Integration of gravity-fed evaporators in CO_2 based heat-pump chillers

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ABSTRACT

This study is carried out to model a gravity-fed evaporator for CO_2 based heat-pump chillers. The gravity-fed evaporator loop consists of a separator, a downcomer, a heat exchanger, and a riser. The working principle is that the evaporation of fluid in the heat exchanger gives rise to a density gradient in the loop. This density gradient generates the buoyancy force which drives the fluid by overcoming all the pressure drops in the loop. Appropriate equations are derived to estimate these pressure resistances. Finally, the balance between the buoyancy forces and pressure resistances dictates the flowrate in the loop. This procedure is implemented in Modelica to develop the simulation model for the gravity-fed evaporator. The loop dimensions are critical to the performance of the system. Hence it is essential to estimate the loop dimensions accurately. In this study, the simulations are carried out to predict the optimum dimensions to achieve the optimum circulation rate in the loop. It is expected that this study will be helpful in designing optimized gravity-fed evaporators for CO_2 based heat-pump chillers.

Keywords: Gravity-fed evaporator, Carbon Dioxide, Heat-pump, Modelica

1. INTRODUCTION

Gravity-fed evaporators are regarded as very simple and effective modification incorporated in a heat-pump unit to enhance the performance. As these evaporators are operated on flooded mode, it is possible to achieve better contact between liquid refrigerant and cooling surface of the heat exchanger. This leads to better heat transfer rates and hence helps in designing compact heat exchanger. For many years, gravity-fed evaporators have been used effectively in ammonia-based system. Several benefits can be obtained by incorporating such evaporators in heat-pump units. These systems eliminate the requirement of thermostatic expansion valve and operate independently irrespective of high-side pressure (Paliwoda, 1992a). In addition, the distribution of liquid refrigerant is better in these evaporators. Lorentzen (1968) presented different methods on how to implement gravity-fed evaporators as well as other types of flooded evaporators in refrigeration system. It was reported that the average efficiency of flooded evaporators is more than twice the efficiency of dry-expansion evaporators. Haukås (1986) presented a methodology and estimated optimum dimensions for gravity-fed evaporator implemented in a halocarbons-based system. It was reported that the ratio of evaporator coil length to inner diameter is the most critical parameter required to be optimized. Paliwoda (1992a) prepared an article to discuss how to develop a mathematical model for gravity-fed evaporator implemented in ammonia-based system. In this article too, it was reported how to optimize the dimensions for gravity-fed evaporator. These all studies show the benefits of implementing gravity-fed evaporator.

The present study is carried out to investigate the integration of gravity-fed evaporators in a CO_2 based heat-pump chiller. The purpose of this proposed CO_2 heat-pump chiller is to produce hot and cold water

simultaneously. Such systems could be implemented in different applications like fishing vessels, large kitchens, hotels etc. To integrate the gravity-fed evaporator in the proposed CO_2 system, a loop is designed which consists of a two-phase separator, a downcomer, the evaporator, and a riser. The dimensions of this loop play a significant role in the performance of the system. This study is performed to estimate the suitable dimensions of the gravity-fed evaporator loop. To fulfill this objective, a simulation model is developed in Modelica for the proposed CO_2 system. Based on the simulation results, the suitable dimensions are estimated for the gravity-fed evaporator loop. These results are presented in this paper.

