peak in the heat conductivity in the pseudocritical point was not even recognized before that time. Kurganov et al. also highlighted the need to expand knowledge in the field of heat transfer for developing more reliable methods of theoretical design and increasing the level of empiric correlations that generalize experimental and calculation data. Prior to any literature summary, the main aspects of heat transfer with supercritical fluids need to be defined:

- **Normal Heat Transfer** is regarded as the process through which normal heat transfer coefficients are found. In this case, normal refers to those of sub-critical convective heat transfer far from the critical region, when calculated with a single-phase Dittus Boelter-type correlation.

- **Enhanced Heat Transfer** is characterized by higher values of heat transfer coefficient, when compared to those obtained at normal conditions, and therefore lower wall temperatures.

- **Deteriorated Heat Transfer** is defined as the abnormal increase of the local wall temperature, caused by a significant drop of the heat transfer coefficient. Temperatures can exceed the limiting temperature of the tube material and increase the risk during operation, since temperature differences can reach above 200 degrees (Kurganov et al., 2012). The topic of HTD (Heat Transfer Deterioration) remains the most studied in the field, with many dedicated scientific studies and reviews attempting to find the explanations to the phenomena observed, as well as to define prediction methods. The general feature of DHT is the strong dependence of the local heat transfer on the previous history of the process (Kurganov et al., 2012), which results in a strong effect of inlet conditions on the temperature curves. Additionally, conditions that do not affect normal heat transfer, such as wall roughness or existence of forced pressure oscillations (due to pump operation), play a role in DHT regimes.

4 PARAMETRIC TRENDS

Several articles have aimed to provide a uniform evaluation of the trend of the heat transfer coefficient and wall temperature profile with the variation of process parameters: diameter, mass flux, heat flux and process pressure. Firstly, only four articles have evaluated the effect of the pipe diameter in heating of sCO_2. Table 1 contains the conclusions reached by the different studies. Ehsan et al. (Ehsan et al., 2018), in their first analysis, highlighted the difficulty of drawing conclusions due to the scarcity of experimental data, however they did add to their conclusion that, as opposed to cooling of sCO_2, the heat transfer rate increased with increasing diameter for heating conditions. In the small diameter region, Liao and Zhao (Liao and Zhao, 2002) reported the opposite trend as Wang et al.(Wang et al., 2020) with very similar diameters tested. For higher diameters, Zahlan et al. (Zahlan et al., 2015) reported very similar heat transfer coefficients for different diameters at similar flow conditions.

Table 1. Comparison of different conclusions reached by four different groups of research in the topic of sCO2 heating

Research article	Diameter mm	Flow	Conclusion
Liao and Zhao, 2002 (Liao and Zhao, 2002)	0.70 1.40 2.16	Horizontal	Nusselt number increases as the diameter increases
Wang et al., 2020 (Wang et al., 2020)	0.50 0.75 1.00	Horizontal	HTC increases as the diameter decreases
Bae and Kim, 2009(Bae and Kim, 2009)	4.40 9.00	Vertical upward	Obtained generally higher HTCs in their smallest pipe but did not draw a conclusion from it
Zahlan et al., 2015 (Zahlan et al., 2015)	8.00 22.00	Vertical upward	Trends of HTC were very similar at comparable flow conditions

Secondly, looking at the effect of the mass flux, the research articles that have evaluated its effect on the local heat transfer coefficient value are listed in Table 2. For horizontal flows, two articles evaluated its effect in small-diameter pipes and reached a uniform conclusion: the heat transfer rate increased with the mass flux. However, in cases where the flow was vertical upward, three of the four studies found an uncanny trend: higher mass fluxes led to lower values of heat transfer coefficient, different from what has been commonly observed. Zhang et al. (Zhang et al., 2018) commented that at low mass fluxes, the contribution of forced convection was weak, but natural convection driven by buoyancy was enhanced. However, Zahlan et al. (Zahlan et al., 2015) did not comment on this issue, but his data matrix did not include what was considered by other authors as ”low mass flux” values.

