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A study on Carbon-dioxide Based Thermosyphon Circulation Loop with End Heat Exchangers

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Abstract*: CFD analysis of a Carbon-dioxide based supercritical thermosyphon circulation loop with end heat exchangers has been presented in the paper. A steady state loop 3D simulation model has been implemented. The simulation considers phenomena such as axial conduction in fluid, viscous Dissipation in the fluid as well as in a solid wall. Standard RNG k-ε model has been used for turbulent flow. Results are obtained for a fixed inlet temperature of cooling water in the cold heat exchanger (305K), the for a various inlet temperature of water in the hot heat exchanger within the limits of 323 K to 343 K. The loop inside of CO_2 maintained the constant operating pressure of 100 bar. Results show that due to the presence of local buoyancy effect, the presence of Bend, fluid parameters such as temperature and velocity vary in all three dimensions. Result also shows that even for a less temperature difference of 17.66 K, Reynolds number obtained a value of the order of 105 is efficient heat transfer.
Keywords: Natural circulation loops, System pressure, Carbon dioxide, Simulation.

I. INTRODUCTION

Thermosyphon circulation loop based secondary fluid systems are very reliable and simple due to the no moving components such as pumps. Water based thermosyphon circulation loop is mostly used in applications such as nuclear reactors, solar collectors, etc. In recent years, the carbon dioxide used as the secondary fluid in both thermosyphon circulation loop as well as in force circulation loops. The carbon dioxide properties are the environment friendliness properties. Works show that for low-temperature refrigeration and air conditioning applications, carbon dioxide based thermosyphons are very compact in comparison to other conventional working fluids [1]. To improve one-dimensional numerical analysis presented by bernier & baliga [2].Pilkwal et al [3] predicted the dynamic behaviour related to a single phase natural circulation loop by using one –dimensional code and 3 dimensional CFD code. Carbon dioxide based systems have been implemented for different application such as solar heaters [4,5],cryogenic refrigeration[6],electronics cooling system [7],new generation nuclear reactors ,refrigeration system [8],in chemical extraction [9], heat pump applications [10], in geothermal applications [11,12], etc. Recently the studies of Keller [13] and Welander [14], presented that loop flow rate said by the interplay between frictional and buoyancy forces. However, shows analyses and modeling of CO2 based thermosyphon circulation loops are relatively scarce in the literature. K. Kiran Kumar Ram Gopal [15] presented 1-dimensional[1-D] steady state examination of a rectangular thermosyphon circulation loops with end heat exchangers for depression temperature applications. Huang & zelaya [16] performed a theoretical investigation of the thermal performance of a rectangular natural circulation loop. Recently Zhang et al.[17] and Chen et al. [18] show work on the effects of heat transfer and the unstableness of supercritical CO2 flow in a 2-dimensional thermosyphon at a specified operating pressure of 90 bar operating over a more heat source temperature limit. However, their examination considered constant temperature heat sink and source (e.g. solar power plant, refrigeration and air conditioning systems and solar collectors). In addition, to account for bends etc., one may have to consider a 3-dimensional model for very good accuracy. Previous studies performed on molten salts mainly focused on measuring its physical properties [19] compatibility of molten salts with different structure materials & its heat transfer characteristic have also been studied. In the present work carry out computational fluid dynamics (CFD) analysis of a 3-dimensional, CO2 based thermosyphon circulation loop with end heat exchangers. Results obtained on the steady state behaviour of the loop at isobaric operating at various operating temperatures. The operating parameter range is chosen such that the loop fluid (CO2) exists as a supercritical single phase fluid.

II. PHYSICAL MODEL

Thermosyphon circulation loop schematic diagram as shown in figure 1. Thermosyphon circulation loop having a hot heat exchanger (HHE), a cold heat exchanger, an up comer and a runner. The circumference fluid is absorbed sensible heat from the external fluid heat is a loss in the hot heat exchanger and the circumference fluid is heat loss sensible heat from the external fluid heat is a gain in the cold heat exchanger. Circulation of the circumference fluid is maintained due to the buoyancy effect caused by cooling at the top side and heating at the bottom side.