2. MODELLING PROCEDURE

Figure 1 shows the schematic of the CO_2 heat-pump chiller with gravity-fed evaporator. To integrate the gravity-fed evaporator, a loop is designed which consists of a two-phase separator, a downcomer, the evaporator, and a riser. The liquid refrigerant (state pt. 6) separated in the separator is fed to the evaporator through the downcomer. Low vapor fraction (≈0.8) is maintained at the exit of the evaporator. The fluid leaving the evaporator (state pt. 7) then goes back to the separator through the riser. From the separator, the vapor fraction (state pt. 8) is sent to the compressor (Comp) through the internal heat exchanger (IHX). The compressed fluid (state pt. 2) is cooled down to low temperature in the gas cooler (GC-DWH) and then in the IHX. The fluid exiting the IHX (state pt. 4), returns to the separator through high pressure control valve (HPV). This completes the cycle. However, the refrigerant flowrate through the compressor and the refrigerant flowrate in the gravity-fed evaporator loop are different. In this paper, the refrigerant flowrate through the compressor is termed as primary flow '$\dot{m}_p$', while the refrigerant flowrate in the gravity-fed evaporator loop is termed as self-circulation flowrate '$\dot{m}_{s-c}$'.
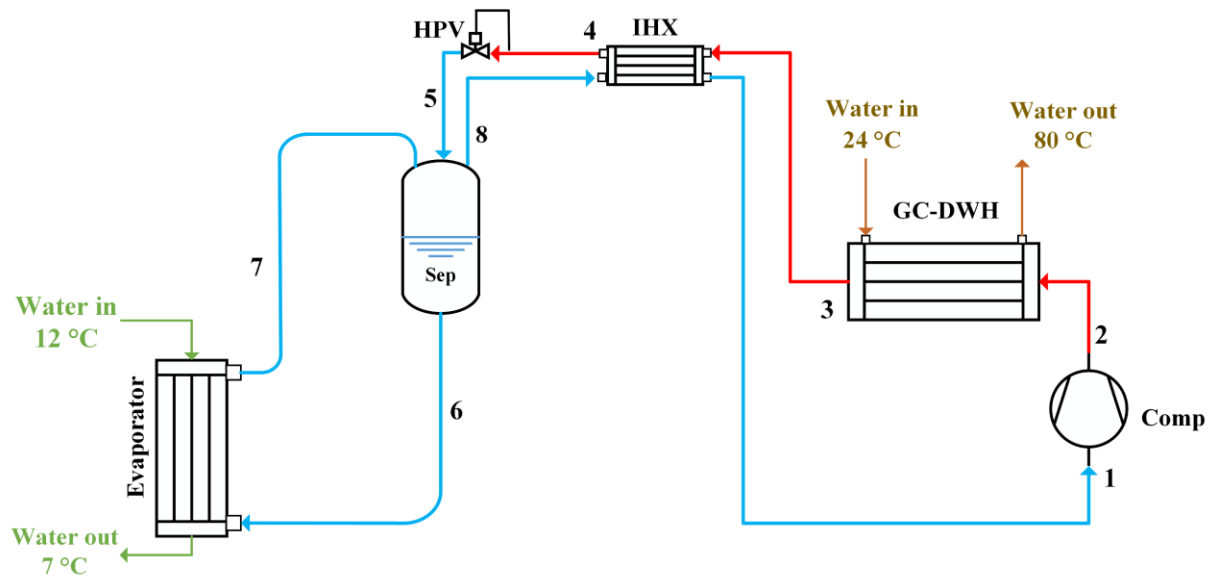

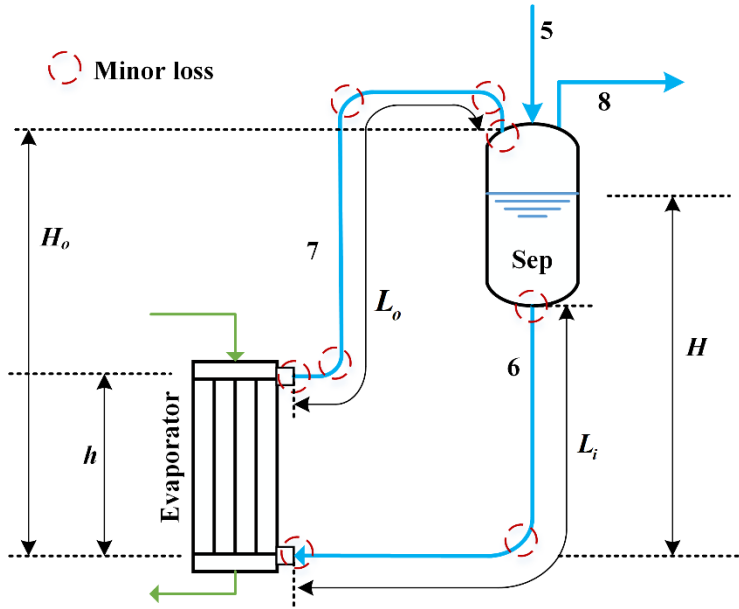Figure 1: Schematic of the CO_2 heat-pump chiller with gravity-fed evaporator

Figure 2: Schematic showing the dimensions of gravity-fed evaporator

Figure 2 shows the dimensions of the gravity-fed evaporator loop. The static height 'H' acts as a prime mover that maintains the desired flow by overcoming all the pressure resistances in the loop.

