THEORETICAL SIMULATION AND EXPERIMENTAL VALIDATION OF A CRITICAL AND TRANSCRITICAL CO2 REFRIGERATION SYSTEM USING A DUAL CAPILLARY TUBE EXPANSION DEVICE

Fourie, M, Dobson, RT
Department of Mechanical and Mechatronic Engineering, University of Stellenbosch, Stellenbosch, Private Bag XI, MATIELAND, 7602
Tel 27 (0)21 8084286, fax 27 (0)21 8084958, e-mail, rtd@sun.ac.za, coenraad@sun.ac.za

Abstract

The objective of this paper is to present the approach that has been adopted in theoretically simulating a CO_2 refrigeration system that uses a multiple of different (length and diameter) capillary tube type expansion devices. The results of literature reviewed in an attempt to establish the state of the art of CO_2 refrigeration control systems will be presented and discussed. The assumptions needed to establish the model will be discussed. The experimental system that is to be built to validate the theoretical model will be described. This theoretical model entails assuming a realistic geometry for the refrigeration system. The system is then discretised into a number of finite sized control volumes and the equations of change (continuity, momentum and energy) are then applied to each control volume and the resulting system of finite difference equations solved numerically. The progress to date will be presented and discussed, and conclusions drawn upon which recommendations will be based as the best way forward.

Keywords CO2, refrigeration, capillary tube, trans- and subcritical, global warming

1. Introduction

In the nineteenth century, carbon dioxide (CO_2) was widely used as a refrigerant; however, hydrochlorofluorocarbons (HCFCs) displaced CO_2 in the 1930s because lower system pressures could be realized. Two decades ago, due to the ozone depleting potential (ODP) of the HCFCs, the Montreal Protocol (1987) dictated the abolishment of HCFCs as refrigerants. New synthetic refrigerants, namely hydrofluorocarbons (HFCs), were introduced. Although HFCs (such as R134a) have no ODP, their global warming potential (GWP) is much higher than that of the so called natural refrigerants; such as carbon dioxide and ammonia. Considering that GWP equals 1 for CO_2, the GWP of the HFCs can be as high as 3000. Therefore there has been much research on the development of environmentally safe refrigerating systems using natural working fluids. Recently, CO_2 has been considered as a promising alternative refrigerant of HFCs. The challenge being faced is to ensure that the energy efficiency and the cost of the CO_2 refrigeration systems are at an acceptable level. Indeed large international companies such as Coca Cola and large South African companies such as Pick and Pay and Woolworths are consciously including sustainability, pollution and global warming with their future marketing strategies and have thereby shown their commitment to the use of CO_2 refrigeration systems in the future.

The burning of fossil fuels creating CO_2 and the release of harmful refrigerants into the atmosphere contribute greatly to the effect of Global Warming (Wikipedia, 2010). Current predictions suggest that there will be an increase in temperature of 2 to 3℃ and a sea level rise of 0.5 m by the end of the 21st century. The Montreal Protocol (1987) issued a statement that the use of all CFCs and HCFCs as refrigerants are to be phased out in order to reduce ozone depletion. The Kyoto Protocol was established in 1997 and identifies six groups of global warming gases: Carbon dioxide, methane (CH_4), nitrous oxide (N_2O), HFCs, PFCs and sulphur hexafluoride (SF_6). (Heap, 2001)

CO_2 has been used since the 1850's, but the practice was abandoned due to the difficulty of using CO_2 at the extremely high pressures – up to 120 bar – required for transcritical operation and also due to its very low critical temperature of 31.5℃. CO_2 has two free – or independent – variables in the transcritical region – where it operates most efficiently - namely temperature and pressure. Other gasses only operate in the subcritical region, where the temperature and pressure are related to each other and there is only one free variable. The two free variables of CO_2 creates an optimum performance point, unlike other refrigerant gasses. This point is often referred to as the Coefficient of Performance (COP). Obtaining the COP in a CO_2 system is difficult, as it fluctuates significantly with varying temperature and pressure. However, with newer technology and a better understanding of CO_2 and its properties, it has become possible to use CO_2 effectively and economically as a refrigerant. (Wikipedia, 2010)