In the case of downward flow, the only article found in heating of sCO_2 in this matter reached the same conclusion found in horizontal flows. The main difference between upward and downward flow is the direction of the buoyancy force, which is also the main effect of buoyancy of heat transfer in forced flow. In downward flows, the direction of the buoyancy component is the same as the fluid velocity component, which means that buoyancy enhances or promotes forced convection (referred to as assisting flow); whereas in upward flows the directions are opposite and is called opposing flow. In the first case it is important to note that although buoyancy effects can significantly enhance heat transfer for laminar forced convection flows, enhancement is typically negligible if the forced flow is turbulent (Bergman and Incropera, 2011). The effect of buoyancy on turbulent convective heat transfer in vertical tubes is a complex topic, even for conventional fluids such as water or air at normal pressures (Jackson, 2017). In this respect, the effect of buoyancy in systems where the density is non-uniform changes the radial distribution of shear stress, which modifies the amount of turbulence being produced, thus affecting the diffusion of heat across the flow by turbulent action. In vertical tubes, the effect of buoyancy in convective heat transfer causes an initial impairment of the heat transfer; however, beyond a particular strength of buoyancy, the heat transfer recovers, and the heat transfer rate is enhanced. The physical explanation to such phenomena was explained in detail in (Jackson, 2017).

In what respects the effect of heat flux, just like the mass flux, its effect is very dependent on the flow direction. Xie et al. (Xie et al., 2020) commented on the fact that the critical heat flux (onset for HTD) was not proportional to the mass flux, since many authors observed HTD under different mass flux conditions at equal values of q/G. The effect of heat flux is immensely related to the topic of heat transfer deterioration, which is very broad and can therefore not be explained in detail in this report. In general, the effect of heat flux should be regarded at different values of mass flux, meaning that its effect depends on the magnitude of the flow velocity; since the interplay between the flow velocity and the buoyancy force experimented due to sharp density variations will determine the flow distribution and thus, have an effect in the heat transfer coefficient. This issue brings us again into discussing forced and natural convection, since the main difference between the two is the relative importance of buoyancy and the external force triggering fluid flow. Xie et al. observed that, for moderate mass fluxes an increase in heat flux led to a reduction in the heat transfer coefficient for upward flows. In the contrary, low mass fluxes showed a critical value above which the heat transfer coefficient worsened after increasing the heat flux, and below which the trend increased proportionally. For horizontal flows, Guo et al. (Guo et al., 2020) obtained lower heat transfer coefficients at higher heat fluxes in their 2 mm inner diameter pipe.

The most evaluated effect of the heat flux is its link to the appearance of HTD. In this regard, low mass fluxes showed less sensitivity to HTD in the 16 mm pipe with upward flow studied by Zhang et al.(Zhang et al., 2018). For Reynolds numbers below 40000, no value of heat flux triggered a peak in the wall temperature in their experimental campaign. Kline et al. (Kline et al., 2018) noted that there is a range of inlet temperature values for which HTD occurs for each diameter, mass flux and heat flux, instead of simply a heat flux that applies to all inlet temperatures. Recently, Zhu et al. (Zhu et al., 2022) demonstrated HTD occurrence at various temperatures under fixed parameters, with temperatures near but below the pseudocritical point being more susceptible to HTD peaks. The effect of system pressure was briefly summarized in Section 3.

Table 2. Comparison of different conclusions reached by four different groups of research in the topic of sCO2 heating: effect of mass flux on the heat transfer coefficient

Research article	Diameter mm	Flow	Mass flux kg/m^2s	Conclusion
Zhang et al., 2018	16	Vertical upward	<300	Heat transfer behaviour of sCO_2 is completely different at low mass flux when compared to higher values. Threshold of ”low mass flux” was fixed at G<300 kg/m2s. They obtained higher values of HTC at lower mass fluxes.
Zahlan et al., 2015	8 22	Vertical upward	197 to 2027	Higher mass fluxes led to higher heat transfer coefficients in both of their test sections.
Research article	Diameter mm	Flow	Mass flux kg/m^2s	Conclusion
Jiang et al., 2008	4.40 9.00	Vertical upward	24 34 49.5	”HTCs for smaller mass flow rates are larger than those for larger mass flow rates due to the variation of the thermophysical properties and buoyancy”
Peng et al., 2022	8.00 22.00	Vertical upward	100 to 1200	”Heat transfer behaviour is much different under different mass flow rates with similar q/G”*. They defined the threshold for ”low mass flux” at values equal or below 200 kg/m2s. At values below, increasing the mass flux reduces the HTC, whereas at values above 200, an increase in the mass flux leads to higher HTC.
Guo et al., 2020	2	Horizontal	400 500 550	Heat transfer coefficient increased with increasing mass flux
Wang et al., 2020	1	Horizontal	1202.5 1025.7 848.8 672	Local heat transfer coefficient increased when increasing the mass flux
Zhang et al., 2022	4	Vertical downward	318.8 to 955.2	Obtained higher wall temperatures (therefore lower local heat transfer coefficients) with lower mass fluxes

*In their research article, q/G varies from 0.65 (at 100 kg/m^2s) to 0.375 (400 kg/m^2s).

