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Determination of Heat Transfer Coefficients During In-Tube Gas Cooling of Supercritical Carbon Dioxide

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ABSTRACT

The world-wide agreement to restrict the use of chlorofluorocarbons (CFCs) and hydrofluorocarbons has prompted recent research exploring the possibilities of replacement refrigerants, particularly those occurring naturally in the environment. One such natural refrigerant, carbon dioxide, has been shown for particular applications to have a COP which is competitive with current refrigerants. However, in order to use carbon dioxide, entirely new components must be designed to accommodate the high pressures. In fact, the entire heat rejection process of a carbon dioxide vapor compression cycle may take place above the critical temperature and pressure. Hence, the design of the heat rejecting heat exchanger requires knowledge of carbon dioxide transport properties associated with temperatures and pressures above the critical point. While some correlations for in-tube cooling of supercritical carbon dioxide do exist in the literature, only very recently has there been heat transfer research of carbon dioxide with respect to its role as a refrigerant.

An experiment at Herrick Laboratories at Purdue University sponsored by ASHRAE is exploring the nature of in-tube cooling of supercritical carbon dioxide. The test apparatus built exclusively for the experiment is being used to quantify the local heat transfer coefficients as well as pressure drop associated with temperature ranges and refrigerant mass fluxes which correspond to typical refrigerating capacities. In addition, the test stand is being used to investigate the transport properties of supercritical carbon dioxide with typical compressor oils entrained in the refrigerant and with in-tube surface enhancement.

INTRODUCTION TO CARBON DIOXIDE

Background

Regulations prohibiting the use of chlorofluorocarbons (CFCs) as refrigerants by industrialized nations are in place, and guidelines for phasing out hydrochlorofluorocarbons (HCFCs) as refrigerants in the future have already been agreed upon by nations of the world. This has prompted researchers worldwide to consider the development of new refrigerants and/or the investigation of refrigerants occurring naturally in the environment. However, the development of new fluids to be used as refrigerants may result in the same dilemma the world currently faces with several fluorocarbon based refrigerants: the potential for damage to the earth's environment. There is no possible way of knowing the long-term environmental or health impact of a brand-new refrigerant with absolute certainty. The use of a substance which is found naturally in the biosphere as a refrigerant eliminates the need for such speculation. Carbon dioxide is a substance which not only is naturally occurring in the environment, but has a successful history of use as a refrigerant. Prior to the development of CFCs in the 1930s, carbon dioxide was widely used as a refrigerant in cooling human occupied enclosures. Ironically, carbon dioxide has recently been considered as a replacement for fluorocarbon based refrigerants (which made carbon dioxide obsolete as a refrigerant). Recent research has indicated that although carbon dioxide may not be a superior refrigerant for all air conditioning/refrigerating applications, there are certain situations for which carbon dioxide may be suitable. These cases take advantage of the thermophysical differences of carbon dioxide compared to the HCFCs.