The following geometric and material parameters are considered for the model:
Internal diameter of NCL (internal pipe), d = 38 mm
Internal diameter of CHE and HHE (external pipe), D = 73 mm
Length of CHE and HHE, L = 1,200 mm
Total width and height of NCL (L×H) = 1,540 mm × 1,245mm
Total length of the loop (Lt) = 545 0 mm
Insulated pipe length in horizontal pipe (L1) =170mm
Tube wall thickness = 2 mm
Material: Copper

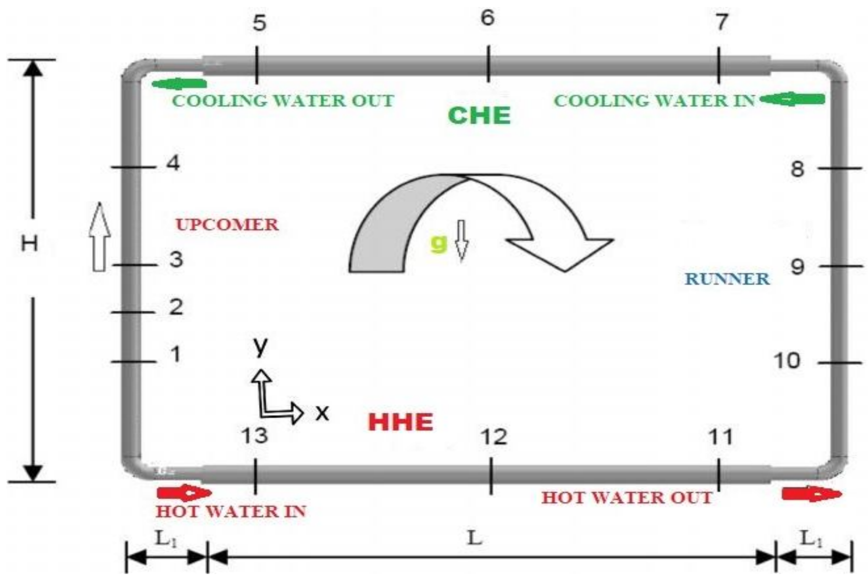

Figure 1: Schematic of Thermosyphon circulation loop with End Heat Exchangers (CHE and HHE).

III. MATHEMATICAL FORMULATION AND SOLUTION

The governing momentum, energy and mass equations are solved using the commercial CFD FLUENT. A 3-dimensional geometry was drawn in GAMBIT 2.4.6 and steady-state simulation was carried out employing ANSYS, where the implicit-coupled finite-volume method is used to discretize the governing equations. The pressure -implicit with the splitting of operators (PISO) algorithm is used to solve the coupling model between Pressure and velocity. The energy and momentum terms in the governing equations are iterated with a second-order upwind scheme that uses the upstream values and gradients to compute the control volume face values. The PRESTO (Pressure Staggering Option) scheme is used to discretize the pressure term. On the pipe walls, the no-slip boundary condition is used. Turbulent models for the supercritical fluid are less developed and still under intense study [20]. Therefore, in the present simulation a general Renormalization Group (RNG) k –ε model is selected as the first step to introduce the expression of turbulent effect. This method has also been used successfully in previous studies on supercritical CO2 turbulent flow yielding more accurate results [21]. Axial conduction and viscous dissipation in the fluid are considered. Axial conduction along the tube wall is also taken into account. Convergence criterion is set at 10-3 for the residuals of velocity, continuity, k and ε, and for energy term, the limitation is 10-6 for residuals of iteration. Mesh generation was implemented in the GAMBIT platform.

IV. CALCULATION OF THERMO–PHYSICAL PROPERTIES OF CO_2

The carbon dioxide is used inside of the loop is super critical, where the thermo physical properties variation is very more, it is essential to examine the properties by taking into account the temperature variation properly. However, as shown in the early year studies, due to very low variation in operating pressure throughout the thermosyphon circulation loop the effect of variation of pressure on the properties of single phase carbon dioxide is not expected to be significant [15, 17]. For example, results from the present study show that the maximum variation CO2 pressure 7000 Pa at an operating pressure of 100 bar (0.07% of operating pressure). Hence, for a given operating pressure, the properties of CO2 at any distance in the loop are examined at the constant operating temperature and pressure. The appropriate properties of carbon dioxide come from the NIST Standard Reference Database [22].Properties of CO2 for the maintaining temperature range are added to the fluent material properties list.

