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ANALYSIS OF THE REFRIGERANT MASS CHARGE FOR A DIRECT EXPANSION SOLAR ASSISTED HEAT PUMP SYSTEM

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Abstract. *The analysis of the amount of mass in a refrigeration system is directly related to the coefficient of performance of the cycle. The mass inventory of solar-assisted heat pumps is still more relevant because the evaporation temperature varies greatly according to the solar radiation flux. In that way, a mass charge model for DX-SAHP operating with* CO_2 *is presented. The results show that a greater solar energy input in the system increases the evaporation temperature, because of that the total mass of the refrigerant in the system need to be increased. The solar evaporator and the gas cooler demonstrated to be the mostly important thermal devices for the mass inventory analysis, which had more than 85% of the total mass of the system. In addition, the mass charge demonstrated a significatigly related to the discharge pressure of the system and the COP.*

Keywords: *Mass charge, refrigeration, heat pump, solar radiation, coefficient of performance.*

1. INTRODUCTION

In the past decade, the solar energy has been drawing the worldwide attention specially to be a cheap, clean, readily available energy (Hawlader, Chou and Ullah, 2001). In this way, the combination of solar energy systems with heat pump have demonstrated to be promising technique since it reduced the consumption of fossil energy sources and increases the performance of the refrigeration systems. The solar heat pumps systems have many applications such as water heating, space heating, space air conditioning and others (Omajaro and Breitkopf, 2013).

As it can see in Fig.1, in the direct-expansion solar assisted heat pump (DX-SAHP), the solar collector serves as an evaporator where the refrigerant absorbs the incident solar energy, and also the ambient energy such as the condenser heat from the humidity of the air. Then the condenser can reject this energy to another source as an example to heat water. Since the solar collector system can supply energy at temperatures higher than the ambient outdoor air, the capacity and COP (coefficient of performance) of the DX-SAHP system would increase over that for the air-source heat pump system alone (Omajaro and Breitkopf, 2013; Kong et. al., 2011; Hawlader, Chou and Ullah, 2001).

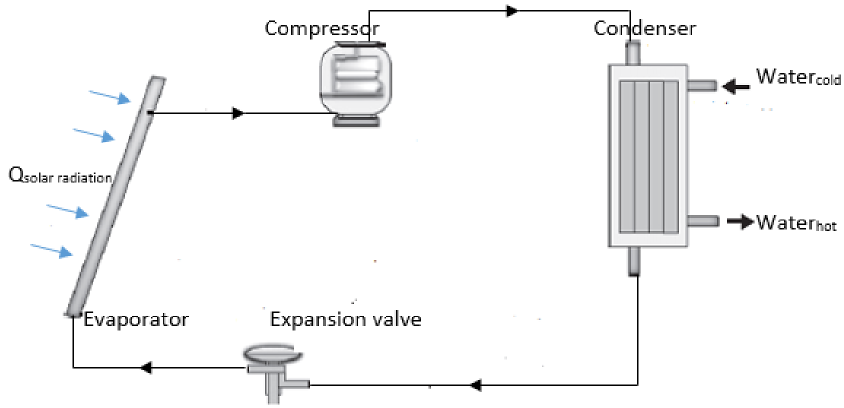

Figure 1. Schematic of a direct expansion solar assisted heat pump system

There are many researches focused on DX-SAHP systems investigating the system structure, thermal performance, working fluid characteristics, operational control, numerical simulation, such as works presented by Kuang *et al.* (2003), Chow *et al.* (2010), Kong *et al.* (2011), Oliveira *et al.* (2015), and Kong *et al.* (2017). However, few studies reported the analysis of refrigerating fluid charge that is ideal to operate those systems.

