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Review Article

CO_2 Based Natural Circulation Loops for Domestic Refrigerators

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Abstract

In the present study, 3-dimensional model of natural circulation loop (NCL) have been developed in design software Pro-e 4.0. The steady flow simulation of single phase CO2, for natural circulation loop (NCL) with end heat exchangers are analyzed in Fluid Flow (Fluent), a module in Ansys-Workbench 14.0. Results are obtained for various mass flow rates/velocities of air in control volume of cold heat exchanger, for a fixed heat flux from air in the hot heat exchanger. Results show that due to the presence of bends and strong buoyancy effects, local velocity and temperature vary in all three dimensions.
Keywords: Natural Circulation Loop; Buoyancy effects; Heat transfer; Carbon dioxide.

INTRODUCTION

Natural circulation loops are flow systems heated from bottom and cooled from top, such that the heat sink is higher than the heat source. The specific configuration creates a density gradient which generates the driving force. When both the heat source and sink conditions are maintained constant, steady state circulation is expected to be achieved, which can continue indefinitely if the integrity of the loop is maintained. Because of the growing popularity of carbon dioxide (CO_2) as a secondary fluid in recent years, it has been witnessed in both forced as well as natural circulation loops. However, in forced circulation, there is a central component to pressurize the fluid and thereby to externally impart enough driving energy for circulation.

The natural tendency of heat is to flow from high temperature to low temperature reservoir. But refrigeration is the process of transferring heat from low temperature reservoir to high temperature reservoir. This is against the natural flow of heat hence external power is required to enable this transfer. Larger the size of refrigerator, more the food items can be kept in it. Accordingly, the amount of heat that is to be removed from the refrigerator to maintain the desired temperature also increases.

In single door refrigerators, since, the freezer section is maintained at temperature below 0^oC and the temperature inside the refrigerator compartment is maintained below the atmospheric temperature, the heat flows naturally from the atmosphere into the refrigerator compartment through the air passages. The problem with this type of system is the loss of cooling from freezer when the door of refrigerator compartment will be open each time.

This problem of single door refrigerators encouraged the design of two-door refrigerators. Since, there are separate doors for freezer and refrigerator cabin; hence freezer compartment now offers a better cooling and higher efficiency. However, the problem with this kind of system is the use of same air for circulation in both the cabins. Hence, if there is food with some bad smell in the refrigerator cabin, that will also go to the freezer cabin. Also, because the size of two door refrigerators is large, the power requirement is also large and now-a-days automatic defrosting element is also installed in it which consumes more power. The system for this kind is very complex too. Thus, there must be a system which can eliminate/or reduce these problems.

In present scenario, environmental concern has become a significant motivating factor in design and development of any domestic or industrial products. Consumers have become increasingly aware of the environment and the need to try and preserve the world. Since a decade, one specific area of concern has been developed about the effects of refrigerants on ozone layer depletion and more recently on global warming. The Kyoto Protocol has caused environmental focus to grow beyond ozone depletion to include global warming and equivalent emission of carbon dioxide into the atmosphere. The idea of monitoring global warming has brought about a new concept of considering not only the ozone depleting nature but also the global warming potential of products throughout their lifecycle. This had lead many researchers to concentrate on the development of new and improved refrigerants to minimize the harmful effect they have on the environment, while others looked at existing substances like hydro-fluorocarbons and naturally occurring gases to try and create a more efficient refrigeration system.

There are several fundamental requirements that any secondary refrigerant must satisfy. These are:

- Low viscosity
- High volumetric heat capacity
- High specific heat
- High thermal conductivity
- High density
- Less corrosive, chemically stable, non toxic etc.

Padalkar and Kadam [1] compared CO_2 to other fluids and found CO_2 offers very low viscosity and very high coefficient of thermal expansion. With these favorable properties one may expect CO_2 to perform well as secondary fluid also. In addition to these, CO_2 is an environmental friendly working fluid with zero ODP. Also it is non-toxic and acts as a flame retardant, thus perfectly safe for use.