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V. RESULTS AND DISCUSSIONS

In this work, water is used as the external fluid in both the hot heat exchanger and cold heat exchanger. The incoming temperature of water in cold heat exchanger is kept constant at 305K and water incoming temperature in the hot heat exchanger is range from 323 K to 343 K in the stage of 10 K. Results are presented for the operating pressures of 100 bar of CO_2.Operating pressure of the system is defined at the centre of the hot heat exchanger. Mass flow rate of water in the hot heat exchanger as well as in cold heat exchanger is kept constant at a value of 0.3 kg/s.

A. Variation of temperature and pressure throughout the loop

Figures 3 and 4 shows the variation of total pressure difference of CO_2 through the loop and local average temperature for different water incoming temperatures of the hot heat exchanger. Results as show that for this loop, flow reversal phenomenon occurs at a hot fluid incoming temperature of 343 K as shown in figures. Due to this, at this temperature, the roles of upcoming and runner are reversed. Due to flow reversal, the hot heat exchanger and cold heat exchanger become parallel flow heat exchangers, since the flow direction of external fluid is the same for all conditions. Due to the assumption of the adiabatic condition in the upcoming and runner, the variation in temperature is almost negligible in these parts, whereas variation can be seen in the hot heat exchanger and cold heat exchanger due to heat transfer with the external fluid. The maximum variation of pressure in a loop is 7000 Pa which is equal to 0.07% of operating pressure.

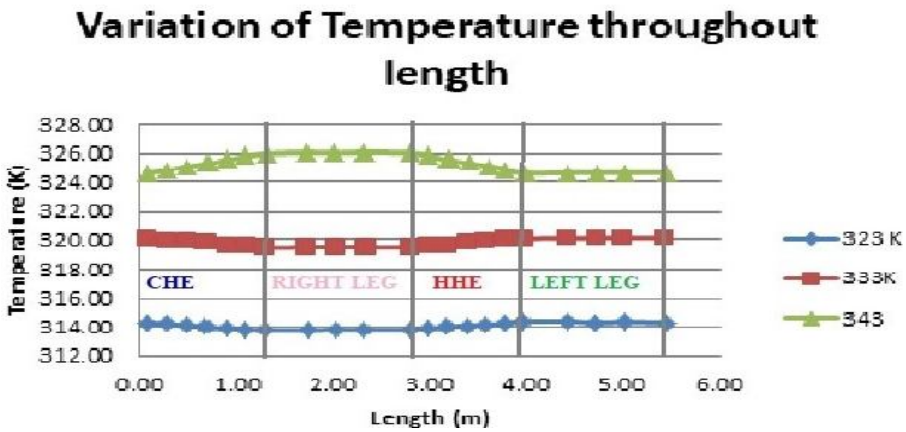

Figure 2: Variation of Temperature Throughout the Loop Length

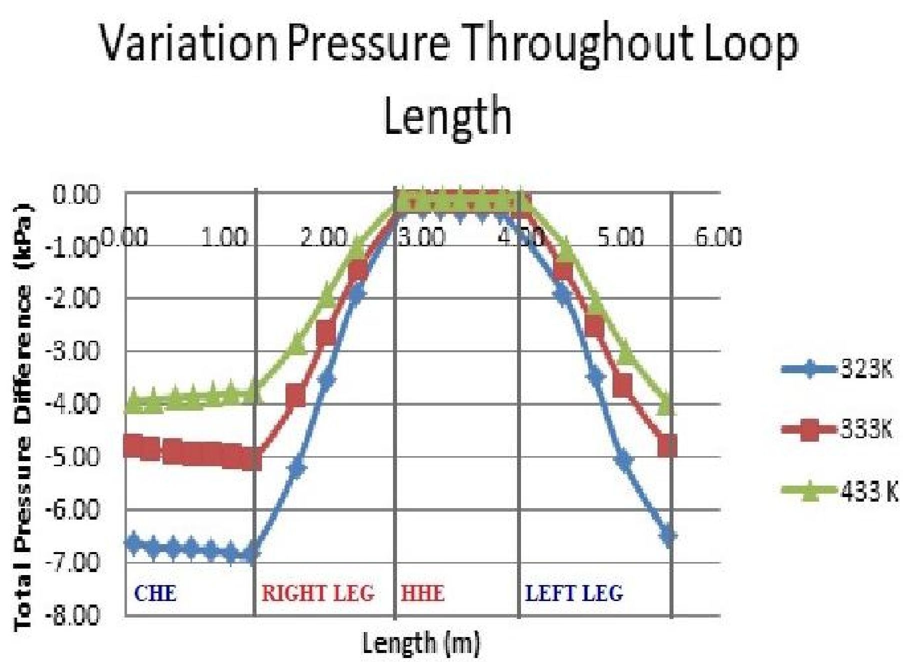

Figure 3: Variation of pressure throughout the Loop Length

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B. Temperature and velocity contour plots at HHE water inlet temperature of 333 K