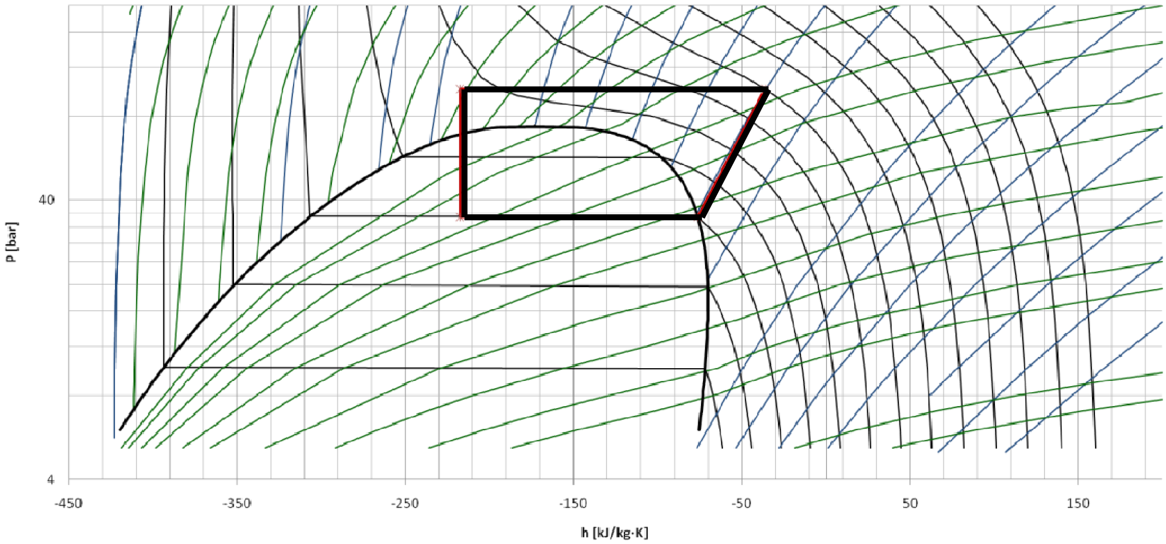

Figure 1 P-h Diagram of a typical CO_2 Transcritical System

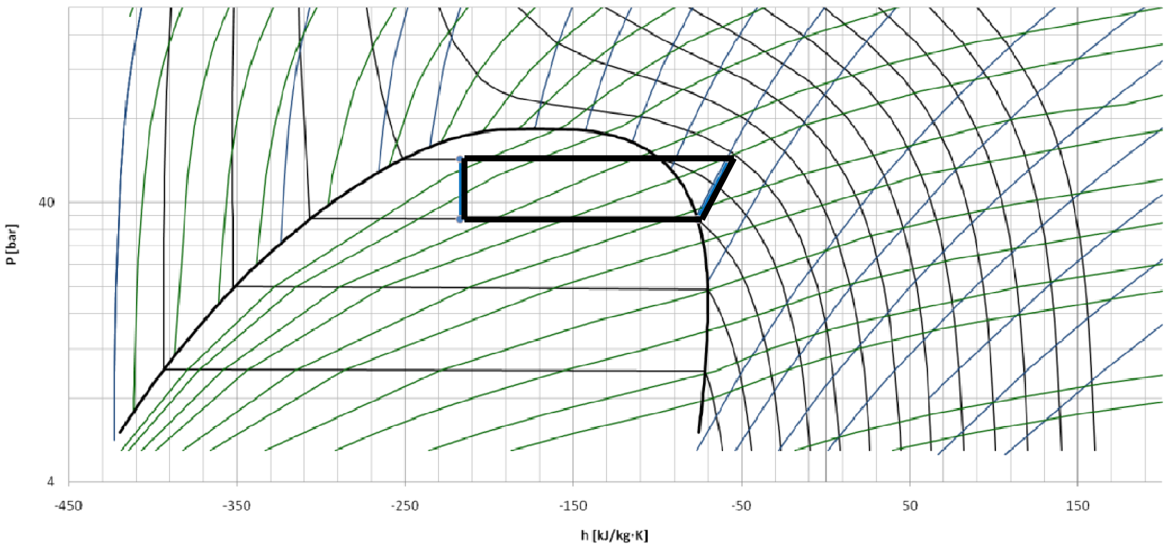

Figure 2 P-h Diagram of a Typical CO_2 Subcritical System

CO_2 is a highly desirable refrigerant to use due to its zero global warming potential and no ozone depletion properties. There are, however, many difficulties in using CO_2 as a refrigerant. Several

studies addressed these difficulties, but none provided conclusive and complete results for a completely independent and fully operational CO_2 subcritical and transcritical refrigeration system.
A study by Madsen, et al. (2005) showed that a capillary tube is an excellent choice as an expansion device in both subcritical and transcritical regions. However, the study included the use of a bypass needle valve in the expansion process. Similar studies concluded the same results, that a capillary tube gives better performance and that it is a competitive solution to other types of expansion units. This project will only use capillary tubes as the expansion device and will also have a system that switches between different capillary tubes during operation. This eliminates the need to switch the refrigeration system off between capillary tube exchanges.

The compressors used for CO_2 refrigeration systems are usually designed to run either in subcritical or transcritical due to the extreme difference in operating conditions between the two regions. Bock Compressors has manufactured a prototype compressor that can operate in both sub- and transcritical regions. It is the aim of this project to use the Bock compressor and also present them with the test results obtained from the experimental setup.

An important facet of the project is the comparison of the experimental results with the theoretical simulation model results. Major deviations between the experimental- and theoretical results can point to possible errors in the experimental system or the simulation.

1.1 Objectives

The project scope is to build, test and theoretically simulate a small CO_2 refrigeration system that can operate in both subcritical and transcritical regions. The system will use a multiple of capillary tube type expansion devices, of varying length and diameter. The capillary tubes will interchange to achieve optimum performance, depending on the ambient conditions of the system. The results obtained from the experimental setup will be compared to the results obtained from the simulation model.

1.2 Future Plans

A small, commercial/residential size system will be built and will include basic components that can be obtained “off-the-shelf”, along with the prototype Bock compressor. The system will then be tested under various ambient conditions by changing the temperature of the water supplied to the evaporator and the gas cooler. Temperature and pressure measurements will be taken throughout the system and this will be compared to theoretical values. Theoretical data will be obtained from the simulation model that is currently being written in the Microsoft Excel 2007 environment with VBA.

1.3 Experimental and Theoretical Process

The steady state model of the theoretical simulation in MS Excel will be finished by December 2010. Along with the first phase of the simulation, the design of the experimental system will be completed. The transient model will then be completed while the experimental system is under construction. Basic tests will be performed on the experimental system, once it is completed, to determine if it works to our expectations.

More thorough tests of the system will follow and all data obtained from the experiments will be logged. This data will later be compared to that obtained from the simulation model. A conclusion will then be drawn up to state the accuracy and efficiency of the system.

1.4 Layout of report

2. Theory

2.1 Literature Survey

Several experimental and theoretical studies on using CO_2 as the refrigerant in the transcritical region have been conducted to date. A basic refrigeration cycle with CO_2 includes a compressor, a gas cooler/condenser (depending on operating parameters), an expansion device and an evaporator. More complicated systems can include an Internal Heat Exchanger (IHX), liquid accumulators, oil separators, expansion tanks, multiple compressors, a cascade system, etc. The type of expansion device can also vary: from a capillary tube to a vortex tube to an ejector-expansion device to a pressostatic expansion valve and more. Refrigeration systems can also vary in size, from large refrigeration plants to small residential or commercial units.