Natural circulation loops can have various configurations. The classification has been made on the basis of state of working fluid, interaction with surroundings, shape, inventory, number of channels and body force field. Systems which are open to atmosphere are called open loops and loops which exchange only energy with the surrounding are called closed loops. Schematic view of the system is shown in Figure-1.

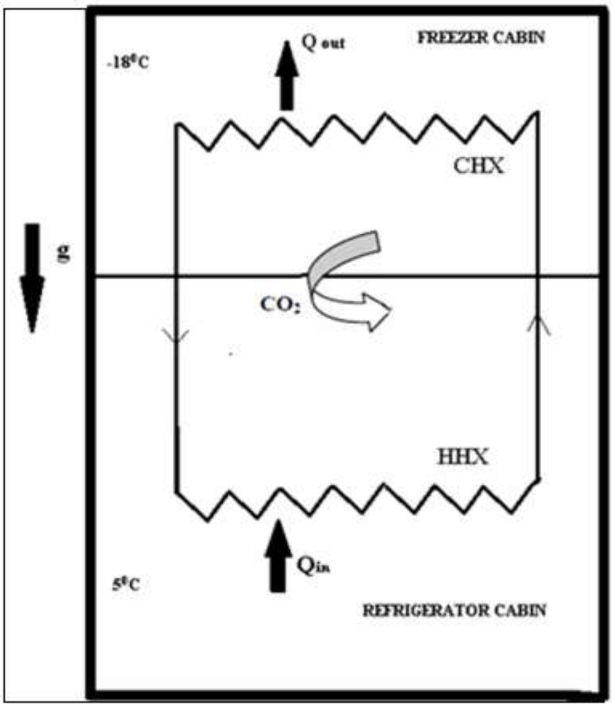

Fig-1: Schematic of NCL in refrigerator

Natural circulation systems essentially offer an assured and highly reliable heat transport mechanism, characterized by enhanced passive safety with respect to thermal failure. The absence of prime mover makes it a suitable option in a number of typical applications. For example, a two-phase NCL can be operated without bothering about the problem of cavitations and pump malfunctioning. Mechanical drives also introduce noise and vibration into the system, whereas NCL comes across as a serene relief. However, due to the strong coupling between momentum and thermal fields, it is not possible to predict desirable operating conditions in the natural circulation domain just by some thumb-rules. The system performance generally depends very strongly on each of the geometric and operating parameters and the slightest change in any may induce dramatic shift in operation. Hence intricate and comprehensive design calculations are mandatorily required to match the precise requirement of the concerned application.

Kumar and Gopal [2-4],-in their theoretical and experimental studies showed that CO_2 based NCLs are suitable for refrigeration and air conditioning applications. From their results, it was clear that CO_2 based NCLs can be used in a wide variety of applications over a fairly large range of capacities and temperatures. Also, it is possible to optimize the system dimensions and CO_2 inventory so that the loop heat transfer rate is maximized.

Yadav *et al*., [5-7],-developed a 3-dimensional computational fluid dynamics (CFD) model of NCLs to study their steady and transient states. Studies were carried out on subcritical as well as supercritical CO_2 loops. Several configurations (i.e., isothermal source and sink, end heat exchangers, heat input at constant heat flux) of the loop were considered. Results were obtained for different operating pressures, operating temperatures, tilt angles of the loop, heat inputs, mass flow rates of external fluid, etc. For the same source and sink temperatures, comparison of CO_2 with water as loop fluids shows that liquid CO_2 as well as supercritical CO_2 near pseudo-critical region exhibit very high heat transfer rates compared to water.

There are several types of problems related to natural circulation loops in which researchers are interested for many decades. Natural circulation loops (NCL) with different geometric configurations, steady state, transient and stability behaviour; single phase and two phase loops are some of the areas of interest. Due to the coupled nature of momentum and energy equations, theoretical analysis of natural circulation loop is relatively complex. So many studies available in literature deal with simple cases like rectangular and toroidal loops, loops with point heat source and sink, etc. Most of the studies are idealized by considering either known heat flux condition, or as convective heat transfer with known coefficient of heat transfer and wall temperature. Even though, some studies are carried out on two and three dimensional variation of flow parameters along the loop, majority of the studies are one dimensional.