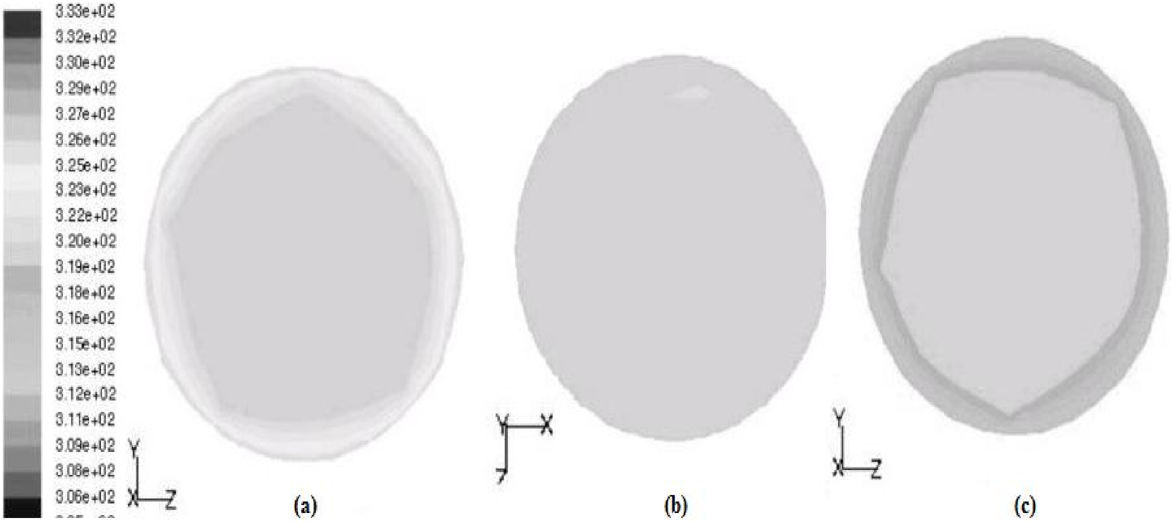

Figure 4: Temperature Contour Plot for HHE Water Inlet Temperature of 333 K at (a) HHE and (b) Left Leg (c) CHE

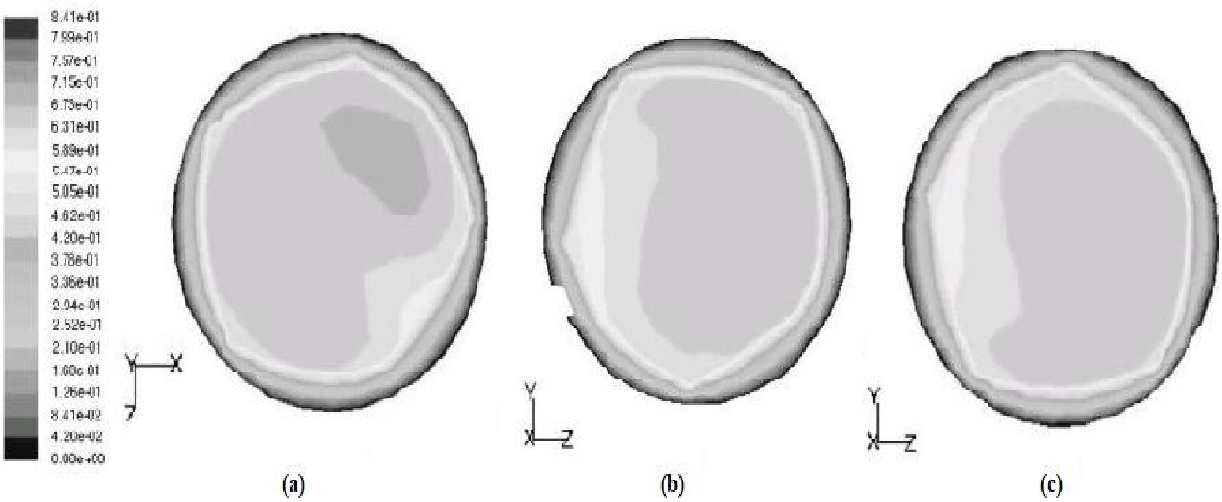

Figure 5: Velocity Contour Plot for HHE Water Inlet Temperature of 333 K at (a) Left Leg (b) CHE(c) HHE

