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# PERFORMANCE COMPARISON OF DIRECT EXPANSION SOLAR ASSISTED HEAT PUMP WORKING WITH R1234yf AS A DROP-IN REPLACEMENT FOR R134a

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**Abstract.** *There are several studies that compare the performance of R-134a and R-1234yf, but no yet for a direct expansion solar assisted heat pump (DX-SAHP). This theoretical analysis was made using lumped models for heat exchangers and a black box model for the hermetic R134a compressor. This modeling was validated using experimental results from DX-SAHP running with R134a in different environment conditions. The average difference between experimental and theoretical COP is 1.7%, lower than the uncertainty of COP experimental that is 4.9%. After that, this model is employed to the performance comparison of R-1234yf as a drop-in replacement for R-134a in a DX-SAHP. The results present that increased of the solar radiation, the ambient temperature or wind speed produce an increase in the COP for both refrigerants. Furthermore, the results show a slightly better performance for the R-134a compared to R-1234yf, the COP of R-134a from 2.1% to 4.2% higher.*

**Keywords:** R-1234yf, R-134a, Heat pump, Solar Assited, Replacement

## 1. INTRODUCTION

Heating water for sanitary propose using a heat pump reduce the energy consumption of the electricity in comparison of the use of the electric heaters. Furthermore, when the heat pump is solar assisted, it operates with better COP than a non-solar assisted heat pump (Sun *et al.*, 2014). Different refrigerants have been used in the DX-SAHP for water heating such as R-12 (Ito *et al.*, 1999), R-22 (Kong *et al.*, 2011), R-134a (Sun *et al.*, 2014), R-744 (Faria *et al.*, 2016) and R-410A (Kong *et al.*, 2017).

Chata *et al.* (2005) analyze the COP in a DX-SAHP with different refrigerants. The refrigerants analyzed were R-12, R-22, R-134a, R-404A, R-407C and R-410A. The results showed the best COP for R-12, R-22 and R-134a respectively,

but the difference between R-12 and R-134a were in the range of 2% to 4%. As the refrigerants R-12 and R-22 have Ozone Depletion Potential (ODP) not equal to zero. R134a has zero ODP, because of that, it was chosen as the best refrigerant for a DX-SAHP.

In the last decade, regarding the Kyoto Protocol, some countries have initiated programs to reduce the utilization of refrigerants with higher GWP (Global Warming Potential) in different refrigeration systems. The refrigerant R-1234yf have been used as the replacement of the R134a in different systems (Lee and Jung, 2012; Direk *et al.*, 2016; Belman-Flores *et al.*, 2017). According to ASHRAE (2013), the GPW (reference to CO<sub>2</sub> with base values of 1) of the R1234yf is 4 instead of 1370 for the R-134a. In addition, for this retrofit besides of refrigerant replacement it is only necessary change the compressor lubricant oil. Then, it is possible to notice that the R-1234yf has less environment impact than the R-134a and the retrofit of both is not difficulty to be implemented.

In this paper for an existing DX-SAHP that operates with R134a are presented a performance comparison with R1234yf to support the retrofit. The authors' best efforts did not identify the use of the R-1234yf in a DX-SAHP. A mathematical model was presented and validated for a R-134a DX-SAHP, after that, this model was used to produce the performance comparison. The objective of this paper is to measure the drop-in performance of the R-1234yf in replacement to R-134a for a DX-SAHP.

## 2. EXPERIMENTAL SETUP

The existing R134a DX-SAHP is installed at UFMG (Universidade Federal de Minas Gerais) in Belo Horizonte (MG), Brazil. Figure 1 shows the heat pump in which the main components are the evaporator / collector, the compressor, the immersed condenser, the 200 liters hot water tank, and the thermostatic expansion valve. The evaporator/solar collector was designed to operate as static air evaporator as described by Reis (2012). The collector has a thickness of 1 mm and the area of 1.65 m<sup>2</sup>, the evaporator tube has the length of 17.3 m, and the distance between the tubes is 103 mm. The inner and outer diameters are 8.73 mm and 9.53 mm for the evaporator/collector and the immersed condenser. The material of the tubes in the evaporator/collector and in the immersed condenser are cooper. The immersed condenser is basic a 4.5 m horizontal copper tube, without fin, folded into tank bottom. The heat pump has a R134a hermetic compressor model FFU100HAK manufactured by Embraco with 7.95 cm<sup>3</sup> of displacement and it is driven by a 2 pole asynchronous electric motor.