Finally, only Wang et al. (Wang et al., 2014) provided a study on the friction factor calculation for heating of sCO_2 and reported a good agreement with established correlations only in laminar flow. For transition regimes, the comparison between their experimental data and five different established methods showed absolute errors over 20%. In the turbulent region, models that consider the effect of surface roughness and Reynolds number lead to smaller absolute errors than models that only considered the latter. Interestingly, their references did not include a single study with experimental values of heating sCO_2, which shows the lack of experimental data under such process conditions.

5 CORRELATIONS

As highlighted in Section 3, although the different correlations developed by authors in the field of sCO_2 is vast, there has not yet been an established and generalised method to predict the heat transfer under different process conditions. The VDI (VDI e. V., 2010) states that the Nusselt number for fully developed turbulent flow in smooth pipes can be obtained by means of Eq. (1), which was put forward by Gnielinski, deriving from the theory of momentum transport by Petukhov and Kirillov.

$$Nu_m = \frac{(\varepsilon/8)\ Re\ Pr}{1 + 12.7\sqrt{\varepsilon/8}\,(Pr^{2/3} - 1)}\left[1 + (d_i/l)^{2/3}\right] \qquad Eq.(1)$$

Where $$\varepsilon = (1.8 logRe - 1.5)^{-2} \qquad Eq.(2)$$

and the rest of the parameters are obtained as follows:

$$Nu_m \frac{\alpha d_i}{\lambda}; Re = \frac{v d_i}{\mu} \qquad Eq.(3)$$

This correlation is applicable for the range described below:

$$10^4 \leq Re \leq 10^6$$

$$0.1 \leq Pr \leq 1000 \; d_i/l \leq 1$$

The physical properties of the fluids are referred to the mean temperature. If the properties of the medium are temperature sensitive, the Nusselt number can be corrected by means of Eq. (4).

$$Nu = Nu_m \left(\frac{Pr}{Pr_w}\right)^{0.11} \qquad Eq.(4)$$

, which is valid when $0.1 \leq Pr/Pr_w \leq 10$.

Additionally, the Nusselt number can be corrected with a factor based on T/T_w for gases, see (VDI e. V., 2010). In this report, the first approach based on the Prandtl number has been used to contrast with existing data and discuss the status on sCO_2 heat transfer research.

To solve the previous equations, an iterative method has been implemented in MATLAB R2022b. For this analysis, the pipe length has been discretized in N, meaning that the total heat flow has been divided into different segments dz as shown in Figure 4. The inlet of each segment is noted by the subscript i, and the outlet as $i+1$. Firstly, the inlet conditions of the different research articles have been specified. The discretization of the pipe has been performed with the condition of $dQ = Q/N$ where N is the number of steps.

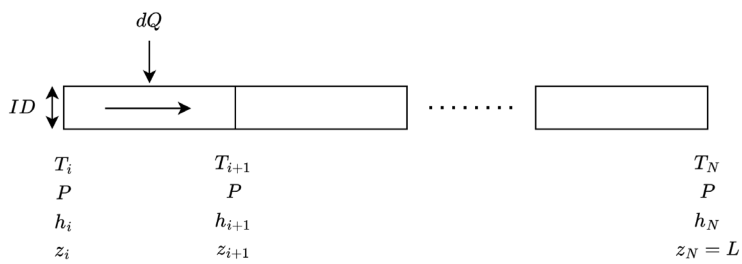


Figure 3: Discretization of the pipe length

Once the inlet conditions are known and dQ is calculated, the fluid conditions at the fluid core for each step can be calculated by means of Eq. (5).

$$dQ = m(h_{i+1} - h_i) \qquad Eq.(5)$$

When the bulk fluid temperature at each step is known, the iterative method to calculate the Nusselt number (therefore heat transfer coefficient and wall temperature) can be performed. As specified in the VDI, the fluid properties are evaluated at $T_m = (T_i + T_o)/2$, where i is the inlet temperature for each dz and o is the outlet temperature of each pipe fragment. In this case, the wall temperature obtained is the inner wall temperature, however for this analysis no radial conduction has been considered and therefore inner and outer wall temperatures are considered equal.