Cabello, et al. (2008) made an experimental evaluation of the energy efficiency of a CO_2 refrigeration plant, which operated under transcritical conditions. They included an IHX and a liquid receiver into the basic CO_2 system and used a pressostatic expansion valve - along with an electronic expansion valve as a secondary expansion unit. All the heat exchangers in the system were shell-and-tube units. Their objective was to test the performance of the refrigeration plant under various conditions - varying evaporator temperature, gas cooler outlet temperature and gas cooler pressure. They performed a total of 92 steady-state tests - each 20 minutes long - and compared their findings with theoretical work as done by Liao, et al. (2000), Sarkar, et al. (2006), Kauf (1999) and Chen & Gu (2005). They concluded that their experimental results confirm Sarkar's expression of optimal gas cooler pressure. They also concluded that a highly accurate system to control the gas cooler pressure is required, since minor deviations or errors in pressure control can cause a major reduction of the efficiency of the system. They used REFPROP – an Excel add-in - to calculate the transcritical properties of CO_2.

Tao, et al. (2010) conducted an experimental study on the performance of CO_2 in a residential air-conditioning system. They included an IHX, a liquid receiver and an oil separator into their system and utilized and electronic expansion valve. Their system differs from the previous (by Cabello, et al.) mainly in their use of air-cooled fin-and-tube heat exchangers. Their objective was to study the effects of working conditions on the system's COP. This included changing the inlet temperatures of the air at both the evaporator and the gas cooler and also changing the velocities of the air flow at both units. They concluded that the COP of the system is minimally affected by changes in both air velocity and air temperature at the evaporator side. They also found that the COP is greatly affected by changes in air temperature and change in air flow velocity at the gas cooler side and also by evaporator temperature, up to 27% increase and 20% decrease in COP. Finally, they concluded that the design, using a fin-and-tube heat exchanger for the gas cooler, must be optimized to enhance heat transfer and thus improve the COP. From this research the author noted that it would be far more effective and easy to use water cooled heat exchangers.

The previous two articles described the use of a transcritical CO_2 system and experimental studies conducted with varying ambient conditions. Madsen, et al. (2005) conducted a study of a transcritical CO_2 refrigeration system utilizing capillary tubes, which is more directly applicable to this project. They theoretically and experimentally tested the effects of capillary tubes – with varying lengths and diameters – in CO_2 cycles. In their simulation, they compared the use of a fixed pressure valve with that of capillary tubes and discovered that the latter performs better. They noted that a system that uses capillary tubes is sensitive to changes in evaporator temperature. They ocncluded that the sensitivity to evaporator temperature changes during start-up can partially be eliminated by using a fixed pressure valve - set at a higher pressure - and a capillary tube in parallel.

Agrawal & Bhattacharyya (2008a) conducted a comparative study on the performance of capillary tubes against expansion valves in a transcritical CO_2 system. Their study included using several capillary tubes with varying diameters, lengths and internal surface roughness'. Their system consisted of a basic refrigeration system, only including a liquid separator before the compressor - to protect the compressor. They set up a mathematical model where-in they calculated the necessary parameters in order to obtain the optimum the capillary tube length. The mathematical model was then generated into a computer simulation of the system. In their conclusion they state that it is indeed possible to theoretically determine the optimum length of capillary tube necessary for optimum performance of a CO_2 system.

Prior to the latter study, Sarkar, et al. (2006) conducted a theoretical simulation study of a transcritical CO_2 system used for simultaneous cooling and heating applications. The work represented in this journal is very similar to that of Agrawal and Bhattacharyya (2008) and it is necessary to go through both documents to fully understand the processes involved and the improvements made by the follow-up study. In the Sarkar, et al. study they mention a CO2PROP code that was developed and used to determine the properties of CO_2 at transcritical state.

Agrawal & Bhattacharyya (2008b) also studied the flow model present within capillary tubes in a refrigeration system. They compared homogeneous flow with separated two phase flow and came to the conclusion that the separated two phase flow models can be accurately predicted. The discrepancy between the separated two phase flow model and the homogeneous two phase model can be as high as 8%. They recommended in their conclusion that a homogeneous model can used and is preferred for its simplicity.

CO_2 refrigeration systems often operate under transcritical conditions. In this region, the transport properties of CO_2 are influenced by two free variables, namely temperature and pressure. Span & Wagner (2003) made several thorough studies into the equation of state of CO_2 in the transcritical region. Their studies produced complex equations that can be used to determine any and all properties of CO_2. Many have used their studies and created programs or sets of code that can be readily and easily used to determine the properties of CO_2 (Sarkar, 2009) (Cabello, et al., 2008) (MegaWatt Soft, 2010). While in-depth knowledge of the equation of state for CO_2 is not always necessary for theoretical and experimental studies, it is necessary to understand the basics and to acknowledge the complications that CO_2 in the transcritical region presents. It is just as important to understand the fundamentals of how a CO_2 transcritical cycle works. Danfoss (2008) published an article that explains the cycle, in both subcritical and transcritical regions. In their article they also cover a basic design methodology for CO_2 systems that include an IHX.

CO_2 transcritical systems can have a wide variety of expansion devices. Daqing & Groll (2005) studied a cycle with an ejector-expansion device. They concluded that CO_2 cycle can be improved by more than 16% just by utilizing the ejector-expansion device. Sarkar (2009) conducted a theoretical study into cycle optimization using a vortex tube expansion device. He concluded on an improved COP and lower expansion losses. While these studies are not directly apllicable to the project at hand, it does give helpful insight into theoretical study procedures. These kinds of studies, where other types of expansion devices are used, also offer an understanding of the different kinds of expansion devices and their individual influence on CO_2 cycles.

Adriansyah, et al. (2006) researched and developed an improvement of their prototype CO_2 air-conditioning and tap water heating plant. They discuss what measures were taken to improve their system overall, and share their experimental results. In their conclusion they state that they achieved a 18.7% higher evaporator capacity and a 14.5% higher cooling COP.