According to Porto *et al.* (2013), the refrigeration system performance is directly related to the refrigeration mass charge in the equipment, which means that there is an optimum fluid amount that requires less power in the compressor contributing indirectly for a reduction in greenhouse output. Björn (2007), Farzard and O'Neal (1991), and Rice (1987) demonstrated on their work that depend on the refrigeration charge system has, the coefficient of performance (COP) may increase or decrease. Each heat pump design have an optimum amount of total charge that gives a higher COP to the unit.

Asfari *et al.* (2017) showed that the charging has a strong effect on heat pump COP and strongly related to the compression ratio. Li *et al.* (2015) demonstrated that the refrigeration in the heat exchangers was about 60% of the whole charge at steady conditions. In addition, the authors observed that for heating mode the increases of the compressor speed decreases the refrigerant mass in the evaporator, and the low temperature affects the distribution of the refrigeration. Björk and Palm (2006) evaluated the performance of a domestic refrigerator under influence of varied expansion device capacity, refrigerant charge and ambient temperature. The authors found that the optimum charge increase at lower ambient temperature.

However, there is not many analysis that predict the charge inventory for a DX-SAHP. Especially for those systems, the effect of solar radiation plays a big rule to determine this parameter since it can significantly vary the evaporation temperature. This work presents a mass charge model to determine the amount of charge for two different conditions of solar radiation flux for a DX-SAHP operating at a transcritical cycle with CO_2.

2. METHODOLODY

2.1 Experimental facility

A protoype of the DX-SAHP was developed at the Federal University of Minas Gerais in Belo Horizonte with the purpose of heating water. The system has a hermetic reciprocating compressor with displacemnte of 1,75cm^3, 110-127V, model SANDEN SRCaDB. Unlike others systems the condenser in this heat pump is a gas cooler. The low critical temperature (31,1°C) of the CO_2 (R744) makes the operation of heat exchanger to be above the critical point resulting in transcritical cycle. Since the refrigerant fluid does not change phase, the heat exchanger is called a gas cooler as showed the diagram pressure versus especific entalphy for a subcritical and transcritical cycle, respectivaly observed in Fig.2.

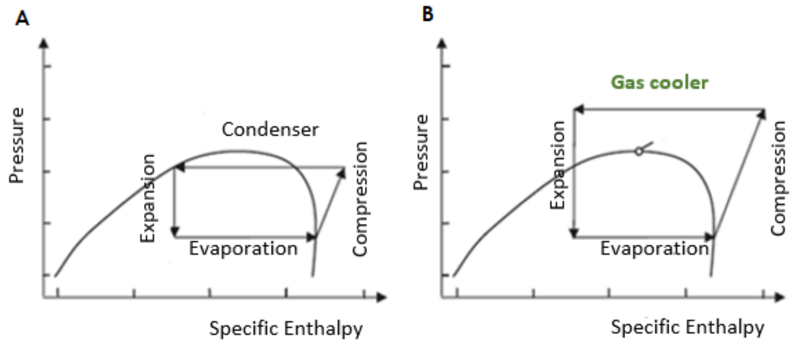

Figure 2. Diagram pressure versus specific entalpy for a subcritical (A) and transcrical cycle (B)

The gas cooller characteristics are showed at Tab.1, as well the evaporator which is a solar collector. The expansion device is a needle valve, model Swagelok SS-31RS4 with a orfice area of 1,6mm^2. Additionally, the system has a water pump and a tank with the capacity of 200L to storage the hot water. Figure 3 demonstrates all the components of the protoype.

Table 1. Characteristics of the gas cooler and the solar evaporator of the DX-SAHP.