Thus, it can be written:

$$H\rho_l g - h\rho_m g - (H-h)\rho_o g = \Delta p_i + \sum \Delta p_i + \Delta p_{ev} + \sum \Delta p_{ev} + \Delta p_o + \sum \Delta p_o \quad (1)$$

The right-hand side of equation (1) contains all the pressure losses terms in the gravity-fed evaporator loop. These pressure losses are due to fluid friction (major loss) and due to change in flow area or flow direction (minor loss). To estimate these pressure losses, the equations are presented below.

In the downcomer, it is liquid refrigerant flowing inside. The frictional pressure drop in the downcomer is estimated from:

$$\Delta p_i = 0.241 \times \frac{\mu_l^{0.25}}{\rho_l} \times \dot{m}^{1.75} \times \frac{L_i}{d_i^{4.75}} \quad (2)$$

Equation (2) is derived from Darcy-Weisbach equation and Blasius formulae:

$$\Delta p = \rho g H_f = \rho g \left(\frac{f L V^2}{2 g d} \right) \quad (3)$$

$$f = 4 C_f = \frac{0.316}{Re^{1/4}} \quad (4)$$

The pressure drop due to minor losses in the downcomer is estimated from (Paliwoda, 1992b):

$$\sum \Delta p_i = \frac{\dot{G}^2}{2\rho_l} \sum \xi_j \quad (5)$$

where, 'ξ' is the coefficient of resistance for pipe components. Paliwoda (1992b) presents a table showing 'ξ' for different pipe components.

As low vapor fraction is maintained at the exit of evaporator, the flow through the riser is two-phase flow. The frictional pressure drop of two-phase flow is estimated from:

$$\Delta p_o = \beta_o \times \Delta p_g \tag{6}$$

where, 'β_o' is the two-phase multiplier and 'Δp_g' is the frictional pressure drop of single-phase vapor:

$$\Delta p_g = 0.241 \times \frac{\mu_g{}^{0.25}}{\rho_g} \times \dot{m}^{1.75} \times \frac{L_o}{d_o{}^{4.75}} \tag{7}$$

It is assumed that '$\dot{x}_o$' is the quality of the two-phase fluid at the exit of the evaporator and it remains constant during the flow in the riser. Pressure drop 'Δp_o' is estimated considering this assumption.

The two-phase flow multiplier 'β_o' corresponding to quality '$\dot{x}_o$' is estimated from (Paliwoda, 1992b):

$$\beta_o = [\vartheta + 2(1 - \vartheta) \times \dot{x}_o](1 - \dot{x}_o)^{0.333} + \dot{x}_o{}^{2.276} \tag{8}$$

where,

$$\vartheta = \frac{\Delta p_l}{\Delta p_g} = \frac{\rho_g}{\rho_l}\left(\frac{\mu_l}{\mu_g}\right)^{0.25} \tag{9}$$

In the riser, the pressure drop due to minor losses is estimated from:

$$\sum \Delta p_o = \frac{\dot{G}^2}{2\rho_g} \sum \xi_j \, \beta_{c,j} \tag{10}$$

The two-phase multiplier 'β_c' for each pipe component is estimated from:

$$\beta_c = [\vartheta + C(1 - \vartheta) \times \dot{x}_o](1 - \dot{x}_o)^{0.333} + \dot{x}_o{}^{2.276} \tag{11}$$

Paliwoda (1992b) presents a table showing the values for coefficient 'C' for different components.