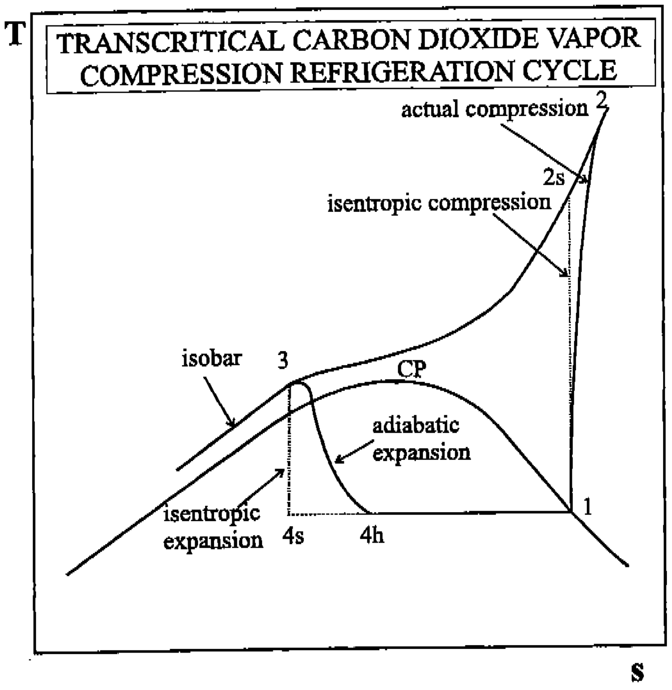


Figure 1: Temperature-entropy diagram for carbon dioxide transcritical process

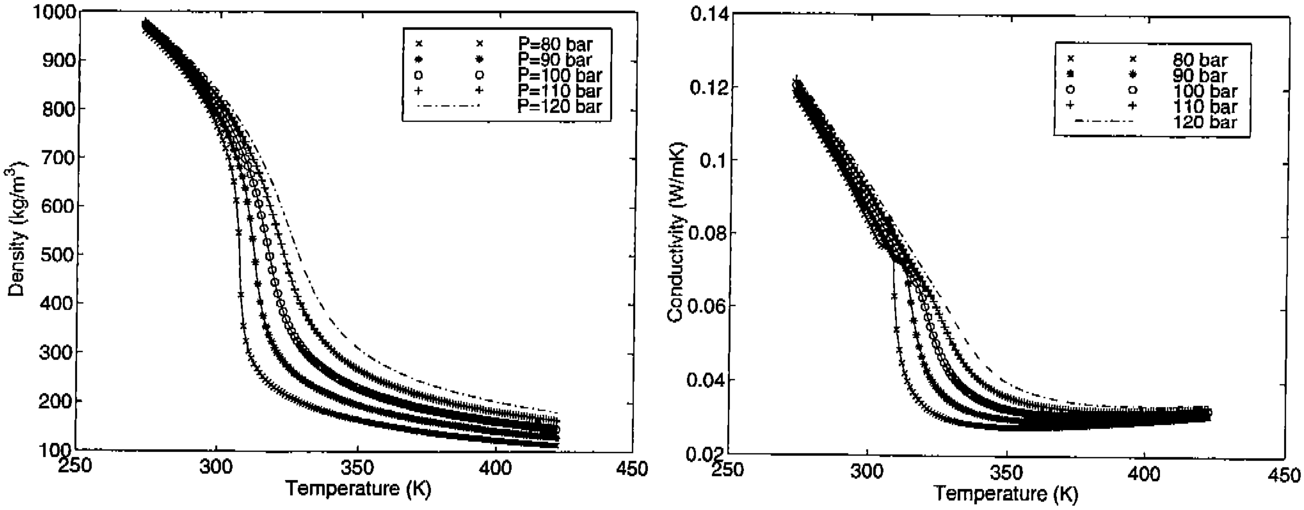
The Transcritical Carbon Dioxide Cycle

Figure 1 shows the transcritical carbon dioxide vapor compression cycle process as depicted on a T-s diagram (Process path 1-2-3-4). The transcritical cycle is made up of the same four basic processes that make up the vapor compression cycle upon which CFCs and HCFCs are based: compression (1-2), heat rejection (2-3), expansion (3-4), and heat absorption (4-1). The process which deviates most from current conventional refrigeration is the supercritical heat rejection process. The critical temperature of carbon dioxide is 30.82°C and the critical pressure is 73.53 bars. Therefore in order for a carbon dioxide based air conditioner to reject heat on a warm summer day (temperature $\approx$ 32°C), the temperature of the carbon dioxide will be higher than the critical temperature (and pressure) during the heat rejection process. Consequently, instead of a constant temperature two-phase heat rejection process which occurs with fluorocarbon-based refrigerants, heat is expelled from the carbon dioxide while it experiences no phase change.