2.2 Mathematical Model

The theoretical simulation was written in Microsoft Excel with VBA Macros. Solving all the equations of the systems requires an iterative process, which is handled through MS Excel automatically by creating circular references. The simulation in MS Excel is only designed to solve steady state systems with parallel flow heat exchangers. Future versions of the simulation will be able to solve steady state as well as transient systems with either parallel or counter flow heat exchangers. CO2Tables is a MS Excel add-in provided by MegaWatt Soft that is used to determine the transport properties of the CO_2.

All the piping and fittings in the system are assumed to have negligible losses in both pressure and temperature. The evaporator and gas cooler are both tube-in-tube water heated/cooled heat exchangers.

The simulation program works by solving a simplified version of the refrigeration system. First, the IHX will not be included into the calculations of the system. Figure 3 shows the simplified approach to the layout of the system, as used in the simulation. It is simplified as follows:

- Parallel flow tube-in-tube evaporator and gas cooler heat exchangers of length 2 m and 4 m, respectively;
- No internal heat exchanger, but rather a temperature controlled receiver to ensure that only dry saturated vapour is available at the suction of the compressor;
- Speed controller (Nrpm) compressor with known operating characteristics given as,

$$\dot{m}_{comp} = \rho_1 V_{comp} N_{comp} \eta \quad (1)$$

where $\eta = \eta_v \eta_i$ (2)

and $\eta = f(T_1, P_1, P_2)$, (3)

$$\rho_1 = f(T_1, P_1) \quad (4)$$

- Ignore frictional effects in the pipe work, but not in the capillary tube, evaporator and gas cooler.

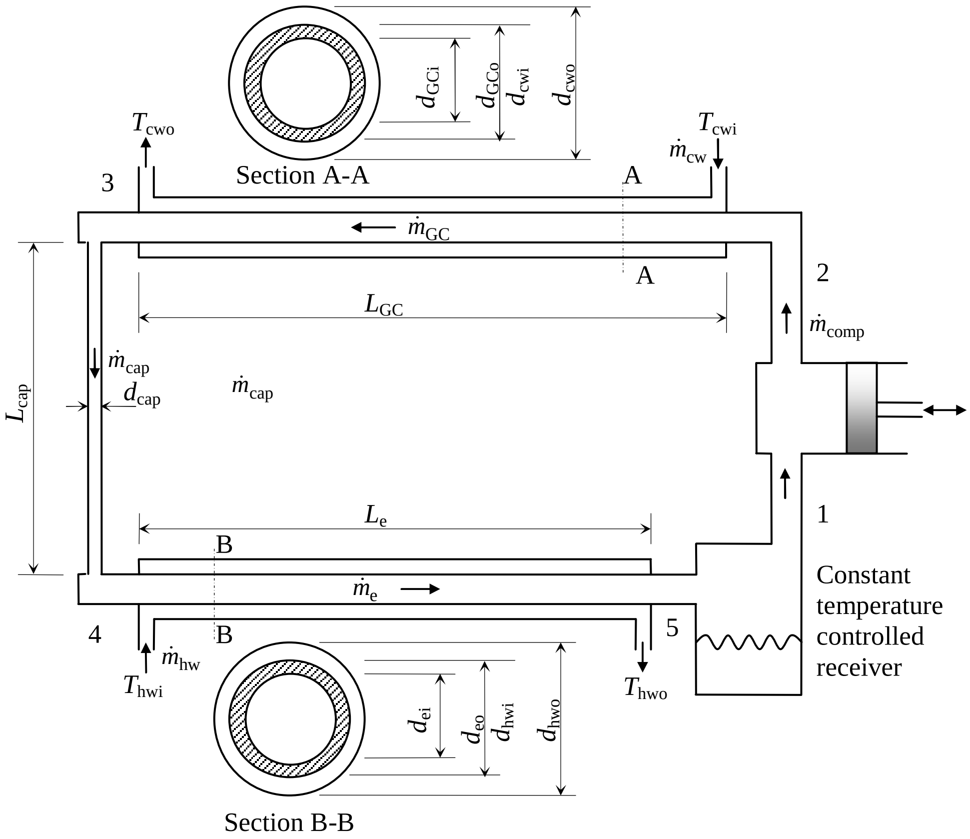

Figure 3 CO_2 Refrigeration system consisting of a compressor (comp), gas cooler (GC), capillary tube (cap), evaporator (e/ev) and constant temperature-controlled receiver

The pressure-enthalpy diagram of the simplified system follows,

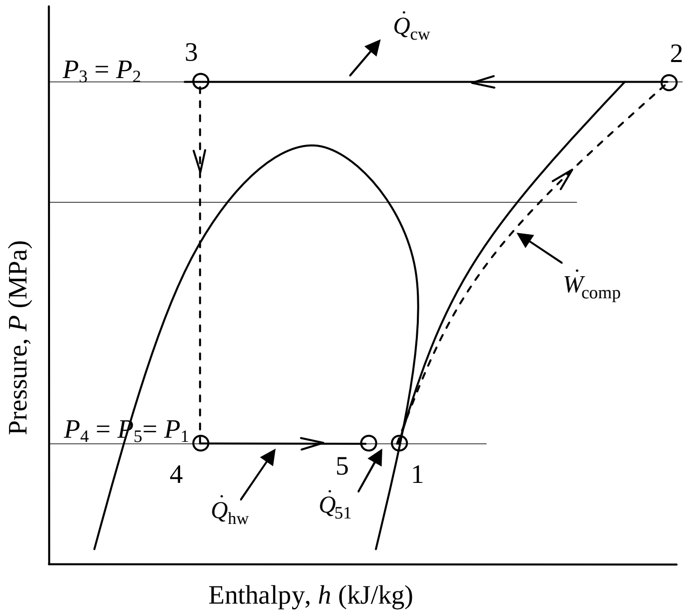

Figure 4 CO_2 Refrigeration system on a P-h diagram

Two-phase flow throughout the system is modelled according to a homogeneous flow model (Whalley, 1987).

