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

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ABSTRACT

Compared with other refrigerants CO_2 has a number of distinct advantages: it has the lowest global warming potential of all known substances; it cannot be regarded as toxic or hazardous; and at high pressures and temperatures, because of its relatively high density, not only is it more efficient as a working fluid, but much smaller physically sized components can be used. These properties make CO_2 a working fluid of choice for process operations seeking a more efficient, sustainable and environmental-friendly working fluid.

Although CO_2 has been known as a superior working fluid for more than a century it has never been commercialised and as such is more expensive than existing, mass produced components and systems. However, with cost savings in mind, a CO_2 system has been designed using existing, available controls and components, with the additional feature of being able to easily investigate the use of a conventional Capillary Tube as a throttling device.

In this paper, details of the as-designed system will be presented and discussed. The assumptions made and the manner in which the differential equations are derived, in order to theoretically simulate the system, are given. The results obtained by numerically solving the set of equations are given and discussed.

Keywords: Sub- and transcritical CO_2 refrigeration systems, mathematical simulation

1. INTRODUCTION

1.1. Background of Carbon Dioxide (CO_2)

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 create 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)

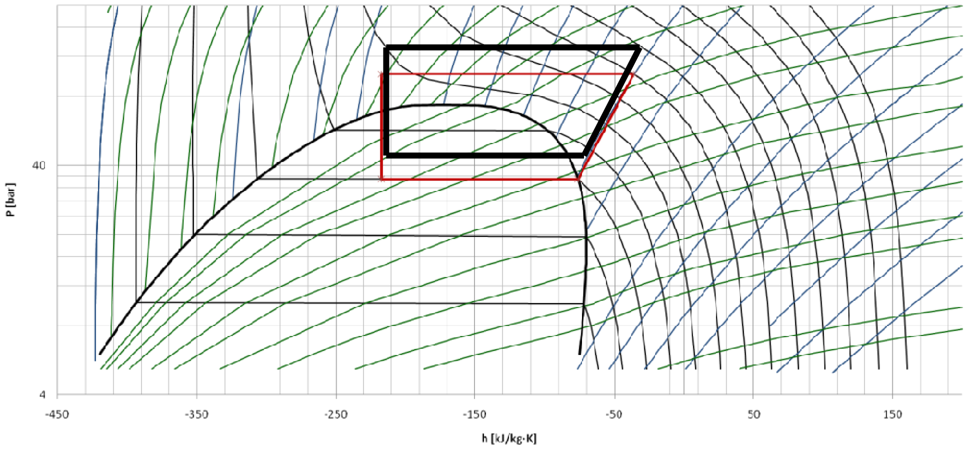

Figure 1. P-h Diagram of a typical CO_2 Transcritical System

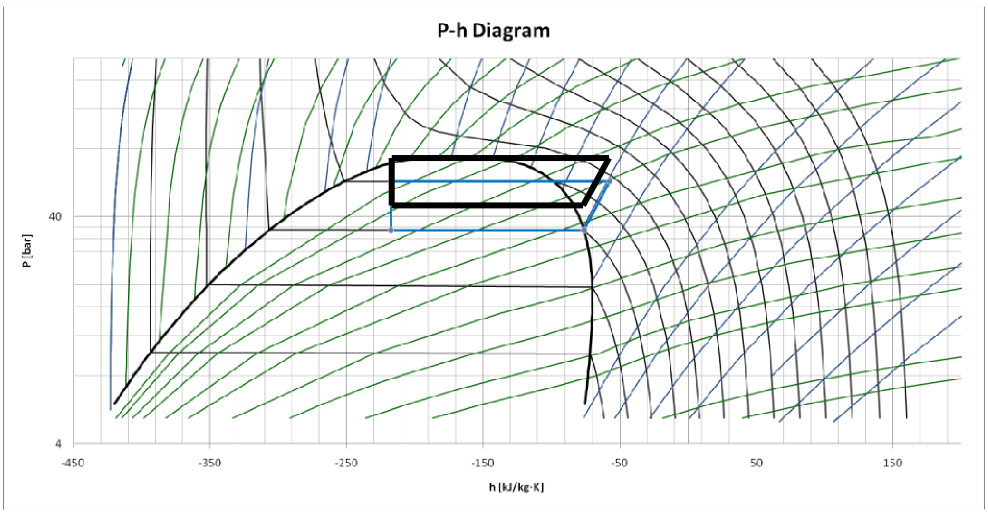

Figure 2. P-h Diagram of a Typical CO_2 Subcritical System

1.2. Motivation

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 make use of both the conventional expansion devices and a variety of Capillary Tubes. The ease of use and efficiency of the two types of expansion systems will be compared as the project nears completion.