A need for research of the supercritical heat rejection process

The significant differences between current conventional refrigeration and transcritical carbon dioxide refrigeration necessitate the use of newly designed components for two primary reasons: 1) The pressure corresponding to heat rejection may be as high as 100 bars, if not higher [Robinson & Groll 1998]. Also, the evaporation pressures which occur in a carbon dioxide-based transcritical cycle are much higher than those found in current fluorocarbon-based cycles (up to 8 times higher) [Robinson & Groll 1998]. This indicates that there is no practical way to adjust or retrofit current air conditioning/refrigeration equipment to use carbon dioxide. 2) In order to develop a transcritical carbon dioxide device which operates at the highest possible COP, individual components must be designed to take advantage of the unique thermophysical properties of carbon dioxide.

The supercritical heat rejection process is one for which there is very little known information upon which to base a heat exchanger design. In order to design such a heat exchanger, detailed information concerning the transport properties of carbon dioxide must be known. Recently, a review of the literature on supercritical carbon dioxide properties and in-tube heat transfer has been made [Pitla, Robinson, Groll & Ramadhyani 1998]. Of the dozens of published works on supercritical carbon dioxide heat transfer, only



(a) (b)

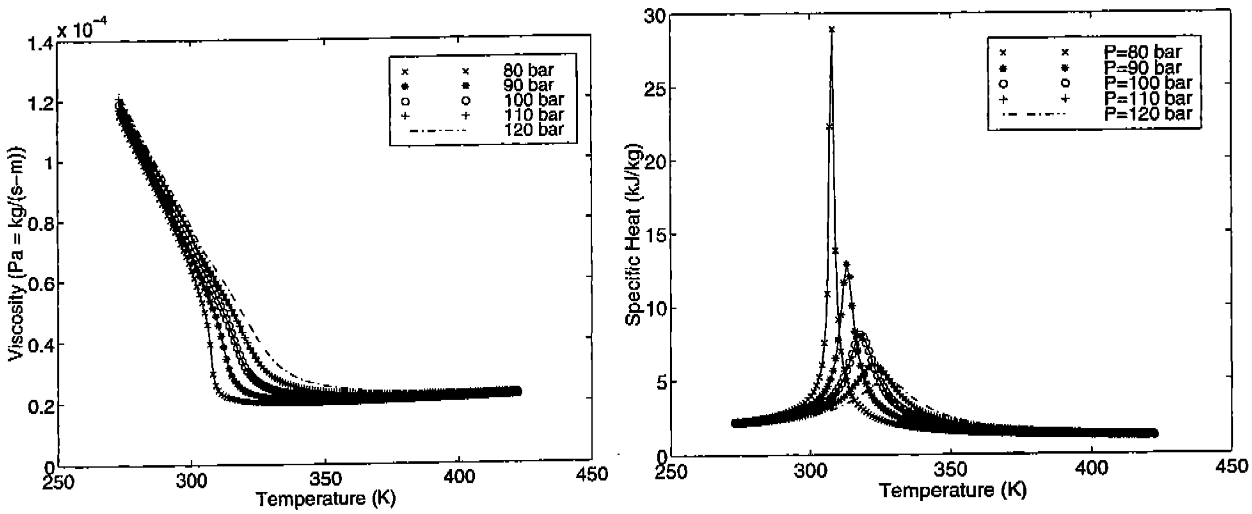
Figure 2: (a) Supercritical carbon dioxide density vs. temperature (b) Supercritical carbon dioxide thermal conductivity vs. temperature

three offered heat transfer correlations for the cooling of supercritical carbon dioxide, and only one studied this process in a horizontal flow arrangement (the most likely flow arrangement for a refrigeration/air conditioning application). None of the reviewed investigations considered carbon dioxide heat transfer as part of refrigeration or air conditioning application.

The literature review did indicate that for pressures just above the critical point ($\approx 1.0\text{-}1.2P_c$), carbon dioxide (and presumably other fluids) experiences large variations in transport properties as the fluid is heated above or cooled below what is known in the literature as the 'pseudocritical' temperature. This temperature (above critical) corresponding to a particular pressure above critical is the temperature where the specific heat is at a maximum. Figures 2 and 3 show the variation of density, thermal conductivity, viscosity, and specific heat as a function of temperature for various pressures above critical. A large change is noted for each property in the vicinity of the psuedocritical temperature (which increases with increasing pressure). It is hypothesized that for a typical air conditioning application, the supercritical carbon dioxide in a transcritical cycle will likely be cooled such that it approaches or drops below the pseudocritical temperature. The knowledge of large property variation around the pseudocritical temperature and the lack of knowledge of in-tube cooling of supercritical carbon dioxide with respect to a refrigeration application has prompted an investigation of heat transfer characteristics during in-tube cooling of supercritical carbon dioxide through experiment.