Heat Exhangers	**Gas Cooler**	**Solar Evaporator**
Configuration	Countercurrent concentric	Flat plat, single pass
Inner and outer tube diamenter	4.66mm and 6.34mm	4.66mm and 6.34mm
Total length of tubing	24.3m	16.3m
Area of heat exchange	0.4589m^2 (internal) and 0.9160m^2 (external)	1.57m^2
Material	Copper	Copper

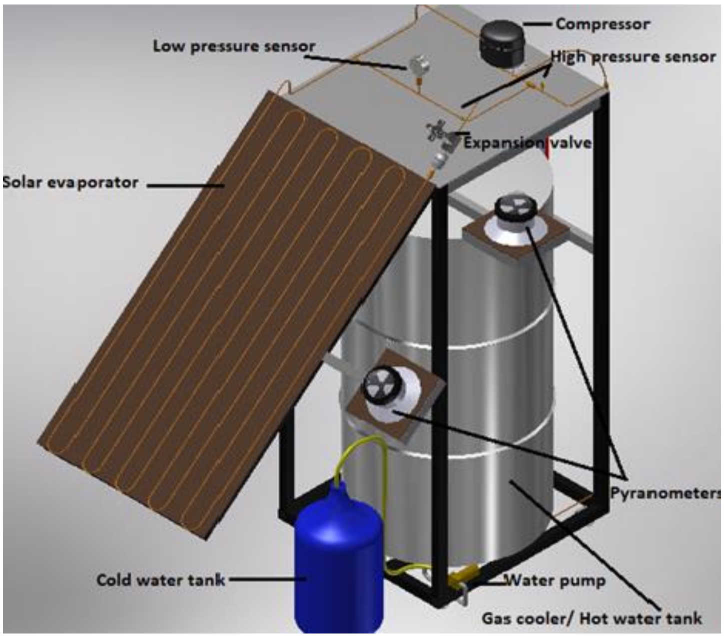

Figure 3. Shemectic of the CO_2 DX-SAHP

Thermocouples T-type were installed at the inlet and outlet of each component to measure the temperatures of the refrigerant. Two pressures sensors were used to measure the high pressure of the system and one mechanical pressure gauge installed at suction of the compressor to measure the low pressure. Pyranometers were installed to measured the solar radiation, one on the horizontal plane (180°degrees) and the other at the same inclination of the solar collector on 30° degrees.

All the piping were measured to calculate the mass inventory of the DX-SAHP. A copper wire was used to measure all the lenghts incluinding the curves of the system. Then, the length of this wire was gauged by a measuring type. The water pipe diameters were measured at the gas cooler inlet, the CO_2 tubing internal diameter was measured at the expansion valve inlet, and the external diameter at the differents points along the piping.

Five measurements were done for each measured quantities. The expanded ucertainty was obtained by BIPM (2008) reference considering the instruments and measurements uncertainties. Equation 1 showed the uncertainty function and the Tab. 2 the measurements obtained for the heat pump piping.

$$u_y^2 = \pm \sum_{i=1}^{N} \left(\frac{\partial y}{\partial x_i} u_{x_i} \right)^2 \tag{1}$$

where u_y is calculated uncertainty, u_{xi} is the uncertainty parameter mesuaremented (x_i) .

Table 2. Piping Measures.

Description	**Measuring Instrument**	**Measurements(mm)**	**Uncertainty (±mm)**
R744 external piping diameter	Caliper	6.34	0.16
R744 internal piping diameter	Caliper	4.6	0.12
Water internal piping diameter	Caliper	12.5	3.3
Water external piping diameter	Caliper	10.52	0.15
Piping length between the expansion valve and solar evaporator	Measuring tape	300.00	2.9
Piping length between the solar evaporator and compressor	Measuring tape	1130.00	8.6
Piping length between the compressor and gas cooler	Measuring tape	2650.00	20
Piping length between the gas cooler and the expansion valve	Measuring tape	2890.00	23

2.2 Mass charge model

The mass charge model was developed considering that the DX-SAHP operating at a transcritical cycle with dioxide carbon in quasi-steady state condition. The pressure drop at the solar evaporator and the gas cooler were neglected as well the heat losses at the gas cooler. The superheat at evaporator outlet was maintained constant for all cases in 10 K, and the compression was considered isentropic. All simulation properties evaluation were carried out in Engineering Equation Solver (EES). The inventory refrigerating charge considered the compressor, heat exchangers, expansion devices, accessories and piping.