The homogeneous density, ρ_h, is a function of the mass fraction, x, the density of the gas only, ρ_g, and the density of the liquid only, ρ_ℓ.

$$\rho_h = \frac{1}{\left(\frac{x}{\rho_g}\right)+\left(\frac{1-x}{\rho_\ell}\right)} \tag{5}$$

The homogeneous viscosity can thus be (arbitrarily) derived as,

$$\mu_h = \frac{1}{\left(\frac{x}{\mu_g}\right)+\left(\frac{1-x}{\mu_\ell}\right)} \tag{6}$$

Where μ_h is a function of mass fraction, viscosity of the gas only, μ_g, and viscosity of the liquid only, μ_ℓ.

The two-phase frictional multiplier for homogeneous flow, $\Phi^2_{\ell o}$, is as follows,

$$\emptyset^2_{\ell o} = \left(\frac{C_{fh}}{C_{f\ell o}}\right)\cdot\left(\frac{\rho_\ell}{\rho_h}\right) \tag{7}$$

However, it can generally be simplified as,

$$\emptyset^2_{\ell o} \approx \left(\frac{\rho_\ell}{\rho_h}\right) \tag{8}$$

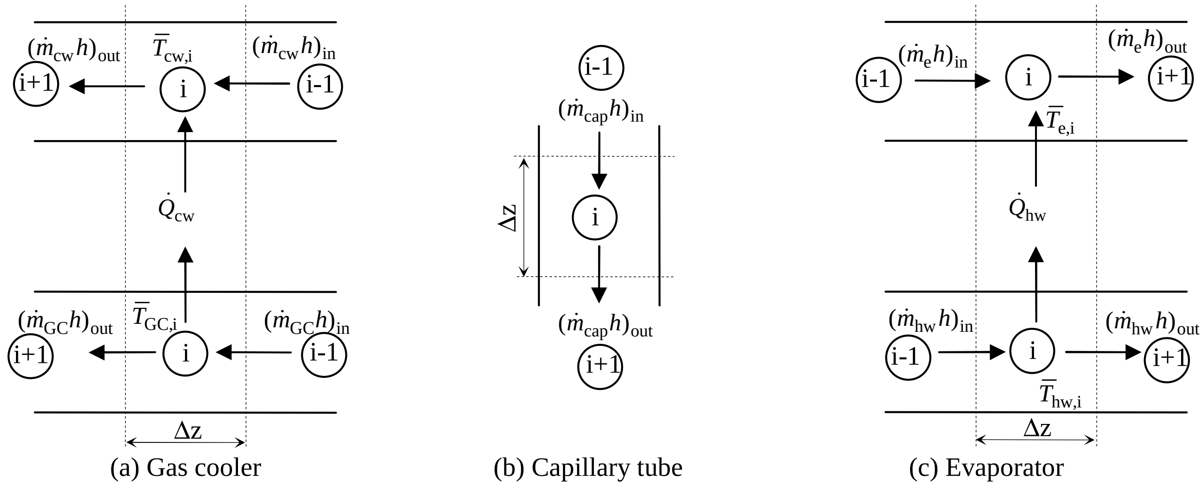

Figure 5 Gas cooler, capillary tube and evaporator i'th conservation of energy control volumes of length Δ*z*

Each of the components in the system can be divided into smaller control volumes of length Δz. Figure 5 shows the control volumes with information regarding the conservation of energy for the gas cooler, capillary tube and evaporator respectively.

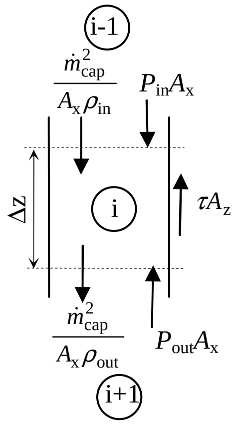

Figure 6 Capillary tube i'th conservation of momentum control volume of length Δz

Figure 6 shows the conservation of momentum in a control volume for the capillary tube. The conservation of momentum is also similar in the evaporator and the gas cooler.

2.2.1 Gas Cooler

The gas cooler process can be seen on figure 4 as the points (2) to (3).

Conservation of energy for the gas cooler is as follows:

$$\frac{\Delta E}{\Delta t} = \dot{E}_{in} - \dot{E}_{out} \tag{9}$$

Energy is flowing from the refrigerant to the water in the control volume. It is assumed that no energy is lost to the environment.

$$\dot{Q}_{GC,i} = \dot{m}_r \cdot \left(h_{GCr,i} - h_{GCr,i+1}\right) \tag{10}$$
$$\dot{Q}_{GC,i} = \dot{m}_w \cdot c_{phw,i} \cdot \left(T_{hw,i+1} - T_{hw,i}\right) \tag{11}$$

Utilizing the Log Mean Temperature (LMTD) method,

$$\dot{Q}_{GC,i} = \frac{\Delta T_{lm,GC,i}}{R_{GC,i}} \tag{12}$$

Where thermal resistance, $R_{GC,i}$, is,

$$R_{GC,i} = \frac{1}{h_{GCr,i} \cdot A_{GCi}} + \frac{1}{h_{hw,i} \cdot A_{GCo}} + \frac{ln\left(\frac{d_{GCo}}{d_{GCi}}\right)}{2\pi \cdot \Delta z_{GC} \cdot k_{GC}} \tag{13}$$

And the log mean temperature difference, $\Delta T_{lm,GC,i}$, is,

$$\Delta T_{lm,GC,i} = \frac{\left(T_{GCri,i} - T_{hwi,i}\right) - \left(T_{GCro,i} - T_{hwo,i}\right)}{ln\left(\frac{\left(T_{GCri,i} - T_{hwi,i}\right)}{\left(T_{GCro,i} - T_{hwo,i}\right)}\right)} \tag{14}$$

Pressure drop in the gas cooler follows from the conservation of motion law,

$$\frac{\Delta m}{\Delta t} = (\dot{m}u)_{in} - (\dot{m}u)_{out} + \sum F_z \tag{15}$$