1.3. 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. Two simulation models will be used: one for the design of the system and one to use as a comparison with the experimental. 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, which is dependent 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.

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.

The compressors used for CO_2 refrigeration systems are usually designed to run in either subcritical or transcritical regions due to the extreme difference in operating conditions. A Dorin TCS340 compressor will be used in the experiment, as it runs under transcritical conditions but also has some subcritical capabilities.

1.4. Theoretical Process

The primary goal of the project is to theoretically simulate a CO_2 refrigeration cycle. To achieve this objective, a transient simulation program need be completed that will emulate the start-up conditions of an actual system. The program will run on the principle of the three conservation laws: conservation of mass, conservation of motion and conservation of energy. Expected completion date of the transient simulation is early 2012. The simulation will provide its results in both graphical and numerical form.

1.5. Experimental Process

To aid in the design of the experimental system, a steady-state theoretical simulation has been completed in MS Excel. It is primarily used to help with the design of the experimental systems' heat exchangers and the Capillary Tubes. Using the design given by the steady-state simulation and other guidelines set by experts in the field of CO_2 refrigeration; a complete experimental system has been designed and is currently under construction in the Mechanical & Mechatronic Engineering workshop at Stellenbosch University. Completion date for the system is early 2012. Once the system is in working order, it will be tested under varying ambient conditions and measurements such as temperature and pressure will be stored in a data bank. These measurements will be compared to results obtained from the transient simulation. Furthermore, the accuracy of the steady-state simulation can be tested by comparing how well the experimental system runs – compared to what is expected in an ideal system – as used in the simulation environment.

2. THEORY

The theory behind the project covers an extensive literature survey and the mathematical models required for the two simulation programs. First, the literature survey is briefly discussed. This is followed by a description of the mathematical model, including assumptions made for each simulation.

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. Systems that are more complicated 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: a Capillary Tube; a vortex tube; an ejector-expansion device; 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 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 concluded 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 wherein 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 easily and readily used to determine the properties of CO_2 (Sarkar, 2009) (Cabello, et al., 2008) (MegaWatSoft, 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 applicable 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 starting basis for the mathematical model is a full set of conservation laws. From the conservation laws we can derive the equations needed for the steady-state model and for the transient model. The last element to consider is the 2-phase flow of the CO_2.

General assumptions made, for both the steady-state and the transient models, are as follows:

- A homogeneous 2-phase flow model;
- All the piping and fittings in the system are assumed to have negligible losses in both pressure and temperature;
- The Evaporator, Gas-Cooler and IHX are tube-in-tube heat exchangers;
- The Evaporator and the Gas-Cooler are water heated and cooled, respectively;
- No heat is lost through the piping to the environment;
- Friction is ignored throughout the system, apart from the Capillary Tube; and
- One-dimensional flow, i.e. $\dot{m} = \rho v A_x$.

For the one-dimensional flow assumption: $\dot{m}$ is mass flow rate, ρ is density, v is velocity and A_x is the area of the tube cross-section.

The theoretical simulations were 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 or by creating an iteration macro with VBA code. The first simulation in MS Excel is designed to solve steady-state systems with parallel- or counter-flow heat exchangers. The second simulation is designed to solve the start-up conditions of a refrigeration cycle. CO2Tables is a MS Excel add-in provided by MegaWatSoft Inc. that is used to determine the transport properties of the CO_2.

2.2.1. Conservation Laws

The conservation laws are adopted from Fluid Mechanics – Fundamentals and Applications (Cengel & Cimbala, 2006).

Conservation of mass

The conservation of mass states that the change in mass Δm - within a control volume - with respect to the change in time Δt is equal to all rate of mass flow into the volume $\Sigma \dot{m}_{\mathbf{in}}$ minus the mass flow rate out of the volume $\Sigma \dot{m}_{\mathbf{out}}$.

$$\frac{\Delta m_{CV}}{\Delta t} = \sum \dot{m}_{in} - \Sigma \dot{m}_{out} \tag{1}$$

Conservation of motion

The conservation of motion states that the change in momentum $\Delta(m\ v)$ - within a control volume - with respect to the change in time is equal to the momentum flux into the control volume $\left(\frac{\dot{m}^2}{\rho\ A_x}\right)_{in}$ minus the momentum flux out of the control volume $\left(\frac{\dot{m}^2}{\rho\ A_x}\right)_{out}$, taking into regard the effects of pressure P, gravity g and shear stress τ working on and against the control volume.

$$\frac{\Delta(m\ v)_{CV}}{\Delta t} = \left(\frac{\dot{m}^2}{\rho\ A_x}\right)_{in} - \left(\frac{\dot{m}^2}{\rho\ A_x}\right)_{out} - (P_{out} - P_{in})\ A_x - m\ g_z - \tau\ A_z \tag{2}$$

With the gravity, it is necessary to take into account the direction of the gravitational force. Thus, $g_z = g\sin(\theta)$. Figure 4 shows an example of the conservation of momentum applied to a control volume of the Capillary Tube.