DESCRIPTION OF THE TEST APPARATUS

The investigation of the nature of in-tube cooling of carbon dioxide includes a series of experiments to be conducted on a test stand built specifically for this purpose. A schematic of the test apparatus is shown (not to scale) in Fig 4. The heat transfer test section consists of eight subsections connected in series, where each subsection is a tube-in-tube counterflow heat exchanger. There are five subsections which are 1.7 m in length and three which are 1.3 m in length. Carbon dioxide flows through the inside stainless steel tube (OD=6.35 mm, ID=4.57 mm) entering the test section at location (a) and exiting at location (b), while water flows through the outer tube (copper, ID=15.75 mm) in counterflow entering the test section at location (b) at about room temperature. The water will be pumped by a centrifugal pump through a turbine mass flow meter prior to entering the test section. Exiting the heat transfer test section, the carbon dioxide enters a water/glycol bath (location (c)) where it is subcooled by constant temperature bath to around 0°C. A receiver downstream from the subcooling heat exchanger maintains a buffer of liquid-like supercritical carbon dioxide. Following the subcooling section, the carbon dioxide is pumped using a sealless magnetic drive gear pump (location (d)) through a micro-motion mass flow



(a)

(b)

Figure 3: Supercritical carbon dioxide viscosity vs. temperature (b) Supercritical carbon dioxide specific heat vs. temperature

meter (location (e)). Following the mass flow meter, the carbon dioxide flow enters a coil inside a heating tank (location (f)) filled with heat transfer fluid at approximately 140°C, after which the flow re-enters the heat transfer test section. Between each subsection, temperature measurements of the inside wall of the stainless steel tubes are taken by a type T probe thermocouple, the tip of which is fixed in a small hole in the stainless steel tube. At the same locations, pressure lines connect the carbon dioxide bearing stainless steel tubes to differential pressure transducers. Valves have been installed on the pressure lines so that the differential pressures of four of the subsections can be measured at one time. At the line carrying water from one subsection to the next, bulk temperature measurements are made using two probe thermistors at each location. Accurate water temperature measurements will be necessary to determine the rate of heat transfer of each subsection (assuming a constant specific heat and a steady flow of the water). At three evenly spaced (in the axial direction) positions on long subsections and at 2 evenly spaced positions on the shorter subsections, two T-type thermocouple beads have been spot-welded to the outside surface of the stainless steel tubes within the subsection. The beads were insulated from the water using epoxy with the intent of measuring the outside temperature of the stainless steel tube. By making an assumption concerning the local heat flux through the stainless steel tube, it is expected that the surface thermocouple measurements will lead to a reasonably accurate estimate of the inside surface temperatures of the stainless steel, assisting in determining local heat transfer coefficients. All fluid bearing lines in the test section are well insulated.

TEST MATRIX

Table 1 shows the test matrix which specifies parameters used to define the experiment. These values represent temperatures, pressures, heat fluxes, flow rates, and oil concentrations which are representative of transcritical cycle heat rejection for a typical air conditioning application. As mentioned previously, a thermodynamic analysis of the transcritical cycle indicated that for a heat rejection outlet temperature of 40°C and a range of evaporation temperatures from -40 to 5°C, the cycle COP would be maximum at a heat rejection pressure of around 100 bars, which is the middle value of the supercritical pressure range to be tested. The same study indicated that for an evaporation temperature of 0°C, an evaporator capacity of about 4.5 kW would correspond to a mass flow rate of 0.03 $\frac{kg}{s}$, which also is the middle value of the mass flow rate chosen for the test matrix. In addition, Robinson and Groll found that for the range of evaporation temperatures, the temperature exiting the compressor should be at least 120°C. The heat fluxes in the test matrix correspond to the temperature changes for the given mass flow rates with the 4.572 mm inside diameter stainless steel tubes used in the experiment.