To estimate the pressure drop in the evaporator, a quadratic equation dependent on the mass flow rate is used:

$$\Delta p_{ev} = b \times \dot{m}^2 \tag{12}$$

where b is:

$$b = \frac{\Delta p_n}{\dot{m}_n^2} \times \frac{l}{l_n} \tag{13}$$

These equations form the basis of the gravity-fed evaporator loop. To develop the model for this gravity-fed loop in object-oriented programming language Modelica [Modelica], component models are considered from TIL-library 3.9 [TLK-Thermo Gmbh]. However, these models from TLK library do not have the appropriate equations (presented above) to estimate the minor losses, major losses and the gravity term in the pressure drop model. Hence, these models are upgraded including all the necessary equations to capture the physics of self-circulating flow in the loop. These upgraded models are then used to develop

the gravity-fed evaporator loop. Finally, the gravity-fed evaporator model is integrated with the proposed CO_2 heat-pump chiller to develop the complete model in Modelica. Dymola 2021 is used as the modelling environment [Dassault Systems].

3. DIMENSIONS OF COMPONENTS

Table 1: Dimensions of components

Downcomer	
Internal diameter (mm)	14
Length (mm)	1500
Riser	
Internal diameter (mm)	14 - 28
Length (mm)	2150
Evaporator (plate heat exchanger)	
Number of plates	40
Plate length (mm)	420
Plate width (mm)	155
Pattern angle (°)	22.5
Wall thickness (mm)	0.5
Pattern amplitude (mm)	2.9
Pattern wavelength (mm)	6

4. RESULTS AND DISCUSSION

This section presents the results to show the implementation of gravity-fed evaprator loop in CO_2 based heat-pump. In a gravity-fed evaporator loop, low vapor fraction (≈0.8) is maintained at the exit of evaporator to ensure better contact between liquid refrigerant and cooling surface of heat exchanger. This helps in achieving high heat transfer coefficient of the refrigerant and hence gives better heat transfer rates between refrigerant and secondary fluid. Therefore, gravity-fed evaporator performs better as compared to dry-expansion evaporator. Figure 3 shows the *p-h* plot for gravity-fed evaporator and dry-expansion evaporator to achieve the same cooling capacity. It is observed that it is possible to raise the evaporation temeprature by 4.3 K for gravity-fed evaporator as compared to dry-expansion evaporator. Figure 4 shows how the heat transfer coefficient and vapor fraction changes across the heat exchanger for gravity-fed and dry-expansion system. It is observed that the average heat transfer coefficient of gravity-fed evaporator is more than twice the heat transfer coefficient of dry-expansion evaporator.

Table 2: Simulation conditions

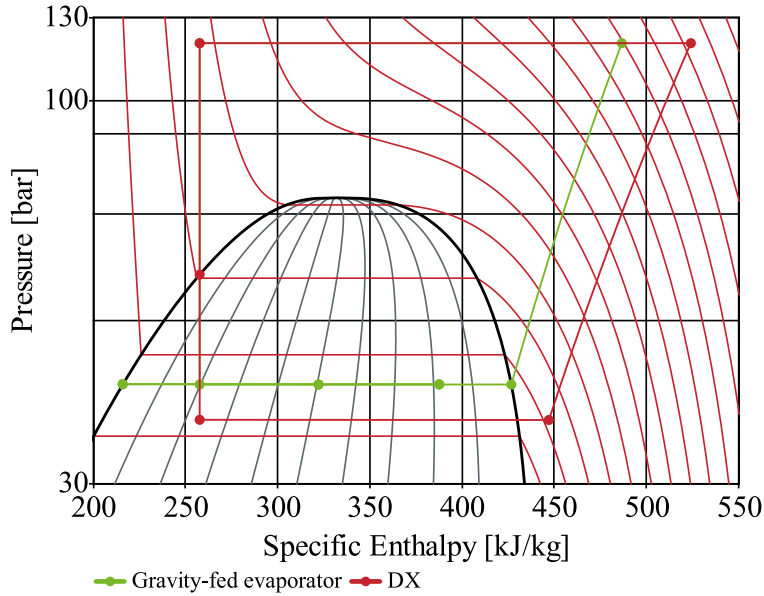| Discharge pressure (bar)	Suction pressure (bar)	Cooling capacity (kW)	Cold water temperature (°C)		Hot water temperature (°C)		GC approach temp. (K)
			in	out	in	out	
120	41	8.3	12	7	24	80	3