Another more widely applied and simpler correlation is the Dittus-Boelter correlation in Eq. (6), applicable in the range $0.6 \leq Pr \leq 160$, $Re > 10000$ and $L/D \leq 10$; and may be used for small to moderate temperature differences, with all properties evaluated at an averaged temperature T_{avg}.

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \; Eq.(6)$$

Applying the Gnielinski correlation to data published by Tanimizu et al. (Tanimizu and Sadr, 2016) results in the heat transfer coefficients and wall temperatures shown in Figure 5, while the results from applying the Dittus-Boelter correlation are shown in Figure 6. Both figures include the data extracted from the research article. Both correlations are, in theory, applicable in all the range of operating conditions.
Comparing their experimental results, it was observed that the campaign did not exhibit the parabolic trend predicted by both correlations. This discrepancy is largely due to the strong dependence of both correlations on the Prandtl number, which significantly influences the heat transfer coefficient. Closer to the inlet and further from the pseudo-critical point, the Dittus-Boelter correlation overestimated the experimental values

by about 60%, while the Gnielinski correlation showed an even greater discrepancy. The relative error between the correlations and the experimental data increased as the fluid temperature approached the pseudo-critical point. At lower temperatures, the fluid behaves more like a liquid, making liquid-based correlations more applicable. However, near the pseudo-critical point, various parameters critically affect the radial temperature distribution and physical properties, causing the correlations to deviate from the experimental values. It's noteworthy that the Gnielinski correlation accounts for the effect of heat flux only through the factor $(Pr/Pr_w)^{0.11}$, since the rest of the variables are evaluated at an average fluid temperature. evaluating other variables at an average fluid temperature. This factor can adjust the temperature difference between the wall and bulk fluid by a factor between 0.77 and 1.28. However, the Prandtl number is not the sole parameter influencing convective heat transfer. The erratic results from these well-established correlations align with the lack of consensus highlighted by several authors, as mentioned in the introduction.

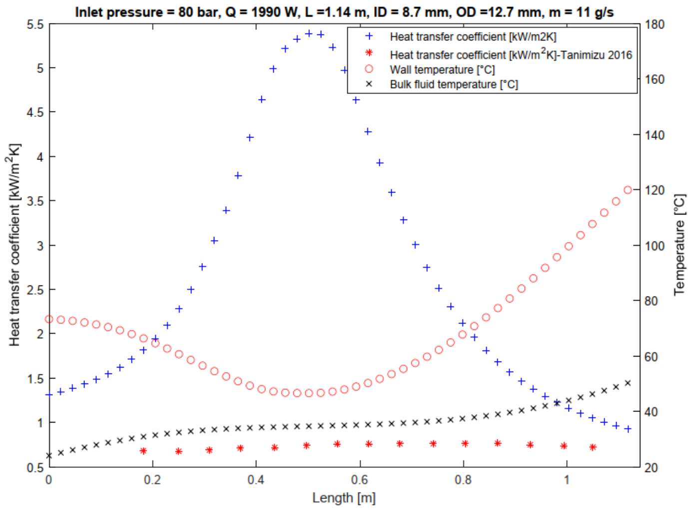


Figure 4: Application of the Gnielinski correlation to the initial conditions of Case 1 in the research article from Tanimizu, 2016 (Tanimizu and Sadr, 2016); and data from their research article (Case 1)