Figure 1. R134a DX-SAHP is installed at UFMG

### 3. MATHEMATICAL MODEL

In order to evaluate the performance of DX-SAHP for producing DHW (domestic hot water) a quasi-steady-state model was developed using the Equation Engineering Solver (EES). The losses in the tubes between components was considered negligible and for the inventory charge of the refrigerant, the pipeline was considered two meters long. The evaporator/solar collector and condenser was assumed as isobaric and a lumped model was used. The expansion device used is a thermostatic valve because of the variations in solar radiation. The expansion valve is adjusted to maintain the superheat at evaporator outlet in 7 K, and expansion process is modeled as isenthalpic. Following is described the modelling equation for each component.

#### 3.1 Compressor model

The refrigerant mass flow rate ( $\dot{m}$ ) in a constant rotation speed reciprocating compressor is given by:

$$\dot{m} = \rho_1 n V_s \eta_v \quad (1)$$

where  $\rho$  is the refrigerant density,  $n$  is the rotation speed,  $V_s$  is the compressor swept volume,  $\eta_v$  is the volumetric efficiency and the subscript 1 refers to compressor inlet or evaporator outlet. The compressor electric power consumption ( $\dot{W}$ ), considering a isentropic compression process, is evaluated as follow:

$$\dot{W} = \frac{\dot{m}(i_2 - i_1)}{\eta_g} \quad (2)$$

where  $\eta_g$  is the global efficiency and  $i$  is the refrigerant specific enthalpy and the subscript 2 refers to compressor outlet or condenser inlet. The global and volumetric efficiency was determinate fitting equations proposed by Minetto (2011) to the compressor performance map available in Embraco website. The global and volumetric efficiency is given by:

$$\eta_v = -0.0143 \left( \frac{P_2}{P_1} \right) + 0.915 \quad (3)$$

$$\eta_g = -0.0004 \left( \frac{P_2}{P_1} \right)^2 + 0.0104 \left( \frac{P_2}{P_1} \right) + 0.4839 \quad (4)$$

where  $P$  is the refrigerant pressure. The coefficient of determination ( $R^2$ ) for volumetric efficiency is 97.6% and for global efficiency is 94.4%.

#### 3.2 Direct expansion solar evaporator

The heat transfer rate received by the refrigerant in the evaporator ( $\dot{Q}_e$ ) is given by:

$$\dot{Q}_e = \dot{m}(i_1 - i_4) \quad (5)$$

where the subscript 4 refers to thermostatic valve outlet or evaporator inlet. To evaluate the energy gain in a flat plate collector in steady-state condition Kong *et al.* (2011) suggest the following equation:

$$\dot{Q}_e = A_e F' [S - U_L (\bar{T}_r - T_a)] \quad (6)$$

where  $A_e$  is the area of evaporator of the solar collector,  $F'$  is the collector efficiency factor,  $S$  is the net radiation absolved per unit of area,  $U_L$  is overall heat loss coefficient,  $\bar{T}_r$  is the average temperature of the refrigerant fluid and  $T_a$  is the ambient air temperature.

The collector effectiveness factor is calculated using the Hottel-Whilliar-Bliss model described by Duffie and Beckman (2013), considering that the resistance to heat flow due the bond between the collector plate and tube can be neglected, is given by:

$$F' = \frac{1}{U_{ev}} \left\{ W \left[ \frac{1}{U_{ev} [D_o + F(W - D_o)]} + \frac{1}{\pi D_i h_i} \right] \right\}^{-1} \quad (7)$$

where the distance between the tubes in the evaporator is  $W$ , the fin efficiency is  $F$ , the outer diameter is  $D_o$ , the inner diameter is  $D_i$ , the internal convective coefficient is  $h_i$  that is calculated by the correlation proposed by Shah (2017) for two phase flow and by the correlation proposed by Gnielinski (1976) for single phase flow.

The fin efficiency can be evaluated by:

$$F = \frac{\tanh \left[ (w - D_o) / 2 \sqrt{U_L / (k\delta)} \right]}{(w - D_o) / 2 \sqrt{U_L / (k\delta)}} \quad (8)$$

where  $\delta$  is the fin thickness and  $k$  is the thermal conductivity. The net radiation absolved is evaluated as made by Kong *et al.* (2017):

$$S = aI - \varepsilon\sigma(T_r^4 - T_s^4) \quad (9)$$

where the absorptivity is  $a$ , the solar radiation intensity normal to evaporator is  $I$ , the emissivity is  $\varepsilon$ ,  $\sigma$  is the Stefan-Boltzmann constant and  $T_s$  is the sky temperature.