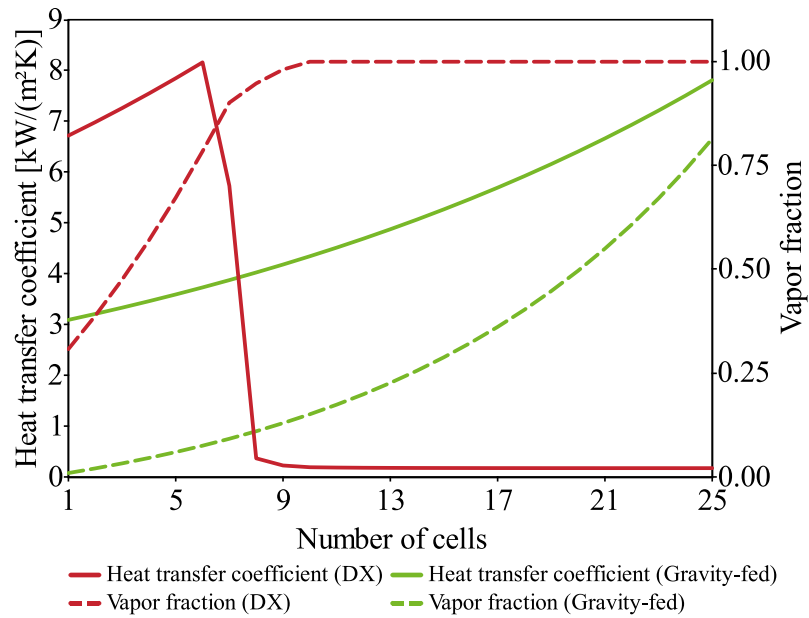**Figure 3:** *p-h* **plot for gravity-fed evaporator and DX evaporator**

Figure 4: Heat transfer coefficient and Vapor fraction for gravity-fed evaporator and DX evaporator

These results clearly show the benefit of gravity-fed evaporator over dry-expansion evaporator. To implement the gravity-fed evaporator, however, it is essential to investigate the effect of loop dimensions on the system performance. One critical parameter is static height which acts as a prime mover to generate the desired flow in the gravity-fed loop by overcoming all the resistances. The flowrate in the gravity-fed loop is defined as self-circulation flowrate in this study. The self-circulation flowrate strongly depends on the static height and total pressure resistances in the gravity-fed loop. To investigate the effect of height, it is varied from 0.35 m to 1.43 m. A very low static height generates low self-circulation flowrates which gets superheated at the exit of evaporator. This degrades the heat transfer rate in the evaporator and hence the overall performance. An increase in static height increases the self-circulation

flowrate and thereby superheating can be avoided and low vapour fractions can be achieved at the exit of evaporator. It is observed that static height of 0.68 m maintains vapor fraction of 0.8 which can be considered as optimum condition for the specified dimensions and capacity of the system. Figure 5 shows the effect of static height on cooling capacity, self-circulation flowrate, and primary flowrate. The pressure difference in the downcomer, evaporator, and riser are shown in figure 6.
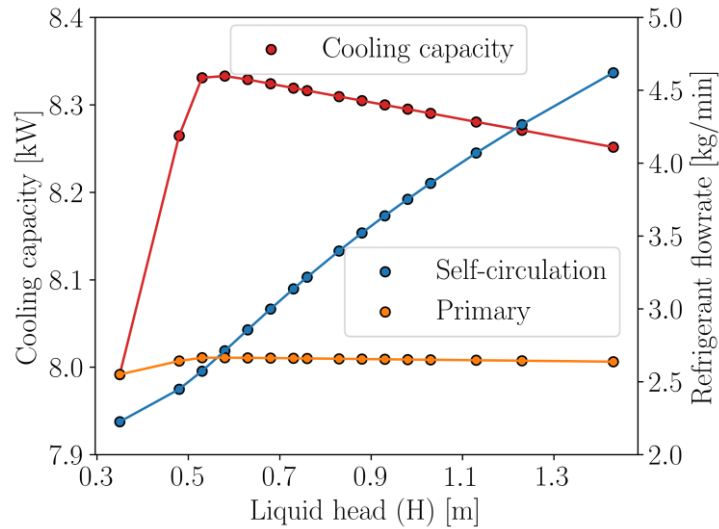