The state of the refrigerant in the system can be divided into two types, single-phase and two-phase, which should be calculated, respectively. According to Rice (1987) and Porto *et al.* (2013) a major difficulty in charge inventory analysis is proper prediction of the refrigerant mass in the two-phase regions of the condenser and the evaporator. The degree of vapor-to-liquid slip at each cross section in the two-phase region, and the variation of refrigerant quality with length through the two-phase region are some parameters that need to be estimate. This model considers linear the variation of vapor quality in the length through the two-phase.

The single-phase refrigerant mainly exist in the superheated area at the solar collector (evaporator), the compressor, connection pipes and in the gas cooler since in this DX-SAHP the refrigerant fluid exchange heat in a transcritical region. Equation 2 showed the calculation of the refrigeration mass in single phase.

$$M_m = \int_0^V p_m \, dV = \sum_{j=1}^{N} p_{m,j} V_j \tag{2}$$

where M_m is the mass of the single-phase refrigeration, p_m the density of the refrigeration of the single-phase, dV is the variation volume of the section, and n is the amount of infinitesimal sections.

The two phase mass that are mainly in the evaporator should be calculated considering the avoid fraction, as showed in Eq.2.

$$M_b = \int_0^V \gamma p_v + (1-\gamma)p_l dV = \sum_{j=1}^{N} (\gamma p_{v,j} + (1-\gamma_j)p_{l,j})V_j \quad (3)$$

The p_l and p_v are the densities of the saturated liquid and vapor, respectively, and the γ is the avoid fraction calculated by Hughmark (1962) correlation that provide the best results in refrigeration systems (Farzad and O´Neal, 2006; Poggi *et al*., 2008).

3. RESULTS AND DISCUSSION

The mass charge model considered two different conditions for the transcritical cycle of CO_2 to determine the refrigeration mass charge for DX-SAHP. Experimentally, it was observed that the evaporator temperature could change range of 0°C to 20°C depends on the flux radiation at the solar collector. Then, Tab. 3 shows the test conditions considering a high and low flux of solar radiation, and Tab.4 represents the grid test applied for the evaporator and gas cooler.

Table 3. Simulation conditions.

Test	1 (Insignificant flux of solar radiation)	2 (High flux of solar radiation)
Evaporation temperature	0°C	20°C
Outlet water temperature	45°C	55°C
High Pressure	85 bars	95 bars
Degree of superheating	10°C	10°C

Table 4. Grid test.

Number of divisions	50	100	200	**400**	800
Mass charge test 1 (g)	399.6	394.4 [-1.3]	392.2 [-0.6]	**390.8 [-0.4]**	390.5 [-0.1]
Mass charge test 2 (g)	434.2	427.1 [-1.6]	423.5 [-0.8]	**423.4 [-0.02]**	421.4 [-0.47]

*[0.0] Represents the percentage variation related to the previous result.

The number of the control volume determined was 400 for all analysis, once it represented a low percentage variation with 0.4% and 0.02%, respectively, for both conditions.

Figure 4 and 5 demonstrated the mass of CO_2 distribution for Test 1 and 2, respectively. It can observed that for higher incidence of solar radiation (Test 2) the mass in the solar evaporator increases while in the gas cooler decreases compared to the Test 1 that has a lower incidence of solar radiation.

Because of the greater energy, input in the solar collector the evaporator temperature increases as well as the specific mass. Thus, the volume in the evaporator is the same, the refrigerant mass also increases in this thermal device for higher radiation flux. The opposite behavior is noticed in the gas cooler. Due to a slightly reduction in the higher pressure of the system and as a consequence of migration of mass to the evaporator, the mass of the gas cooler reduces for higher radiation flux.