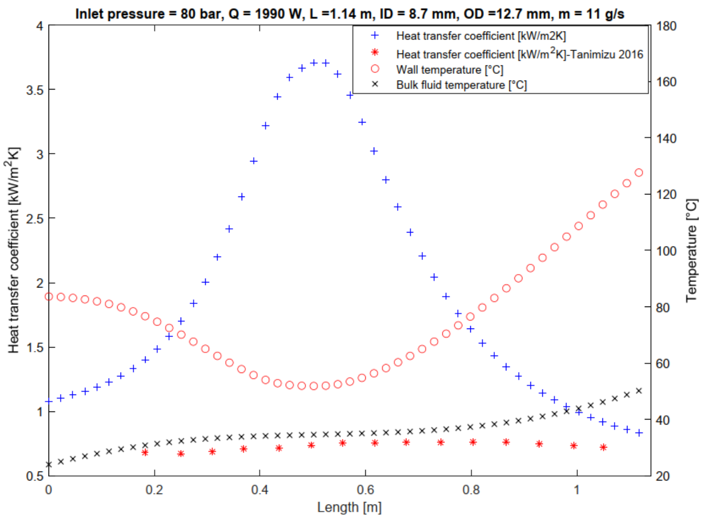


Figure 5: Application of the Dittus-Boelter correlation to the initial conditions of Case 1 in the research article from Tanimizu, 2016 (Tanimizu and Sadr, 2016); and data from their research article (Case 1)

6 TEST SETUP

A facility is currently being built at CERN to study heat transfer of carbon dioxide at supercritical conditions. In view of the difficulties in the field, a main requirement for a setup is the capacity of performing measurements in a wide range of parameters. This means, the obtention of heat transfer coefficients and pressure drops must be carried out while varying other parameters as the mass flux, heat flux, inlet temperature and flow orientation, among others. Therefore, a test matrix has been defined and is shown in Table 3. Additionally, the process flow diagram is shown in Figure 7 and is described below.

A cooled hydraulic metal bellow accumulator (AC101) is used in order to bring carbon dioxide to the desired pressure, where nitrogen gas from a bottle is used to pressurize the liquid. The pressure on the gas side of the bellow accumulator is set by means of the pressure regulator FR101 (Gloor, MANODETENDEUR 200/200 bar). Once the fluid pressure is set to its desired level, the fluid is pumped by means of a magnetic driven rotary vane pump LP101 (VA02 MODULAR, M-PUMPS). After the pump, any debris or free particles (>3µm) that could be in the fluid are filtered. A by-pass valve is installed to regulate the mass flow through the test section, MV105. After the by-pass flow regulation, a Coriolis mass flow meter is installed (RHEONIK RHM02 with transmitter RHE45). A further regulating valve was installed to have more flexibility in the regulation of the flow, MV106. The fluid is then pre-heated in EH101 (CAST-X 500 heater, 1200W) to reach the desired temperature conditions at the inlet of the test section. The test section is delimited by a dashed square in the PFD. The test section consists of a 1-meter-long pipe, with diameters ranging from 1 mm to 3 mm, where a constant heat flux is applied by means of a DC power supply (Delta Elektronika SM52-30). Additionally, an initial fragment with length of 50 times the diameter is left unheated to allow the flow to develop. After the test section, the fluid is cooled down to the accumulator temperature in a plate heat exchanger HX101 (SWEP B4THx20) by means of cooling water (supplied by a chiller CU101, VDH5000A) and is sent back to the start of the process.
Pressures and temperatures are measured in different points of the test rig (PT and TT shown in Figure) with absolute pressure sensors (KELLER PAA-23SX) and PT100 (RODAX). The test section is equipped with type K thermocouples (0.2 mm wire diameter, Roessel Messtechnik) at different locations, spaced 50 mm from each other, electrically insulated with electrically insulating liquid tape and attached to the tube by means of steel epoxy putty. The test section is insulated in a box filled with Armaflex, contained in a structure that allows to tilt the whole section to 90 and -90 degrees. The rest of the fluid connections are also insulated by means of Armaflex. All the CO_2 connections are Swagelok fittings and SS316L pipes.