Conservation of energy

The change in internal energy ΔU - within a control volume - with respect to the change in time Δt is equal to the rate of energy entering the control volume $(\dot{m}\ h)_{in}$, minus the rate of energy leaving the control volume $(\dot{m}\ h)_{out}$ and including the rate of heat transfer entering or leaving the volume $\dot{Q}_i$.

$$\frac{\Delta U_{CV}}{\Delta t} = (\dot{m}\ h)_{in} - (\dot{m}\ h)_{out} \pm \dot{Q}_i \tag{3}$$

Figure 3 shows examples of the conservation of energy implied on the Gas-Cooler, Capillary Tube and Evaporator respectively.

2.2.2. Steady-State Model

Under steady-state conditions, the system is no longer undergoing any changes with respect to time. Thus, all variables changing with respect to time is equal to zero.

For the conservation of mass,

$$\frac{\Delta m_{\mathrm{CV}}}{\Delta t} = 0 \tag{4}$$

Thus,

$$\Sigma \dot{m}_{i-\frac{1}{2}} = \Sigma \dot{m}_{i+\frac{1}{2}} \tag{5}$$

Where the subscripts $\left(i-\frac{1}{2}\right)$ and $\left(i+\frac{1}{2}\right)$ is "in" and "out", respectively.

Conservation of motion

$$\frac{\Delta (m\, v)_{\mathrm{CV}}}{\Delta t} = 0 \qquad (6$$

$$(P\, A_x)_{i+\frac{1}{2}} + \left(\frac{\dot{m}^2}{\rho\, A_x}\right)_{i+\frac{1}{2}} = \left(\frac{\dot{m}^2}{\rho\, A_x}\right)_{i-\frac{1}{2}} - (P\, A_x)_{i-\frac{1}{2}} - (m\, [g_z)]_{i-\frac{1}{2}} - (\tau\, A_z)_{i-\frac{1}{2}} \qquad (7$$

Iteration is required to determine the pressure exiting the control volume and the mass flow rate and density exiting the control volume.

Conservation of energy

$$\frac{\Delta U_{\mathrm{CV}}}{\Delta t} = 0 \tag{8}$$

$$\dot{\mathrm{Q}}_i = \pm\left((\dot{m}\, h)_{i-\frac{1}{2}} - (\dot{m}\, h)_{i+\frac{1}{2}}\right) \tag{9}$$

And the heat transfer rate is determined using the log-mean-temperature-difference,

$$\dot{Q}_i = \frac{\Delta T_{\mathrm{lm}\,i}}{R_i} \tag{10}$$

Where thermal resistance R_i is,

$$R_i = \frac{1}{h_{r,i}\ A_i} + \frac{1}{h_{w,i}\ A_o} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi\ \Delta z_i\ k} \tag{11}$$

Caused by the convective heat transfer of the refrigerant, the convective heat transfer of the water and the conductive heat transfer through the tube wall.

The log mean temperature difference $\Delta T_{lm,i}$ is,

$$\Delta T_{lm,i} = \frac{\left(T_{r,i-\frac{1}{2}} - T_{w,i-\frac{1}{2}}\right) - \left(T_{r,i+\frac{1}{2}} - T_{w,i+\frac{1}{2}}\right)}{\ln\left(\frac{\left(T_{r,i-\frac{1}{2}} - T_{w,i-\frac{1}{2}}\right)}{\left(T_{r,i+\frac{1}{2}} - T_{w,i+\frac{1}{2}}\right)}\right)} \tag{12}$$

2.2.3. Transient Model

Further assumptions:

- Quasi-equilibrium conditions in the Capillary Tube
- Up-wind-differencing

The conservation laws are updated to the following forms:

Conservation of mass

The conservation of mass is used to determine the mass at the new time step and for that specific control volume.

$$m_i^{t+\Delta t} = m_i^t + \Delta t\ \left(\dot{m}_{i-\frac{1}{2}} - \dot{m}_{i+\frac{1}{2}}\right)^t \tag{13}$$

Conservation of motion

The conservation of motion is used to determine the velocity v at the new time step for that specific control volume.