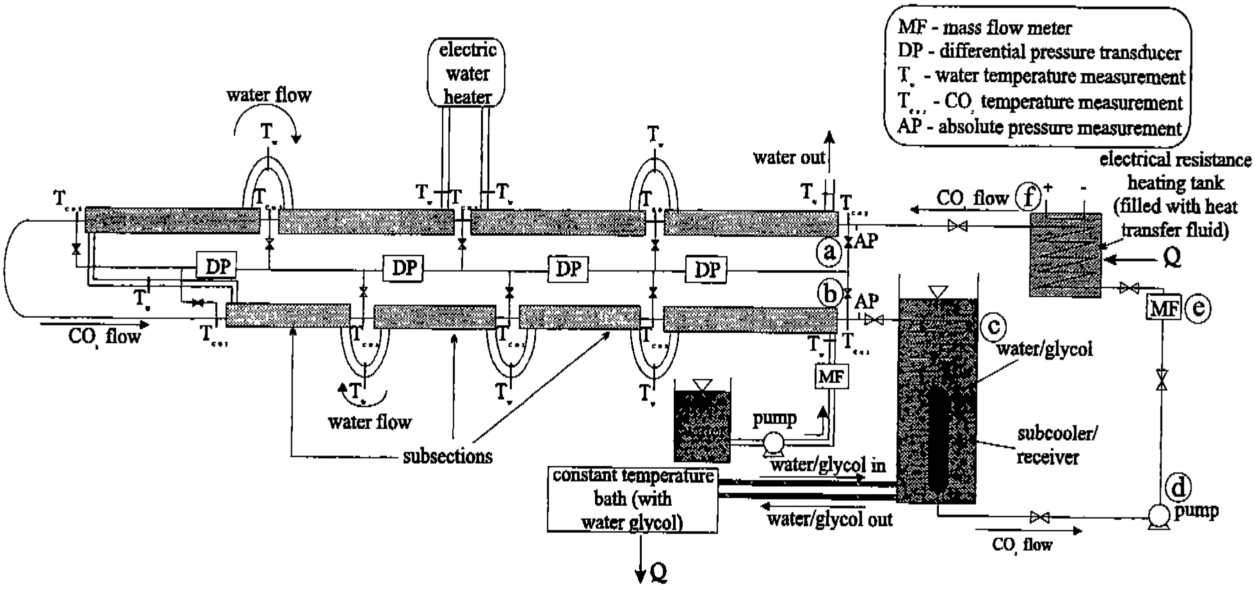
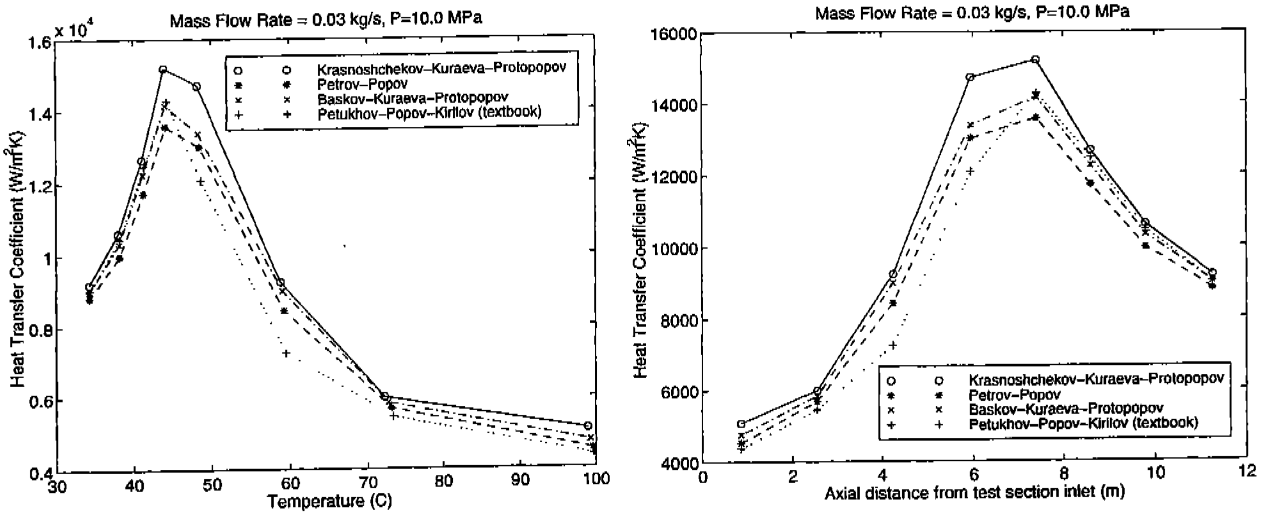