General reviews of NCL are presented by Zvirin *et al*., [8] and Grief [9]. Their articles give an insight on categorization of loops and basic understanding of the phenomena. Keller [10] developed a one dimensional model to study the stability of a rectangular NCL with point heat source and heat sink. He showed that oscillations can occur in the absence of inertial effects, merely requiring an interplay between frictional and buoyancy forces. Welander [11] discussed the cause of instabilities in rectangular loops. Both Keller and Welander showed analytically that loop flow instabilities are predicted by dynamics of the system independent of fluid properties.

Yanagisawa *et al.,* [12] have carried out experimental studies on a natural circulation loop that uses carbon dioxide as secondary fluid. In their system, the primary vapour compression refrigeration system uses ammonia, while the secondary fluid, i.e., carbon dioxide undergoes phase change as it flows through the loop. They experimentally proved that natural circulating CO_2 secondary loop system is feasible.

Objective

Food stored at or below −18°C is safe indefinitely. Most household freezers maintain temperatures from -23 to -18 °C. Refrigerators generally do not achieve lower than −23 °C. Since the same coolant loop serves both compartments, so, lowering the freezer compartment temperature excessively causes difficulties in maintaining above-freezing temperature in the refrigerator compartment. In this study, a loop consisting of hot heat exchanger (HHX) and cold heat exchanger (CHX) for domestic refrigerator is developed such that it can transfer around 50 W of heat from refrigerator cabin to freezer cabin in order to reduce the complexity of previous system.

Numerical Model

All the parts of the loop are shown in Figure-1. The different views of refrigerator with NCL are show in Figure-2. Short descriptions of the parts are also listed in results and discussion. A natural circulation loop is modelled using a detailed three dimensional model. For single phase flow, the flow inside the loop consisting liquid CO_2 is assumed to be incompressible. Further, for the sake of simplicity, the Boussinesq approximation is used for the buoyancy term. The loop is placed vertically with respect to the ground. The loop fluid is heated sensibly by extracting heat from the external fluid (air) in HHX and is cooled sensibly by rejecting heat to the external fluid (air) in the CHX. Circulation of the loop fluid is maintained due to the buoyancy effect caused by heating at the bottom and cooling at the top. Geometric and material specifications of the model are given below (shown in Table).

Table-1: Double door refrigerator under consideration

External dimensions	630 x 700 x 1930 mm
Internal dimensions	520 x 560 x 1380 mm
Refrigerant	R134a
Sp. consumption of power	1.5 kWh/24 hour

The refrigerant of VCRS extracts this heat from the cold heat exchanger and is compressed in the compressor and is then condensed in the condenser by rejecting heat to an external heat sink. Since the flow in the NCL is buoyancy driven, the hot heat exchanger is kept at a lower level compared to the cold heat exchanger.

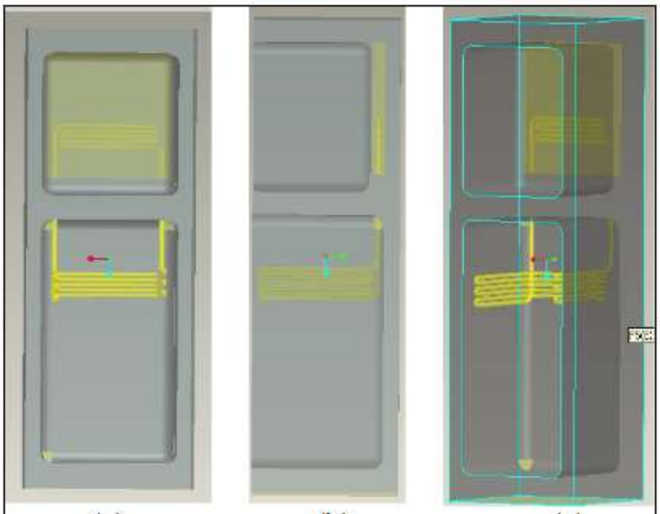

Fig-2: (a) Front view, (b) Side view, (c) Isometric view of refrigerator with NCL

System of Parameters

From Kumar *et al.*, for a given application with specified input parameters, there exists optimum values for the diameters of riser and downcomer. At these optimum values, the balanced mass flow rates are close to the maximum possible values and the temperature change across heat source/sink are close to the minimum possible values. Thus increasing the sizes of riser and downcomer beyond the optimum values does not offer any significant benefits. For low capacity domestic refrigerator (100 W), the optimum riser and downcomer diameters are found to be around 0.015 m and 0.01 m, respectively. For a 30 kW of cold storage capacity, the optimum riser and downcomer diameters are around 0.08 m and 0.07 m, respectively.