$$(m\,v)_i^{t+\Delta t} = (m\,v)_i^t \;+ \Delta t\left(\left(\frac{\dot{m}^2}{\rho\,A_x}\right)_{i-\frac{1}{2}} - \left(\frac{\dot{m}^2}{\rho\,A_x}\right)_{i+\frac{1}{2}}\right)^t - \Delta t\left(\left(P_{i+\frac{1}{2}} - P_{i-\frac{1}{2}}\right)A_x\right)^t - \Delta t\,(m_i\,g + \tau_i\,A_s)^t$$

With the rate of heat transfer $\dot{Q}_i$ is described as,

$$\dot{Q}_i = \frac{T_{\mathrm{in}} - T_{\mathrm{out}}}{R_i} \tag{15}$$

Conservation of energy becomes,

$$U_i^{t+\Delta t} = U_i^t + \Delta t\left((\dot{m}\,h)_{i-\frac{1}{2}} - (\dot{m}\,h)_{i+\frac{1}{2}} - \frac{T_{i-\frac{1}{2}} - T_{i+\frac{1}{2}}}{R_i}\right)^t \tag{16}$$

And specific internal energy is,

$$u_i^{t+\Delta t} = \frac{U_i^{t+\Delta t}}{m_i^{t+\Delta t}} \tag{17}$$

2.2.4. Homogeneous 2-Phase Flow Model

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.

$$\rho_{\mathrm{h}} = \frac{1}{\left(\frac{x}{\rho_{\mathrm{g}}}\right) + \left(\frac{1-x}{\rho_{\ell}}\right)} \tag{18}$$

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

$$\mu_{\mathrm{h}} = \frac{1}{\left(\frac{x}{\mu_{\mathrm{g}}}\right) + \left(\frac{1-x}{\mu_{\ell}}\right)} \tag{19}$$

Where homogeneous viscosity 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 Φ_{lo}^2 is as follows,

$$\phi_{lo}^2 = \left(\frac{C_{fh}}{C_{f\ell o}}\right) \cdot \left(\frac{\rho_\ell}{\rho_h}\right) \tag{20}$$

However, it can generally be simplified as,

$$\phi_{lo}^2 \approx \left(\frac{\rho_\ell}{\rho_h}\right) \tag{21}$$

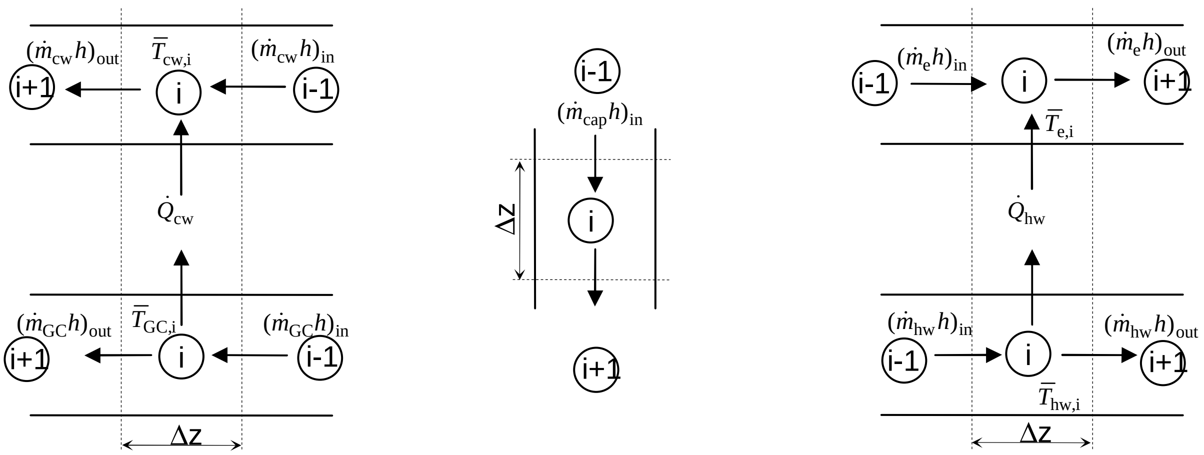

Figure 3. Gas-Cooler, Capillary Tube and Evaporator i'th conservation of energy control volumes of length Δz

Each of the components in the system can be divided into smaller control volumes of length Δz. Figure 3 shows the control volumes with information regarding the conservation of energy for the Gas-Cooler, Capillary Tube and Evaporator respectively.