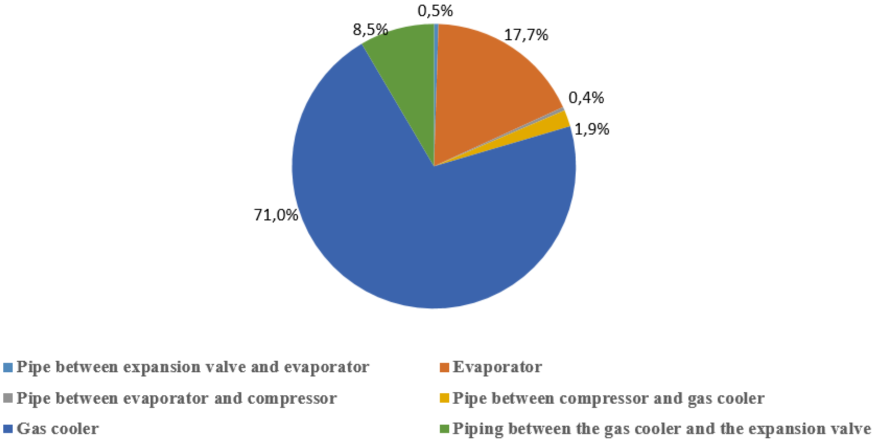

Figure 4. Distribution of mass charge in the DX-SAHP for Test 1

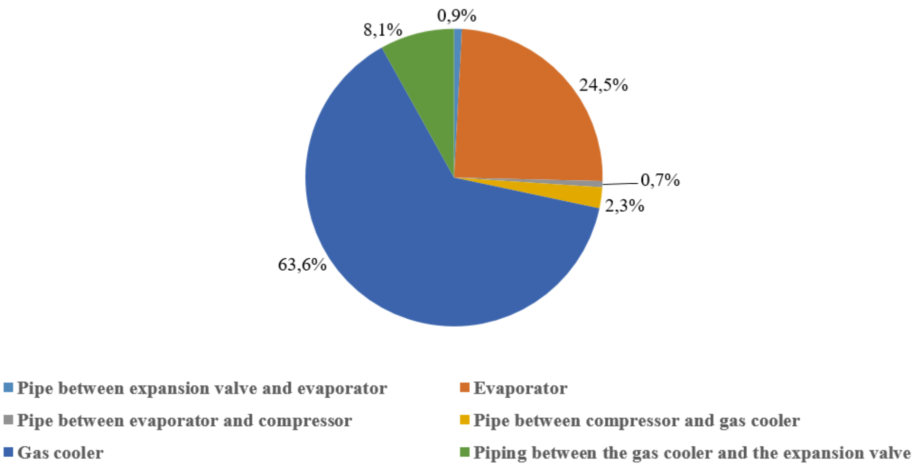

Figure 5. Distribution of mass charge in the DX-SAHP for Test 2

Also, it can observed that in the mass analysis of DX-SAHP, the most important thermal devices are the heat exchangers, since they contained the largest amount of mass in the system, upper to 85% of the total refrigerant mass. In the others devices and pipes the mass has a slightly variation, mainly function of the transition of the mass in the heat exchangers. For an example, the piping between the gas cooler and the expansion valve reduced the mass from 8.5 to 8.1%, accompanying the observed trend for the gas cooler for a high radiation flux.

Figure 6 shows a comparison between the mass charge, coefficient of performance and the high pressure of the system. It can noticed that for a higher evaporation temperature (20°C), which consequently is the result of a higher solar radiation flux, the system needs a larger amount of refrigerant mass. Besides of that, the coefficient of performance of the cycle is significantly better, consequence to the greater energy input.