Or, $0 = \dot{m}_1 u_1 - \dot{m}_2 u_2 + (P_1 - P_2)A - \bar{\tau} \wp \Delta z - \bar{\rho} A \Delta z g \sin\phi$ (16)

From equation (16), after some derivation and ignoring the effects of gravity, follows,

$$\Delta P_{GC,i} = -(\Delta P)_{F,GC,i} \cdot \Delta z_{GC} - (\Delta P)_{M,GC,i} \cdot \Delta z_{GC} \tag{17}$$

Where momentum pressure gradient, $(\Delta P)_{M,GC,i}$, is,

$$(\Delta P)_{M,GC,i} = \left(\frac{\dot{m}_r}{A_{GCx}}\right)^2 \cdot \left(\frac{1}{\rho_{GCho,i}} - \frac{1}{\rho_{GChi,i}}\right) \tag{18}$$

Where friction pressure gradient, $(\Delta P)_{F,GC,i}$, is,

$$(\Delta P)_{F,GC,i} = \left(\frac{\Delta P}{\Delta z}\right)_{F,lo,GC,i} \cdot \phi^2_{lo,GC,i} \quad \text{if } 0 \le x_{GC,i} \le 100 \tag{19.a}$$

$$(\Delta P)_{F,GC,i} = \left(\frac{\Delta P}{\Delta z}\right)_{F,h,GC,i} \cdot \phi^2_{h,GC,i} \quad \text{if } 0 \nleq x_{GC,i} \nleq 100 \tag{19.b}$$

Where friction pressure gradient for the liquid only, $(\Delta P)_{F,lo,GC,i}$, is,

$$\left(\frac{\Delta P}{\Delta z}\right)_{F,lo,GC,i} = \frac{-C_{flo,GC,i} \cdot \rho_{GCli,i} \cdot u^2_{lo,GC,i} \cdot \wp_{GC,i}}{2 \cdot A_{GCx}} \quad \text{if } 0 \le x_{GC,i} \le 100 \tag{20.a}$$

$$\left(\frac{\Delta P}{\Delta z}\right)_{F,h,GC,i} = \frac{-C_{flo,GC,i} \cdot \rho_{GChi,i} \cdot u^2_{h,GC,i} \cdot \wp_{GC}}{2 \cdot A_{GCx}} \quad \textit{if } 0 \nleq x_{GC,i} \nleq 100 \tag{20.b}$$

And where the two-phase friction multiplier, $\phi^2_{lo,GC,i}$, is,

$$\phi^2_{lo,GC,i} = \frac{\rho_{GCli,i}}{\rho_{GChi,i}} \quad \text{if } 0 \le x_{GC,i} \le 100 \tag{21.a}$$

$$\phi^2_{h,GC,i} = 1 \quad \text{if } 0 \nleq x_{GC,i} \nleq 100 \tag{21.b}$$

2.2.2 Evaporator

The evaporator model is similar to that of the gas cooler. The only difference now being that the water in the evaporator is seen as “cold” with respect to the water in the gas cooler. The evaporator process can be seen on figure 4 as the points (4) to (5).

2.2.3 Capillary Tube

The capillary tube process can be seen on figure 4 as the points (3) to (4).

Flow in the capillary tube is considered to be isenthalpic,

$$h_4 = h_5 \tag{22}$$

Pressure drop in the capillary tube follows out of the law of conservation of motion, the same as in equation (15).

$$\Delta P_{cap,i} = -(\Delta P)_{F,cap,i} \cdot \Delta z_{cap} - (\Delta P)_{M,cap,i} \cdot \Delta z_{cap} \tag{23}$$

Where momentum pressure gradient, $(\Delta P)_{M,cap,i}$, is,

$$(\Delta P)_{M,cap,i} = \left(\frac{\dot{m}_r}{A_{capx}}\right)^2 \cdot \left(\frac{1}{\rho_{capho,i}} - \frac{1}{\rho_{caphi,i}}\right) \tag{24}$$

Where friction pressure gradient, $(\Delta P)_{F,cap,i}$, is,

$$(\Delta P)_{F,cap,i} = \left(\frac{\Delta P}{\Delta z}\right)_{F,\ell o,cap,i} \cdot \phi^2_{\ell o,cap,i} \qquad \text{if } 0 \leq x_{cap,i} \leq 100 \tag{25.a}$$

$$(\Delta P)_{F,cap,i} = \left(\frac{\Delta P}{\Delta z}\right)_{F,h,cap,i} \cdot \phi^2_{h,cap,i} \qquad \text{if } 0 \nleq x_{cap,i} \nleq 100 \tag{25.b}$$

Where friction pressure gradient for the liquid only, $(\Delta P)_{F,\ell o,cap,i}$, is,

$$\left(\frac{\Delta P}{\Delta z}\right)_{F,\ell o,cap,i} = \frac{-C_{f\ell o,cap,i} \cdot \rho_{capli,i} \cdot u^2_{\ell o,cap,i} \cdot P_{cap}}{2 \cdot A_{capx}} \qquad \text{if } 0 \leq x_{cap,i} \leq 100 \tag{26.a}$$

$$\left(\frac{\Delta P}{\Delta z}\right)_{F,h,cap,i} = \frac{-C_{f\ell o,cap,i} \cdot \rho_{caphi,i} \cdot u^2_{h,cap,i} \cdot P_{cap}}{2 \cdot A_{capx}} \qquad \text{if } 0 \nleq x_{cap,i} \nleq 100 \tag{26.b}$$

And where the two-phase friction multiplier, $\phi^2_{\ell o,cap,i}$, is,

$$\phi^2_{\ell o,cap,i} = \frac{\rho_{capli,i}}{\rho_{caphi,i}} \qquad \text{if } 0 \leq x_{cap,i} \leq 100 \tag{27.a}$$

$$\phi^2_{h,cap,i} = 1 \qquad \text{if } 0 \nleq x_{cap,i} \nleq 100 \tag{27.b}$$

2.2.4 Receiver and Compressor

Both the liquid receiver and the compressor are simplified into single control volumes. On figure 4 the process of the receiver is shown from point (5) to point (1) and the compressor is from point (1) to (2).