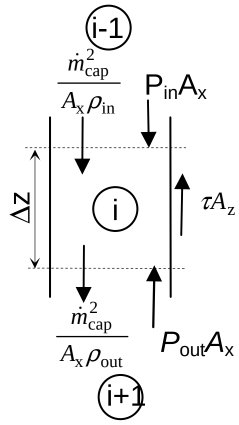

Figure 4. Capillary Tube i'th conservation of momentum control volume of length Δz

Figure 4 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.

3. EXPERIMENT

The design of the refrigeration system has been completed. This includes choosing off-the-shelf components from catalogues and designing parts such as the Evaporator, Gas-Cooler, IHX and Capillary Tubes. The system will consist of the basic CO_2 system and will include an IHX, a separation Accumulator, a mid-pressure receiver, a Capillary Tube exchange system, an auxiliary electronic expansion valve and Gas-Cooler pressure control valve.

Secondary components include:

- 10 PT1000-type Temperature Sensors
- 10 Pressure Transducers
- 2 Analogue Pressure Gauges
- 3 Sight Glasses
- 3 Flow Meters
- A Variable Speed Drive (VSD) for the Compressor
- 3 Shut-Off Valves (for 50 bar)
- 3 Shut-Off Valves (for 120 bar)
- 7 Three-Way Control Valves (for 50 bar)
- 5 Three-Way Control Valves (for 120 bar)
- 2 Safety Relief Valves (one for 50 bar and one for 120 bar).

The system will also include a monitoring unit, a frame/platform to hold the components in-place and the piping and fittings necessary to connect all the components.

3.1. General Arrangement

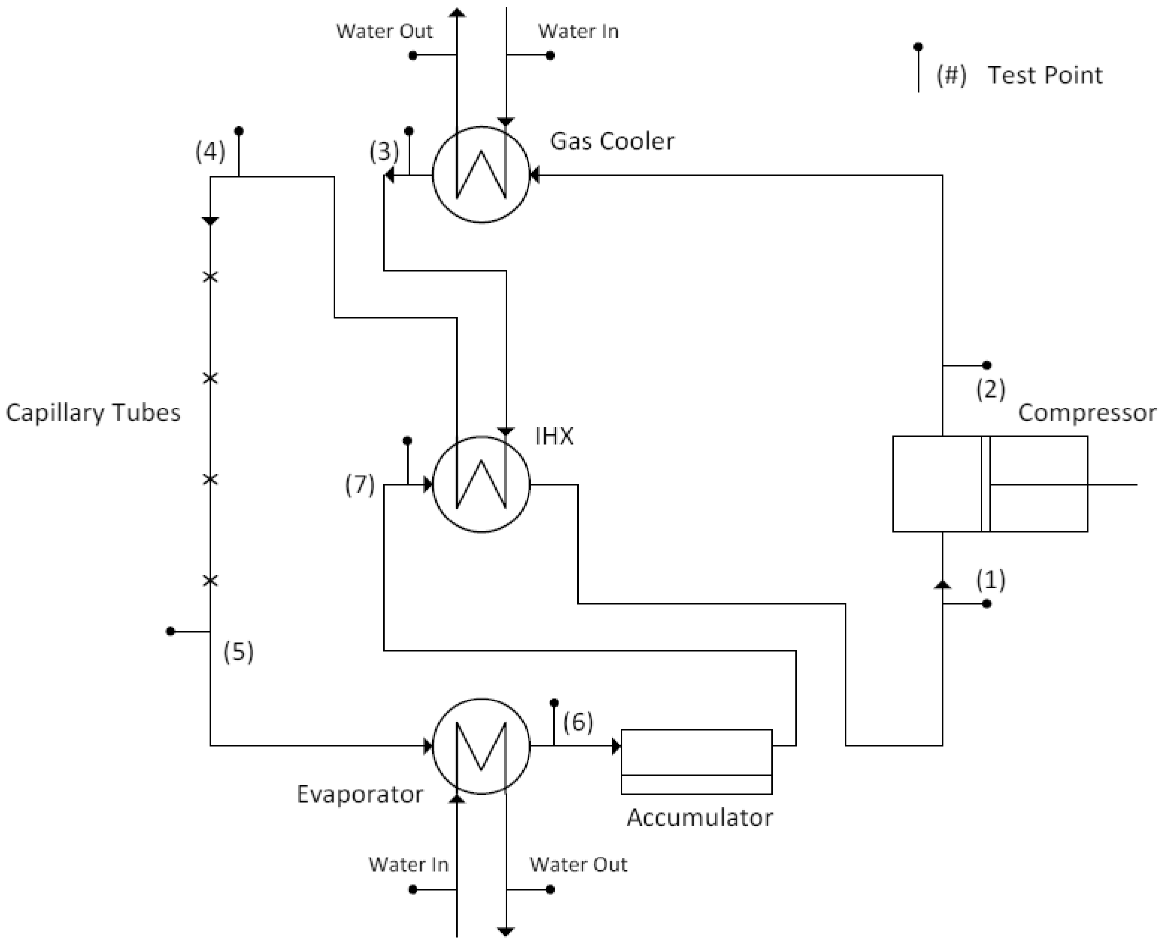

Figure 5. Typical CO2 refrigeration system consisting of a compressor (comp), Gas-Cooler (GC), Capillary Tube (cap), Evaporator (e), Accumulator and Internal Heat Exchanger (IHX)