The overall heat loss coefficient proposed by Kong *et al.* (2011) is determined by:

$$U_L = h_o + 4\varepsilon\sigma T_a^3 \quad (10)$$

where the external convective coefficient ( $h_o$ ) is calculated by the collection of correlations for free and forced convection, depending on wind speed, for tilted flat plate listed by Neils and Klein (2009).

### 3.3 Immersed condenser at hot water tank

Considering non-stratification in the water at the tank, the energy balance of water in the tank  $\dot{Q}_t$  can be evaluated as follow:

$$\dot{Q}_t = \rho_w V_w C_w \frac{\partial T_w}{\partial t} \quad (11)$$

where  $V_w$  is the volume of water inside the tank,  $\rho_w$  is the water density,  $C_w$  is the water heat capacity at constant pressure,  $T_w$  is the water temperature at the tank and  $t$  is time.

The balance of energy at the condenser proposed by Kong *et al.* (2017) is given by:

$$\dot{Q}_c = \dot{m}(i_2 - i_3) = UA(\bar{T}_r - T_w) \quad (12)$$

where the subscript 3 represents the outlet of condenser and the  $UA$  value is evaluated as follow:

$$UA = \left( \frac{1}{h_i \pi D_i L_c} + \frac{\ln(D_o/D_i)}{2\pi k L_c} + \frac{1}{h_o \pi D_o L_c} \right)^{-1} \quad (13)$$

where  $L_c$  is the condenser length. The inner convective coefficient is obtained from the correlations proposed by Gnielinski (1976) and Shah (2016) for single and two phase flow, respectively. To outer convective coefficient is adopted the correlation for free convection in a horizontal cylinder presented by Rohsenow *et al.* (1998).

In order to consider the heat loss in the hot water tank Kong *et al.* (2017) proposed a heat leakage coefficient ( $\zeta$ ) of 95% that is defined as follow:

$$\zeta = \frac{\dot{Q}_t}{\dot{Q}_c} \quad (14)$$

The coefficient of performance (COP) is defined as follow:

$$COP = \int \frac{\dot{Q}_t}{\dot{W}} dt \quad (15)$$

## 4. RESULTS AND DISCUSSION

### 4.1 Modelling validation

The model validation is performed comparing the experimental COP presented by Diniz (2017) combined by the data available in Brazilian National Institute of Meteorology (INMET) web site with the calculated COP using the model. The comparison between measured and calculated COP is shown in the Tab. 1. In table 1,  $T_{wi}$  is the initial temperature and  $T_{wf}$  is the final temperature at hot water tank. The mean and mean absolute difference between the measured COP and calculated COP are respectively 1.7% and 2.7%, lower than the average uncertainty of experimental COP that is 4.9%.

### 4.2 Comparative analysis of DX-SAHP operating with R134a and R1234yf

In the following results, the parameters listed in Tab. 2 is fixed and the solar radiation, ambient temperature and wind speed are changed. Figure 2 (a) shows the effects of increased solar radiation ( $I$ ) at the COP for the R134a and the R1234yf. The figure 2 presents the increase of the COP with the increase of solar radiation. The COP of R1234yf is in average 2.7% lower than R134a. The lower COP of R1234yf is justified by higher difference of the condensing and evaporating temperature, as shows Figure 2 (b), where condensing temperature is presented in red lines and evaporating temperature is presented in black lines.

Table 1. Comparison between experimental and theoretical COP

Test	Date	$T_a$ ( $^{\circ}C$ )	$I$ ( $W/m^2$ )	Wind ( $m/s$ )	Total time	$T_{wi}$ ( $^{\circ}C$ )	$T_{wf}$ ( $^{\circ}C$ )	Measured COP	Calculated COP
1	01-07-17	28.2	0	0	3:45	32.8	44.9	$2.31 \pm 0.12$	2.33
2	01-14-17	26.1	0	0	4:00	32.6	45.3	$2.27 \pm 0.11$	2.28
3	01-18-17	26.0	0	0	4:30	31.5	45.2	$2.29 \pm 0.11$	2.30
4	01-20-17	26.6	0	0	5:00	30.9	45.9	$2.30 \pm 0.11$	2.31
5	01-24-17	27.3	0	0	4:15	32.2	45.2	$2.27 \pm 0.11$	2.31
6	02-04-17	29.8	482	0.72	3:00	29.6	44.8	$2.88 \pm 0.14$	2.72
7	02-11-17	26.4	346	0.86	4:15	28.1	45.3	$2.58 \pm 0.12$	2.58
8	02-16-17	33.2	520	0.99	3:45	29.8	46.1	$2.64 \pm 0.13$	2.76
9	02-17-17	32.6	671	1.53	3:00	29.3	44.8	$2.80 \pm 0.14$	2.95
10	02-18-17	31.6	807	1.25	3:00	29.4	45.5	$2.91 \pm 0.15$	3.04