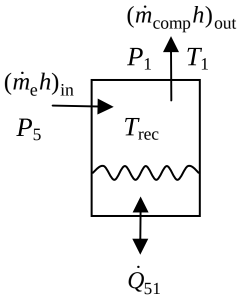

Figure 7 Receiver as a single control volume

The duty of the liquid receiver is to ensure that only saturated vapour is present at the suction point of the compressor.

The isentropic efficiency of the compressor is,

$$\eta_{is,c} = \frac{h_{2actual} - h_1}{h_2 - h_1} \tag{28}$$

2.2.5 Internal Heat Exchanger

Follows the same process as the gas cooler and evaporator. The major difference being that both the hot and the cold fluid/vapour is refrigerant.

3. Progress

3.1 Computer Simulation

The simulation program was divided into four stages, to simplify the development of the completed program. The first stage covers a steady state model with parallel flow heat exchangers. The second stage is also steady state, but simulating counter flow heat exchangers. The third and fourth stage simulate a transient model with parallel and counter flow heat exchangers respectively. The first stage of the simulation is near completion and the second stage will follow shortly. The third and fourth stages will only be completed in the following year.

The simulation has produced good results to date, results that are comparable with other theoretical work in the field. The MS Excel add-in, CO2Tables is used throughout the simulation to determine the transport properties of the CO_2.

3.2 Experimental Setup

The design of the refrigeration system is currently underway. This includes choosing off-the-shelf components from catalogues and designing parts such as the evaporator, gas cooler, IHX and capillary tubes. The experimental setup will consist of a basic CO_2 system and will include an IHX, one or two separation accumulators, an expansion tank and a capillary tube exchange system. Secondary components include T-type thermocouples, pressure sensors (gauges and transducers), sight glasses, flow meters, a compressor power meter, a tachometer for the compressor, shut-off valves (rated at 46 bar and 120 bar respectively) and 2 Safety Relief Valves (one at 46 bar and one at 120 bar). The system will also include a monitoring unit (for all the sensors), a frame/platform to secure the components in-place and the piping and fittings necessary to connect all the components.

Most of the off-the-shelf parts have already been selected and only a few - within the high pressure range - must still be selected. Danfoss (Pty) Ltd. and Parker (Pty) Ltd. will supply most of the components, pending availability of the units. Other companies to supply components include: Carel Controls South Africa (Pty) Ltd., Temprite USA and Heldon Products (Pty) Ltd. We are looking into the following supplier for the compressor: Bock Compressors.

3.3 System Layout

The system consists of the following components: one or two accumulators; capillary tube expansion device; compressor; evaporator; gas cooler/condenser; internal heat exchanger; temperature, pressure and mass flow sensors and shut-off and safety relief valves. The following figure shows the layout of the system.

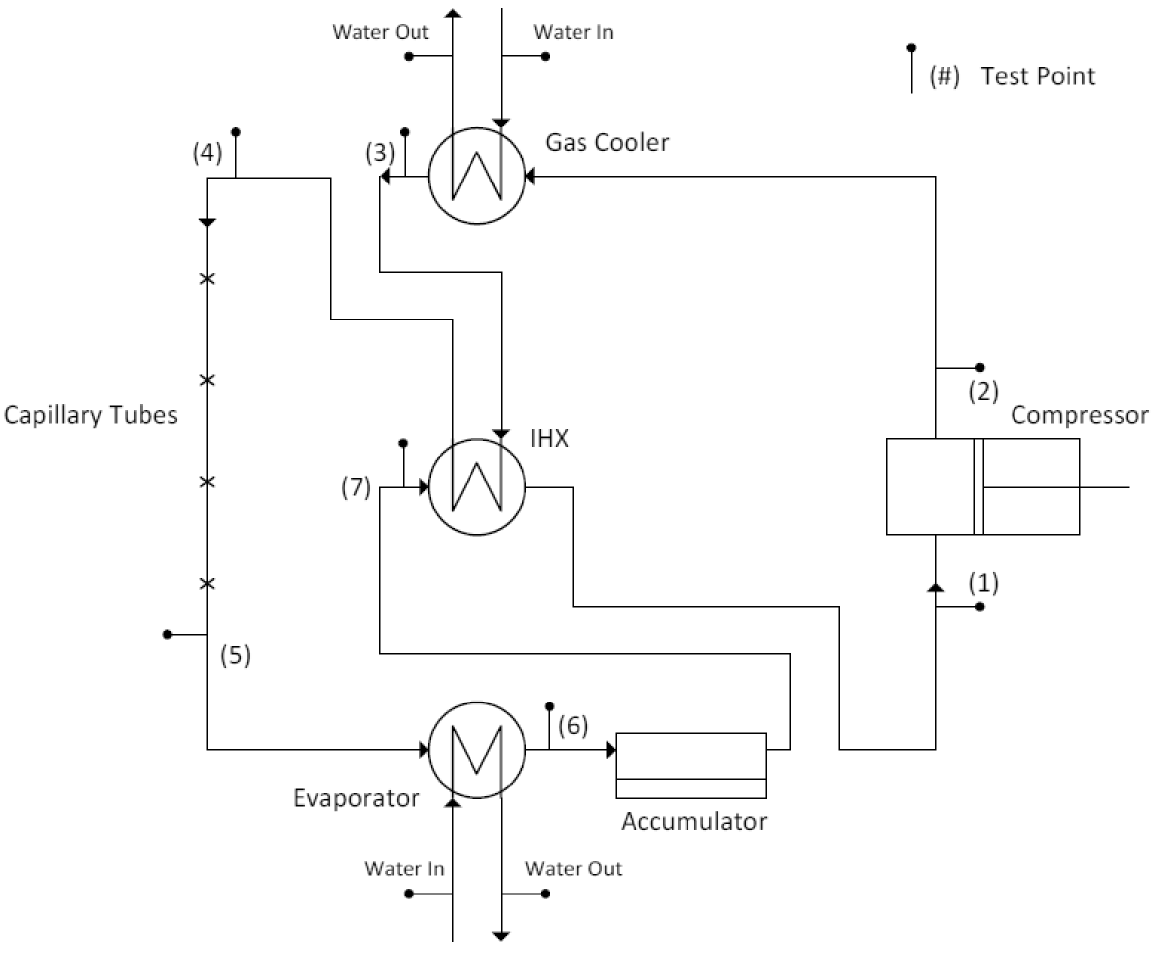

Figure 8 Typical CO_2 refrigeration system consisting of a compressor (comp), gas cooler (GC), capillary tube (cap), evaporator (e), accumulator and internal heat exchanger (IHX)