Figure 5 shows a basic layout of the experimental system. The configuration includes the use of an IHX. It shows the Gas-Cooler and Evaporator as parallel flow heat exchangers while the IHX is set as counter flow. The components sponsored by Danfoss Group are not included in this basic model.

4. PROGRESS

4.1. Steady-State Simulation

The goal of the steady-state program is to aid in the design of the experimental setup. The program determines the sizes of the heat exchangers and the Capillary Tube that matches specific, chosen, operating conditions. The simulation program was completed in July 2011, with only minor GUI (Graphic User Interface) changes left to be made.

Opening the steady-state simulation, the user is first shown a welcome screen, as seen in Figure 6.

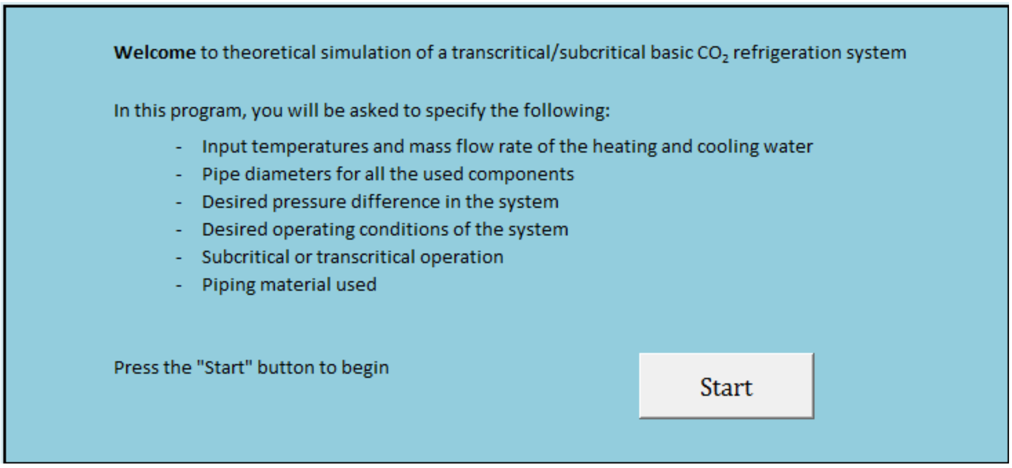

Figure 5. Welcome Screen to Steady-State Simulation

The user starts by choosing the operating conditions of the experimental system, the material specifications (stainless steel or copper as a material and tube diameters) and the direction of flow (parallel or counter flow). Figure 7 shows an example of the Gas-Cooler input screen.

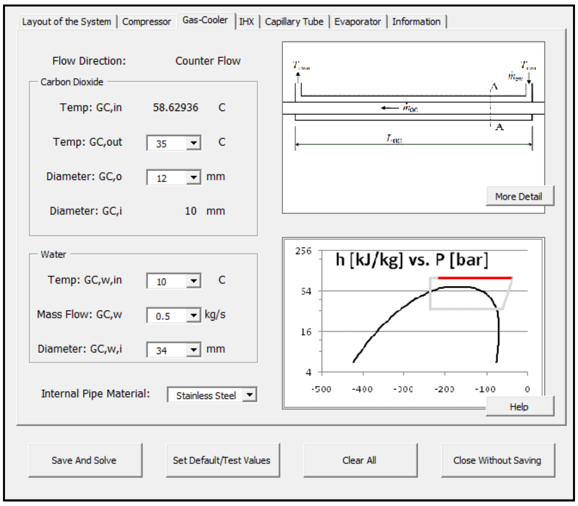

Figure 6. Steady-State Simulation - Input Screen Example

It takes a few minutes for the program to calculate all the necessary parameters, before it displays the results in the form of numerical values and graphical charts. Figure 8 shows an example of the Gas-Cooler results. The graph shows varying refrigerant temperature, water temperature and refrigerant enthalpy with respect to the number of control volumes in the Gas-Cooler – in this example there are 500 control volumes in the Gas-Cooler.