Figure 4: Supercritical carbon dioxide heat transfer experimental test stand

Temperature Inlet-Outlet (°C)	Pressure Level (bar)	Mass Flux ($\frac{kg}{s-m^2}$)	Heat Flux ($\frac{W}{m^2}$)	Oil Concentration	Tube
100-20	80	1020	26800	0%	smooth
110-25	100	1530	40200	2%	macro-fin
120-30	120	2040	169400	5%	

Table 1: Experiment Test Matrix



(a)

(b)

Figure 5: (a) Calculated supercritical carbon dioxide heat transfer coefficients vs. temperature (b) Calculated supercritical carbon dioxide heat transfer coefficients vs. axial distance from test section inlet

THEORETICAL RESULTS

In an effort to gauge the magnitude of the heat transfer coefficients that will be obtained with the test stand by cooling supercritical carbon dioxide, a heat transfer analysis of the test stand using the median values of the pressure and mass flow rate specified by the test matrix has been carried out. The heat transfer coefficients have been calculated based on three correlations found in the literature as well as a textbook correlation for single phase heat transfer (as a reference) [Incropera & Dewitt 1990]. Figure 5 (a) shows the heat transfer coefficients plotted as a function of temperature. Figure 5 (b) shows the heat transfer coefficients plotted as a function of axial distance from the test section inlet. The peak in the heat transfer coefficient occurs in the vicinity of the pseudocritical temperature. For carbon dioxide, the pseudocritical temperature is 35°C at 80 bars, 45°C at 100 bars, and 54°C at 120 bars. For this reason, in order to make more discreet measurements at the peak in heat transfer, the shorter subsections were positioned in the test section such that the pseudocritical temperature occurs during the shorter subsections. Also, the temperature ranges for the test matrix were chosen such that the expected peak in the heat transfer coefficient should occur during the experiment for each supercritical pressure.

The figures illustrate the difference between the heat transfer coefficients predicted by the supercritical cooling correlations and those predicted by the single phase textbook correlation. The discrepancy between the textbook correlation and the others is due to fact that the cooling correlations account for a temperature gradient in the radial direction, while the textbook correlation takes into account only the bulk fluid temperature. Further explanation of the cooling correlations and the interpretation of the form of their heat transfer coefficients can be found in a detailed literature review [Pitla et al. 1998].

CONCLUSIONS AND FUTURE WORK

Conclusions of the theoretical analysis

Based on the theoretical analysis of the heat rejection process, it is expected that the proposed experiment will:

- generate large peaks in the heat transfer coefficient of in-tube, cooled supercritical carbon dioxide in the vicinity of the pseudocritical temperature.
- indicate heat transfer coefficients , the magnitude of which are generally measured only for two-phase heat transfer.

- produce data which will allow accurate internal heat transfer correlation to be developed for in-tube cooled supercritical carbon dioxide.

Future Work

In the next step, the entire experiment will be conducted in accordance with the test matrix. When that is completed, the resulting data can be used to develop a new cooling correlation for supercritical carbon dioxide for the purpose of modeling the supercritical heat rejection process.

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