Table-2: Initial assumptions

Total length of loop	12m
Internal diameter of tube(0.005 to 0.025m)	0.010m
Thickness of tube, t	0.00135m
Outer diameter of tube (0.005 to 0.025m)	0.0127m
Height of loop	0.7m
Tube material	Copper

Table-3: Properties of CO_2 (at 40 bar, -5^{0}C)

Density, ρ	963.5 kg/m^3
Specific heat, c_p	2.34 kJ/kgK
Coeff of expansion, β	6.241 x 10^{-3} 1/K
Kinematic viscosity, ν_{co2}	0.1211 x 10^{-6} m^2/s
Dynamic viscosity, μ_{co2}	116.45 x 10^{-6} Pa.s
Prandtl number, Pr_{co2}	2.32
Thermal conductivity, k_{co2}	0.107 W/mK

Properties of air (at 1 bar and 5^{0}C) and (at 1 bar and -18^{0}C) are taken respectively.

Assumptions taken to develop the single phase NCL

- The internal (CO_2) & external (air) fluid streams are in single phase throughout the loop i.e. only sensible heat transfer takes place in the heat source and sink.
- Riser and downcomer are adiabatic.
- The system operates in steady state.
- Riser and downcomer are of uniform cross section.
- Fouling in both the heat exchangers is negligible.
- Wall material is isotropic with constant thermal conductivity.
- It is assumed that for both base case and system with NCL the condenser is designed such that it rejects required amount of heat. Due to this assumption, the sizing of the condenser is not considered in the present work.

Grid Information

Mesh generation has been implemented in the MESHING module of ANSYS 14.0. Meshing of a cross-section at heat exchanger for internal fluid is shown in Figure-3. Total number of nodes was 626383. Wall thickness is not considered for discretization, it is considered for the calculation of resistance to heat transfer only.

Solution Methodology

For the internal fluid, standard conservation equations are written as Eqs. (1-4). these equations with relevant boundary conditions are solved for steady state by the ANSYS-FLUENT.

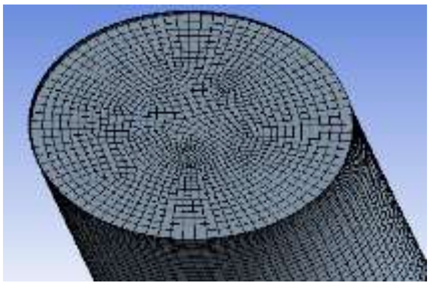

Fig-3: Meshing of cross section at CHX/HHX

The fluid velocity in the loop is generated from the density variation and this is related to the temperature differences within the fluid. In this analysis, the governing equations were written in axial and radial directions for the vertical legs of the loop. Flow directions in the legs are shown in Figure-4. The terms Tc and Th are the cold and hot temperatures of cold leg and hot leg respectively.