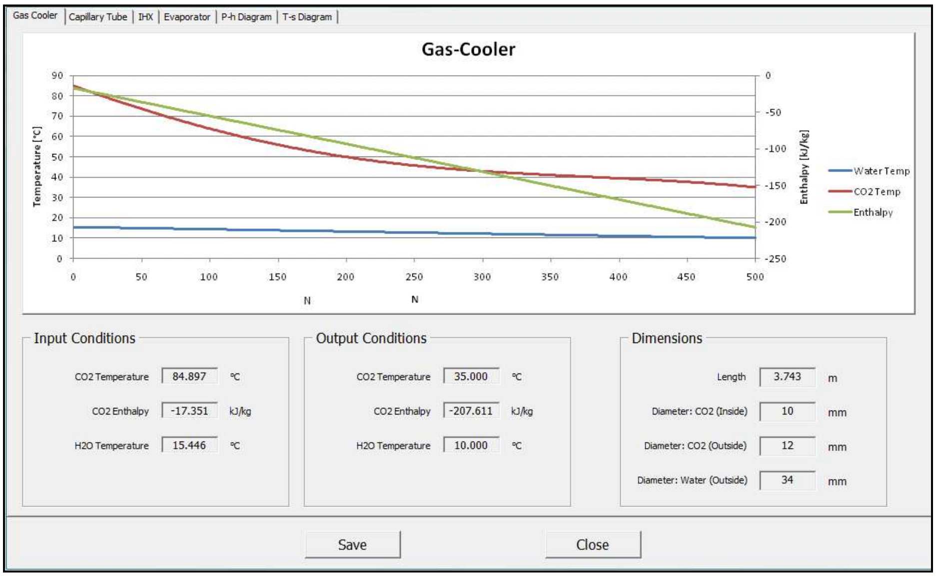

Figure 7. Steady-State Simulation - Results Example (Gas-Cooler)

The Ph-diagram and the Ts-Diagram for the system in the example is shown in Figures 9 and 10 respectively. It can clearly be seen in the graphs that the system was specified to run under transcritical conditions and that the IHX was chosen to be included in the system.

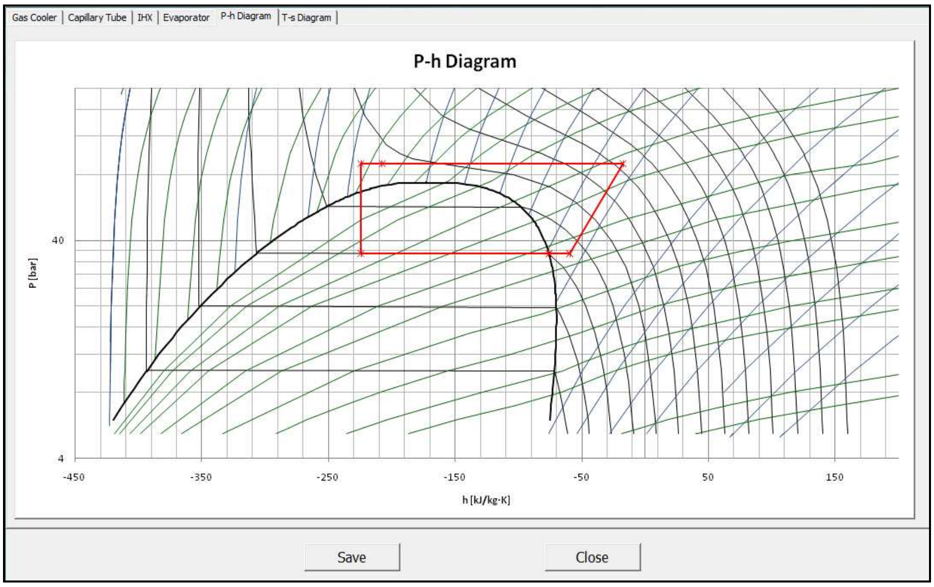

Figure 8. Steady-State Simulation - Results Example (Ph-Diagram)