4. Discussion, Conclusion and Recommendation

The design of the system must still be finalized and the remaining components must still be sourced. The author is currently busy conversing with Danfoss, Parker and Bock to source the final components needed for the system. D Kriel will be contacted as soon as all the components have been selected, he will be in-charge of funding the equipment. After the components have all arrived, a frame/platform must be made to hold all the components in place and the piping must be bent and fitted between the individual components.

The system will then be used as basic tests are performed to determine if the system is working. Thorough and extensive tests/experiments will then follow until the author and her study leader is satisfied they have collected all the necessary data.

A simulation program must also still be written that will fully simulate the system under varying conditions. The simulation must be able to produce the same manner of data as the experiment.

Finally, a comparison will be made between the experimental data and the simulation data and conclusions will be drawn from that.

Nomenclature

C_f	Friction Coefficient
d	Diameter
h	Enthalpy
$\dot{m}$	Mass Flow Rate
N	Compressor Speed, rpm
p	Perimeter ,m
$\dot{Q}$	Heat Transfer Rate, W
R	Overall Heat Transfer Resistance, ℃/W
T	Temperature, ℃ or K
u	Velocity
V	Volume, m^2
x	Quality, %

Subscripts

c	Compressor
cap	Capillary Tube
ev	Evaporator
F	Friction
f	Friction
GC	Gas Cooler
h	Homogeneous
i	Inside/In
i	ith segment
is	Isentropic
lm	Log Mean
ℓo	Liquid Only
M	Momentum
o	Outside/Out

r	Refrigerant
s	Surface
v	Volume
w	Water
x	Cross-section

Greek Letters

η	Efficiency
μ	Viscosity
ϕ	Two-Phase Multiplier
ρ	Density, kg/m^3

Abbreviations

CFC	Chlorofluorocarbon
COP	Coefficient of Performance
LMTD	Log Mean Temperature Difference
IHX	Internal Heat Exchanger
HCFC	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbon
PFC	Perfluorocarbon

References

Adriansyah W, Purwanto TS & Suwono A. 2006. Improvement of Prototype of CO2 combined air-conditioning and tap water heating plant, *7th IIR Gustav Lorentzen Conference on Natural Working Fluids*, Trondheim, Norway

Agrawal N & Bhattacharyya S. 2008a. Optimized transcritical CO2 heat pumps: Performance comparison of capillary tubes against expansion valves, *International Journal of Refrigeration*, vol 31 pp 388-395

Agrawal N & Bhattacharyya S. 2008b. Homogeneous versus separated two phase flow models: Adiabatic capillary tube flow in a transcritical CO2 heat pump, *International Journal of Thermal Sciences*, vol 47 pp 1555-1562

Bellstedt M. 2010. [Personal Communication] June

Cabello R, Sanchez D, Llopis R & Torella E. 2008. Experimental evaluation of the energy efficiency of a CO2 refrigeration plant working in transcritical conditions, *Applied Thermal Engineering*, vol 28 pp 1596-1604

Chen Y & Gu J. 2005. The optimum high pressure for CO2 transcritical refrigeration systems with internal heat exchangers, *International Journal of Refrigeration*, vol 28 pp 1238-1249

Danfoss (Pty) Ltd. 2008. Transcritical Refrigeration Systems with Carbon Dioxide (CO2), *Danfoss, Refrigeration and Air-Conditioning Division*, [Online] Available: www.danfoss.com

Daqing Li & Groll EA. 2005. Transcritical CO2 refrigeration cycle with ejector-expansion device, *International Journal of Refrigeration* vol 28 pp 766-773

Heap RD. 2001. Refrigeration and air conditioning – the response to climate change, *Bulletin of the IIR* – No 2001-5

Kauf F. 1999. Determination of the optimum high pressure for transcritical CO2 refrigeration cycles, *International Journal of Refrigeration* vol 38 pp 325-330

Kriel D, MGC Industrial, Stellenbosch, *Personal Communication*, 2010 January

Liao SM, Zhao TS & Jakobsen A. 2000. A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles, *Applied Thermal Engineering* vol 20 pp 831-841

Madsen KB, Poulsen CS & Wiesenfarth M. 2005. Study of capillary tubes in a transcritical CO2 refrigeration system, *International Journal of Refrigeration* vol 28 pp 1212-1218

Sarkar J. 2009. Cycle parameter optimization of vortex tube expansion transcritical CO2 system, *Department of Mechanical Engineering, Institute of Technology*, BHU

Sarker J, Bhattacharyya S & Gopal MR. 2006. Simulation of a transcritical CO2 heat pump cycle for simultaneous cooling and heating applications, *International Journal of Refrigeration* vol 29 pp 735-743

Span R & Wagner W. 2003. Equations of State for Technical Applications, *International Journal of Thermophysics* vol 24 no 1

Tao YB, He YL, Tao WQ & Wu ZG. 2010. Experimental study on the performance of CO2 residential air-conditioning system with an internal heat exchanger, *Energy Conversion and Management* vol 51 pp 64-70

Walley, PB. 1987. *Boiling Condensation and Gas-Liquid Flow*, Claredon Press, Oxford

Wikipedia. 2010. *Global Warming* [Online] Available: http://en.wikipedia.org/wiki/Global_warming [06 July 2010]