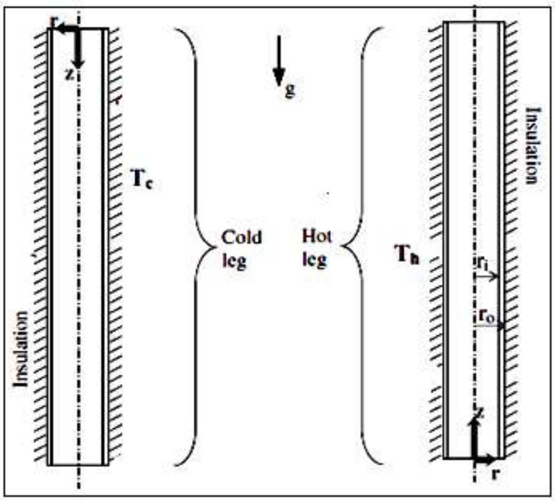

Figure-4: Sections of the vertical legs of the loop presented in 2-D model

The continuity equation (Eq.1), the momentum equations (Eqs. 2, 3) and the energy equation (Eq. 4) are used to describe the transport processes within the heat transfer sections of the loop. In Eq. 3, the axial component of the momentum equation, the reference temperature (T_0) is used to determine the fluid density (ρ_0). The flow direction modifier ξ is taken to be –1 for vertically upward flow and 1 for downward flow.

$$\frac{1}{r}\frac{\partial}{\partial r}(ru)+\frac{\partial v}{\partial z}=0 \qquad (1)$$

$$\rho\left(u\frac{\partial u}{\partial r}+v\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial r}+\mu\left(\frac{\partial^2 u}{\partial r^2}+\frac{1}{r}\frac{\partial u}{\partial r}+\frac{\partial^2 u}{\partial z^2}+\frac{u}{r^2}\right) \qquad (2)$$

$$\rho\left(u\frac{\partial v}{\partial r}+v\frac{\partial v}{\partial z}\right)=\xi g\rho_0\{1-\beta(T-T_0)\}-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 v}{\partial r^2}+\frac{1}{r}\frac{\partial v}{\partial r}+\frac{\partial^2 v}{\partial z^2}\right) \qquad (3)$$

$$\rho c_p \left(u\frac{\partial T}{\partial r} + v\frac{\partial T}{\partial z} \right) = k\left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} \right) \qquad (4)$$

The boundary conditions are required to complete the model.

Assume that:

- Fully developed laminar flow exists at flow inlet of vertical sections.
- No slip boundary condition is applied near the walls. Radial and axial components of the fluid velocity are zero at the inner surface of the loop wall.
- All external walls of riser & down-comer are perfectly insulated. The fluid and wall temperatures are homogeneous at inlet section of the vertical legs and it is assumed to be equal to outlet temperature of the other leg.
- The flow and temperature distributions are axially symmetric throughout the legs.
- The temperature gradient is zero at the outside surface of the insulated section.
- Mass flow rate of air at the inlet of CHX-control volume is known.
- Air inlet temperature at CHX is known.

The solution was converged considering SIMPLEC scheme for pressure-velocity coupling. For discretization, second order upwind scheme for momentum and energy with 0.7 & 0.8 was used as the under-relaxation factors for faster convergence. Also fixing the convergence criterion for residuals such as continuity, x,y,z-velocity and energy are 10^{-3}, 10^{-2} & 10^{-6} respectively.

RESULTS AND DISCUSSION

In this study, air is employed as the external fluid in CHX. Inlet temperature of air in CHX is kept constant at 255 K. Operating pressure of the system is defined at the centre of loop. Velocity of air in control volume of cold heat exchanger is varied from 0.9 m/s to 0.6 m/s with assumption that flow is from top to bottom wall of evaporator space and a constant heat flux of 190 W/m^2 is given to the HHX. Results showed, at velocity of 0.9 m/s the heat transfer in CHX was greater than the heat absorbed in HHX. At value of 0.75 m/s, the results were found close enough of desired value.

Because of the large size of refrigerator cabin, the numbers of cells/nodes generated were very high which were creating problem in simulation too. So, the control volume of refrigerator cabin was not taken under consideration, only CHX side control volume has been taken for simulation.