Figures 4 and 5 show the temperature and velocity contour plot for hot heat exchanger water incoming temperature of 333 K at the centre of the hot heat exchanger, cold heat exchanger and left leg section. Figures show clearly velocity variation and the non-uniform temperature inside the tube at a given cross section. The temperature at the bottom part of the cross section of the hot heat exchanger and the cold heat exchanger is lower than that at the upper part due to local buoyancy effect. In the upcoming, the temperature is almost uniform over the cross section as there is no heat transfer from the wall. Since the temperature variation has a bearing on the stability of the loop, it is essential to capture it in order to understand the loop behavior. This evidently justifies the use of a 3-dimensional model.

C. Effect of HHX water inlet temperature

Figure 6 shows the variation of loop heat transfer rate (Q), Reynolds number (Re) and CO_2 mass flow rate (m) with hot heat exchanger water incoming temperature. Result observed that as water incoming temperature of hot heat exchanger increases Reynolds number and the heat transfer rate increase whereas mass flow rate increases initially and then start reducing. This could be attributed to the fact that at constant pressure, with a density of the fluid decrease, the temperature of the fluid increases and coefficient of thermal expansion decrease with temperature and near the critical point.

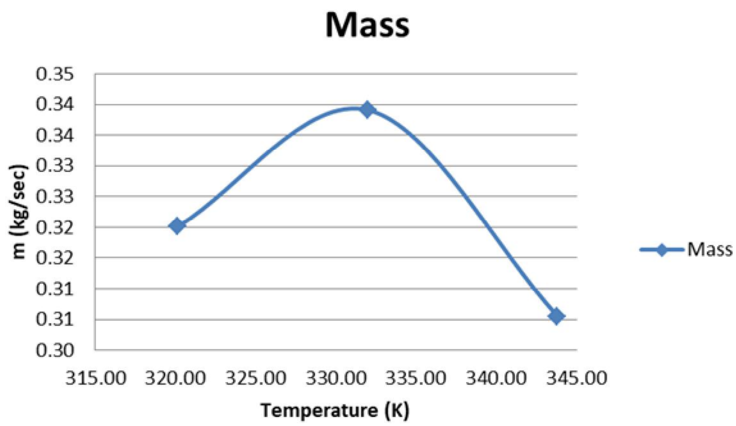

Figure.6. Variation of Mass Flow Rate of CO2 with HHE

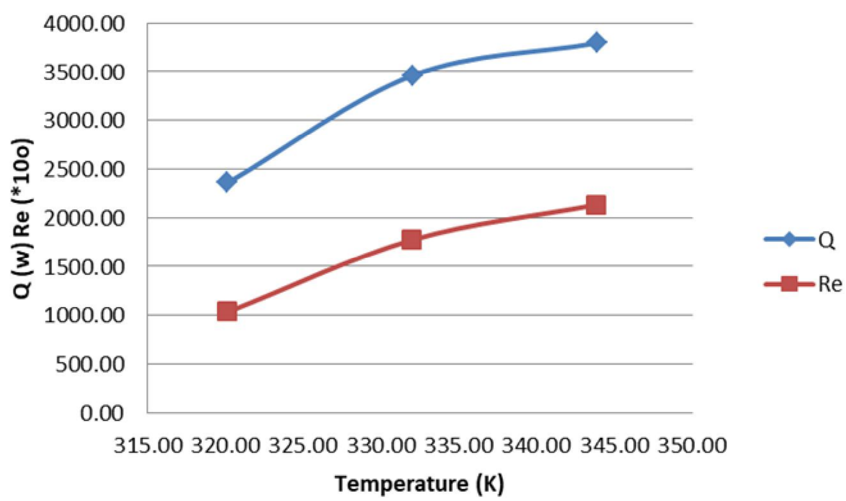

Figure .7.Variation of Heat Transfer Rate and Reynolds Number of CO2 with HHX Water Inlet Temperature

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