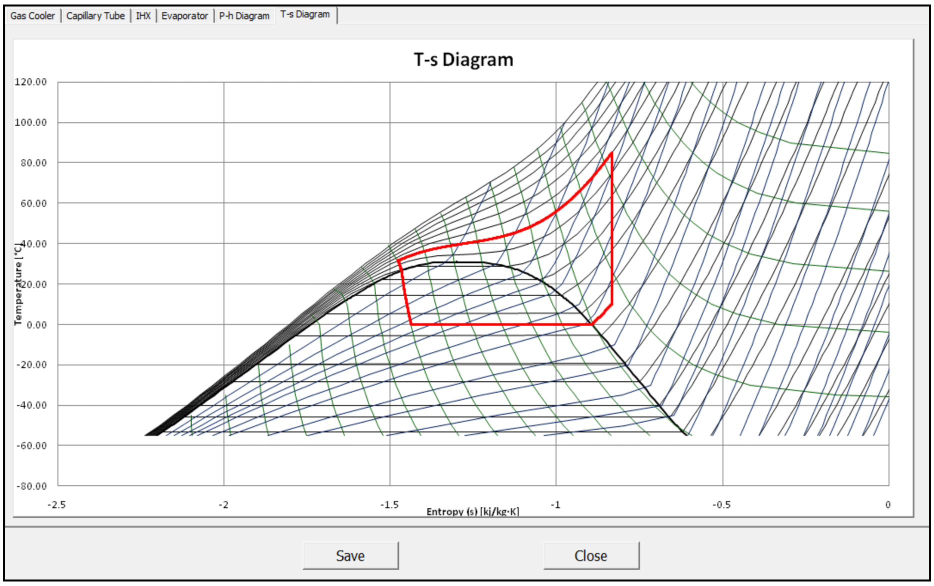

Figure 9. Steady-State Simulation - Results Example (Ts-Diagram)

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.

4.2. Transient Simulation

The goal of the transient simulation is to duplicate the start-up conditions of the experimental system. The user enters the dimensions of the physical system and ambient conditions at start-up. The simulation then determines the transient changes in the system, until it reaches steady-state conditions.

Although the program is yet to be completed, some results can already be obtained. The aim is to have a completely functioning transient simulation by the end of the year 2011 and a user friendly program by mid-2012.

4.3. Experimental Setup

Most of the components for the experimental system have already been ordered and delivered. Johannesburg Valves and Fittings Co (PTY) Ltd. have supplied the Swagelok fittings and valves. In addition, Inline Trade supplied the project with Unilok components. Danfoss Group has supplied most of the controllers and electronic valves free of charge, and will continue to supply further products for the system in the near future. Dorin S.p.A. supplies the compressor and all other components are currently under construction at the Mechanical and Mechatronic Engineering Workshop in Stellenbosch.

5. CONCLUSION

The design of the system has been finalized and a small portion of the remaining components must still be purchased. The frame/platform to hold all the components have been manufactured and is ready to hold the rest of the components for the system. Once the experimental system is complete, it will be used for basic tests to determine if the system is working correctly. 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 steady-state simulation program has been completed and used as an aid for the design process of the experimental system. The transient simulation has a expected completion date in early 2012. This transient simulation will be used as a comparison with the experimental systems' start-up sequence.

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

5.1.1. NOMENCLATURE

A	Area, m^2
d	Diameter, m
g	Gravitational Force, $\frac{\mathrm{m}}{\mathrm{s}^2}$
h	Enthalpy, $\mathrm{kJ/kg}$
h	Convective Heat Transfer Coefficient, $\frac{\mathrm{W}}{\mathrm{m}^2}\mathrm{K}$
m	Mass, kg
$\dot{m}$	Mass Flow Rate, $\mathrm{kg/s}$
P	Pressure. bar
$\dot{Q}$	Heat Transfer Rate, W
R	Overall Heat Transfer Resistance, $°\mathrm{C/W}$
s	Entropy, $\mathrm{J/K}$
T	Temperature, $°\mathrm{C}$ or K
t	Time, s
U	Internal Energy, kJ
u	Specific Internal Energy, $\mathrm{kJ/kg}$
v	Velocity, $\mathrm{m/s}$
x	Quality, %
z	Control Volume Length, m

Subscripts

g	Gas
h	Homogeneous
i	Inside/In
i	ith segment
lm	Log Mean
ℓ	Liquid
ℓo	Liquid Only
o	Outside/Out
r	Refrigerant
w	Water
x	Cross-section
z	z-direction

Greek Letters

μ	Viscosity, $\frac{kg}{m}s$
ϕ	Two-Phase Multiplier
ρ	Density, kg/m^3
τ	Shear Stress, $\frac{N}{m^2}$
θ	Angle, rad

Abbreviations

CFC	Chlorofluorocarbon
COP	Coefficient of Performance
GWP	Global Warming Potential
GUI	Graphical User Interface
LMTD	Log Mean Temperature Difference
MS	Microsoft
ODP	Ozone Depletion Potential
IHX	Internal Heat Exchanger
HCFC	Hydrochlorofluorocarbons
HFC	Hydrofluorocarbon
PFC	Perfluorocarbon

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