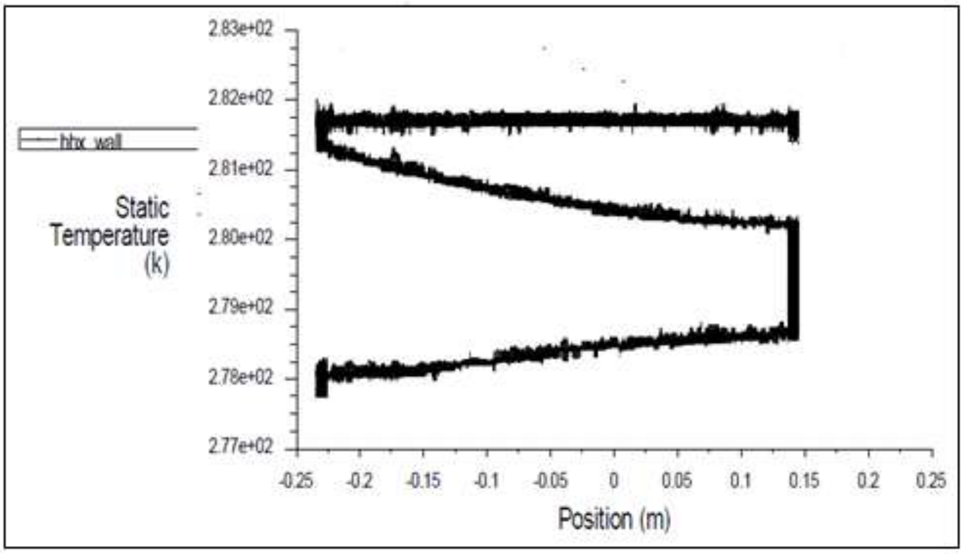

Fig-5: Variation in temperature of CO_2 in HHX along X-axis (front view)

Variation of temperature in HHX

From Figure-5, variations in CO_2 can be seen in HHX due to heat transfer from the heat-flux supplied along x-axis. Flow reversal is observed at cold fluid (air) inlet temperature of 255 K but the CHX remains cross-flow heat exchanger due to small variation in external fluid temperature. In this study, it is observed that external fluid (air) temperature difference in CHX is around 4 K.

Temperature and velocity contour plots

Figures 6 & 7 show the temperature (K) contour plot at CHX-inlet and CHX-outlet and at HHX-inlet and HHX-outlet respectively at air inlet temperature of 255 K in CHX control volume. Figure-8 shows the velocity vector at HHX inlet and outlet. The temperature and velocity contours clearly show the non-uniform temperature and velocity distribution inside the tube at given cross-section. Temperature at upper parts of CHX and HHX is higher than that at the lower parts due to local buoyancy effect.