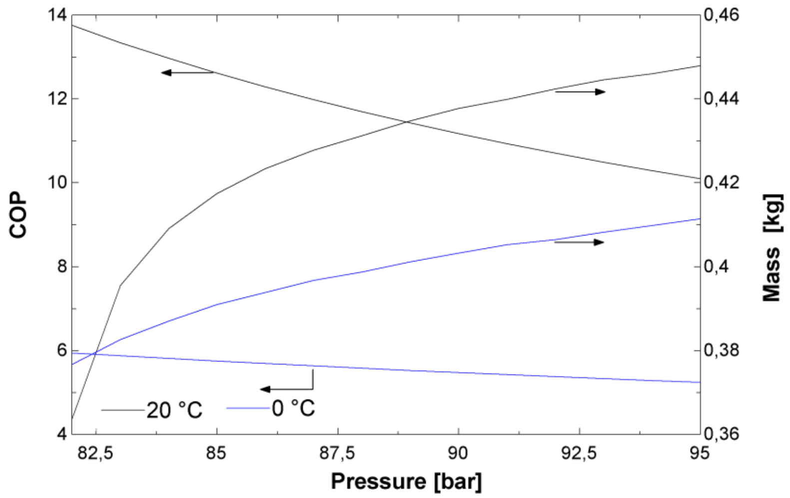

Figure 6. High pressure function of COP and refrigerant mass charge

For both cases, it is possible to notice that the higher discharge pressure of the system, the greater is the required mass charge. In addition, it is observed that the increase of the high pressure, the COP of the system decreases slightly.

Figure 7 shows that the degree of superheating in the outlet evaporator just have a slight influence over the mass charge of DX-SAHP and almost no influence on the COP. For the evaporator temperature of 0°C the amount of refrigerant mass requires is about 418 grams, and for 20°C is 390 grams.

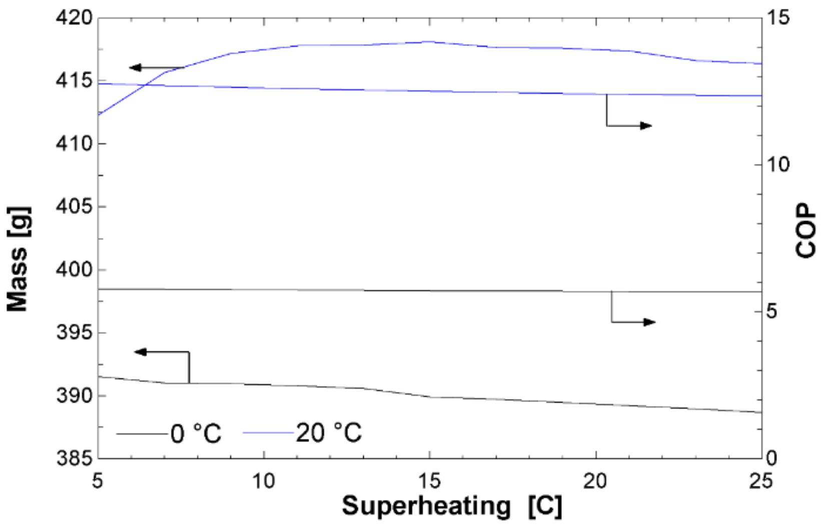

Figure 7. Superheating degree function of the refrigeration mass charge and COP

For the results that were presented, it can seen that use of a liquid accumulator is fundamental for a DX-SAHP. The accumulator will provide a mass of the system as needed, since solar radiation is a non-controllable variable of this type of system, and changes frequently.

4. CONCLUSION

A mass charge model for a DX-SAHP operating in a transcritical cycle with carbon dioxide is presented. For high solar radiation rates, the system has been shown to require a greater mass of refrigerant compared to a lower solar radiation rates. Charging demonstrated to have a great effect at the COP of a solar assisted heat pump. Besides the peculiarities of each system, the environmental conditions, especially the flux of solar radiation, has an influence on the amount of refrigerant mass increasing or not the COP of the system. Also, heat exchangers showed to be the most important thermal devices for the mass inventory. The refrigerant in the heat exchangers were about 85% of the whole charge at steady conditions for both cases analyzed.

5. ACKNOWLEDGEMENTS

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