Figure 5: Effect of static height on cooling capacity, self-circulation flowrate, and primary flowrate
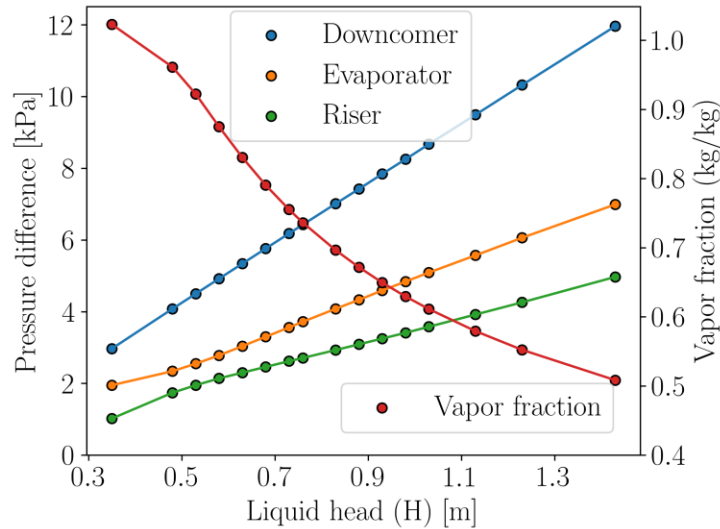

Figure 6: Effect of static height on pressure difference in downcomer, evaporator, and riser

Another important parameter is diameter of the riser tube in the gravity-fed loop. In this study, the riser diameter is first considered same as the downcomer diameter which is 14 mm. Then the riser diameter is varied from 14 mm to 28 mm. It is observed that as the riser diameter is increased, the pressure drop in the riser decreases and hence the self-circulation flowrate increases. Initially the flowrate increases sharply and then the rate of increment becomes steady for a riser diameter of 28 mm. Hence, it is not

effective to increase the riser diameter beyond 28 mm for the investigated case. Figure 7 shows the pressure difference in the downcomer, evaporator, and riser with changes in riser diameter, while figure 8 shows the cooling capacity. It is also observed that with higher self-circulation flowrate, the vapor fraction decreases at the exit of evaporator. Hence it can be concluded that selecting larger riser diameter as compared to downcomer diameter is beneficial. The riser diameter should be twice the size of downcomer diameter. Selecting a larger riser diameter gives the opportunity to reduce the static height of the gravity-fed loop.
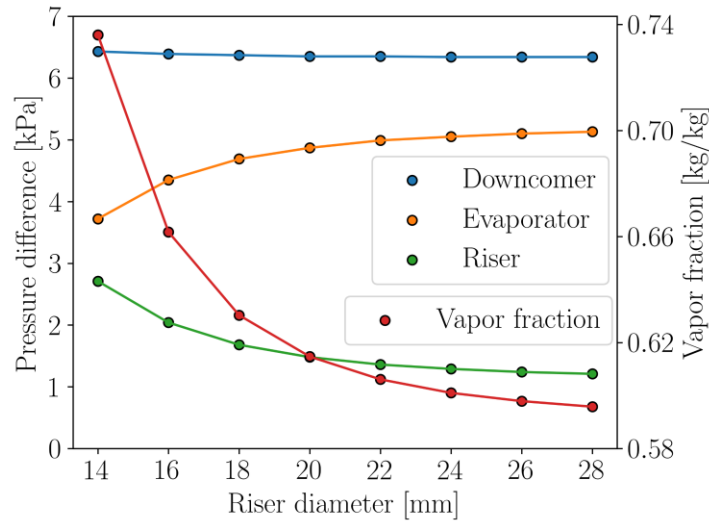
Figure 7: Effect of riser diameter on pressure difference in downcomer, evaporator, and riser
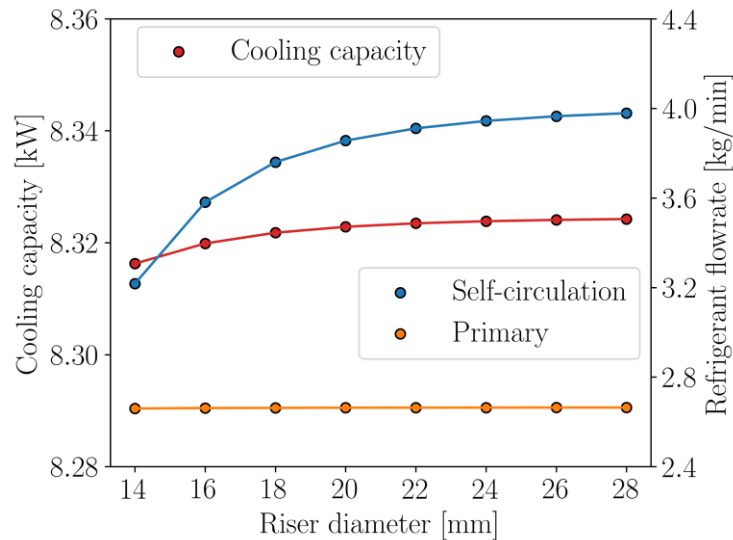
Figure 8: Effect of riser diameter on cooling capacity and refrigerant flowrate