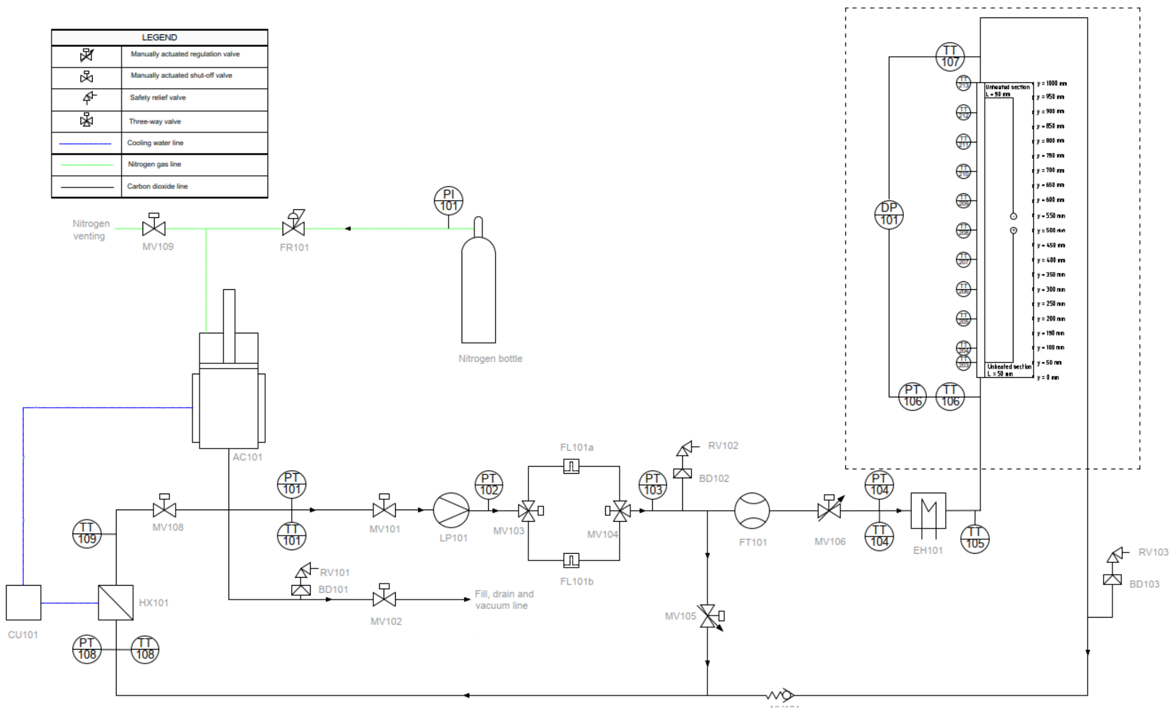


Figure 6: Process and instrumentation diagram of the setup being built at CERN

Table 3. Test matrix for the setup being built at CERN.

Orientation	Pressure Bar	Diameter mm	Mass flux kg/m^2s	Heat flux	Inlet temperature
Vertical upward Horizontal Vertical downward	50 to 120	1 2 3	1600 1200 800 500	Up to 115 kW/m^2	From 20 to 40 °C

7 CONCLUSIONS

This study assessed the current research status on heating of sCO_2. Experimental studies and their main findings were summarized, showcasing the lack of uniform conclusions in many crucial parameters that influence heat transfer with carbon dioxide at supercritical pressures. Moreover, the obvious lack of uniformity and established understanding of the underlying factors affecting heat transfer with supercritical carbon dioxide was shown. The effect of process parameters like the diameter, mass and heat flux were evaluated. In this regard, the conclusions reached by different studies on the effect of the diameter on the heat transfer rate was inconclusive. Additionally, a crucial influence of the flow direction on the effect of the mass flux was shown, however rather uniform trends were reported by different authors. Finally, the effect of heat flux was briefly evaluated since its influence on the process is extensive and could not be included in this short manuscript. However, the main parameters governing heating of sCO_2 were discussed to prove the need for uniform and systematically obtained data. In addition to that, two generic common correlations were applied to experimental data, revealing significant differences in the obtained values with respect to data obtained experimentally. The reasons for the discrepancies were briefly assessed. All the above justified the need for the design of an optimized setup to study sCO_2, which was presented together with the test matrix to be studied in the coming years.

NOMENCLATURE

p	pressure (bar)	E_q	relative work of expansion *(-)*
T	temperature (C)	V	volume *(m^3)*
Q	heat flow (W)	B	thermal expansion coefficient (1/K)
ρ	density (kg/m^3)	Nu	Nusselt number
Cp	heat capacity (J/kg·K)	Re	Reynolds number
α/HTC	heat transfer coefficient ($W/m^2 \cdot K$)	Pr	Prandtl number
d	diameter (m)	λ	heat conductivity (W/m·K)
v	velocity (m/s)	μ	dynamic viscosity (Pa·s)
h	enthalpy (kJ/kg)	l/L	length (m)
OD	outer diameter	ID	inner diameter (m)
m	mass flow (kg/s)	ε	friction correction coefficient

Subscripts

cr	critical	i	inner
w	wall	m	mean

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