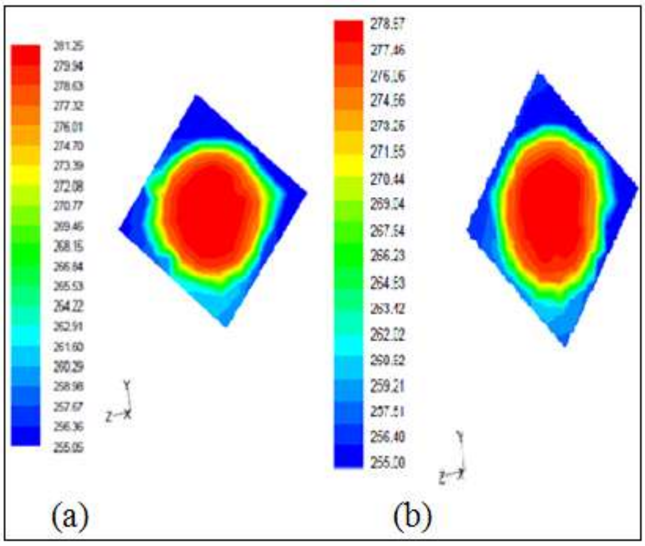

Fig-6: Temperature contour plots for (a) CHX inlet, (b) CHX-outlet

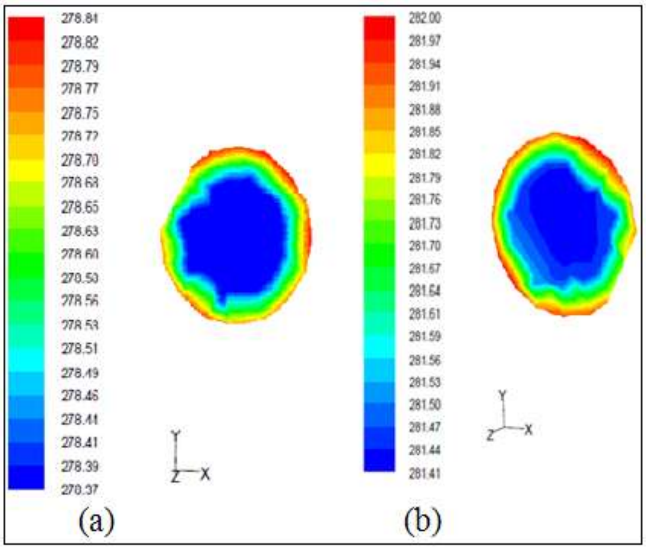

Fig-7: Temperature contour plots for (a) HHX-inlet, (b) HHX-outlet

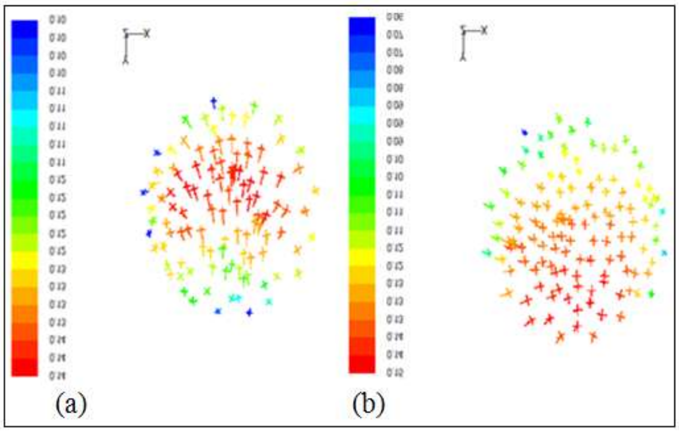

Fig-8: Velocity vector for (a) HHX-inlet, (b) HHX-outlet

Results at air velocity 0.9 m/s in CHX control volume is shown in Table-4.

Table-4: Results obtained for 0.9 m/s air velocity

Mass Flow Rate	***(kg/s)***
air_in	0.016537501
air_out	-0.016537501
Total Heat Transfer Rate	***(W)***
chx_wall	46.225459
hhx_wall	49.679325
ins_tube	0

The results at air velocity of 0.9 m/s show that the assumed velocity of air in evaporator space is large, so that the heat transfer from CHX is greater than HHX. So, further reduction in the velocity was made and found that at air velocity of 0.75 m/s the results are close enough of desired value. Results at air velocity 0.75 m/s in CHX control volume is shown in Table-5.

Table-5: Results obtained for 0.75 m/s air velocity

Mass Flow Rate	***(kg/s)***
air_in	0.012400002
air_out	-0.012400002
hhx-inlet (CO_2)	-0.0060953216
hhx-outlet (CO_2)	0.0060979728
Total Heat Transfer Rate	***(W)***
chx_wall-shadow	50.000818
hhx_wall	49.679325
ins_tube	0

CONCLUSION

In this study of application of NCL in refrigeration system, based on the theoretical studies, a CO_2 based natural circulation loop with end heat exchangers is modelled and analysed. It was found that the amount of heat rejected by CHX is almost equal to than the amount of heat taken by the HHX in refrigerator cabin with a difference of 0.241 W. Since, the value of air velocity is varied in CHX-CV from 0.7 to 0.9 m/s from top to bottom (which is not the actual case), with an interval of 0.05 m/s; it may be reduced to 0.01 m/s for better accuracy in results. Also, at velocity of 0.75 m/s, the mass flow rate of air comes out to be 0.0141 kg/s, which is expected under the capacity of evaporator fan and the mass flow rate of the carbon dioxide in the loop, is 0.00479kg/s.

Future Scope

- Experiments are to be designed with finer control of the operating parameters to see and understand the flow reversal phenomenon.
- Results with greater precision and accuracy can be obtained by carrying out studies with larger heat transfer rates and higher temperature differences on the external fluid side.
- Since stability is always an important issue with NCL systems, detailed stability analysis of CO_2 based NCLs with end heat exchangers should be carried out.
- A detailed economic analysis of the systems should be done to evaluate the economic viability of these systems.

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