5. CONCLUSIONS

This study presents the results for implementation of gravity-fed evaporator in a CO_2 heat pump chiller. Such evaporator configurations are operated on flooded mode to maintain low vapor fraction (≈0.8) at the exit. This gives the possibility to achieve enhanced and high heat transfer coefficients. Simulations results show that the average heat transfer coefficient of a gravity-fed CO_2 evaporator is more than twice the heat transfer coefficient of dry-expansion evaporator. As a result, there is improvement in heat transfer rate. Better heat transfer rate gives the opportunity to raise the evaporation temperature. It is observed in the investigated case, that the evaporation temperature of gravity-fed evaporator is 4.3 K higher as compared to dry-expansion evaporator. These results clearly show how the system performance and energy efficiency can be enhanced by implementing gravity-fed evaporator. However, it is essential to design the gravity-fed evaporator loop properly to achieve the desired system performance. The static height '*H*' and riser diameter are two critical parameters that significantly affect the system performance. Simulation results show that the static height of 0.68 m maintains vapor fraction of 0.8 which can be considered as optimum condition for the specified dimensions and capacity of the investigated system. Also, it is found that selecting a larger riser diameter helps in reducing the total resistances in the gravity-fed loop and hence gives the opportunity to reduce the static height. Results suggest that the riser diameter should be twice the size of downcomer diameter.

ACKNOWLEDGEMENTS

NOMENCLATURE

				Greek letters
d	Internal diameter of connecting tubes (m)		ρ	Density
f	Friction coefficient for flow inside pipe		μ	Dynamic viscosity (kg m^{-1} s^{-1})
$\dot{G}$	Mass flux (kg m^{-2} s^{-1})		ξ	Coefficient of resistance for pipe components
g	Gravitational acceleration (m s^{-1})		β	Two-phase multiplier
H	Liquid head above the entrance of evaporator (m)			*Subscripts*
h	Height of evaporator (m)		*ev*	evaporator
L	Length (m)		*g*	gas
$\dot{m}$	mass flowrate (kg s^{-1})		*i*	Referred to inlet or downcomer
Δp	Pressure drop due to flow resistance (N m^{-2} = pa)		*l*	liquid
$\sum \Delta p$	Sum of pressure drops in pipe components (N m^{-2} = pa)		*o*	Referred to outlet or riser
V	Velocity of fluid (m s^{-1})			
$\dot{x}$	Refrigerant vapor fraction (kg kg^{-1})			

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Open access to applied data