Table 2. Simulation parameters list

Parameter	Value	Parameter	Value	Parameter	Value
R134a charge	0.433 kg	R1234yf charge	0.408 kg	Initial water temperature	$25^{\circ}C$
Atmospheric Pressure	101.3 kPa	Collector tilt angle	$30^{\circ}$	Final water temperature	$45^{\circ}C$
Emissivity	0.95	Solar absorptivity	0.95	Ambient temperature	$25^{\circ}C$
Solar radiation	$500 W/m^2$	Wind speed	0 m/s	Superheating	$7^{\circ}C$

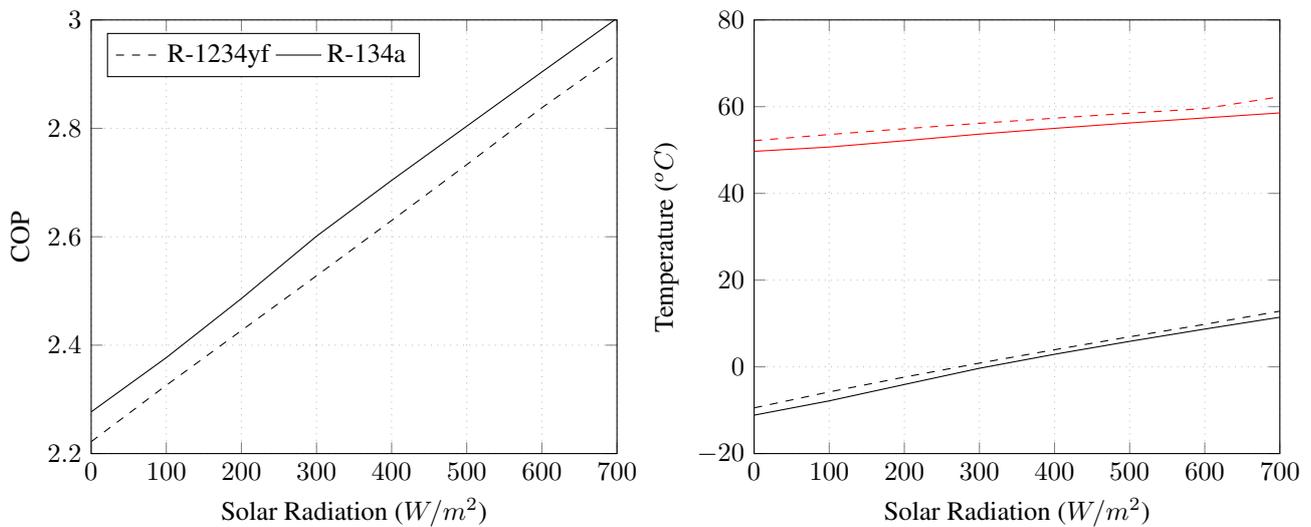


Figure 2. (a) COP for different solar radiation. (b) Evaporating and condensing mean temperatures for different solar radiation.

The same behavior of the solar radiation is noticed for the ambient temperature. Because of the greater energy input in the evaporator/collector. The increased of the ambient temperature rise the COP for the R-134a and the R-1234yf as shows Fig. 3.

Figure 4 shows a slight increase on the COP due to the increase in the wind speed for both refrigerants. The simulation was performed with solar radiation equal to  $500 W/m^2$  and ambient temperature equal to  $25^{\circ}C$ .

Through the analysis of the Fig. 2, Fig. 3 and Fig. 4 it is possible to notice that the greatest variations in the COP are caused by solar radiation, the same conclusion was found by Kong *et al.* (2011); Ito *et al.* (1999); Sun *et al.* (2014). On the other hand, the smaller ones are due to the wind speed. Furthermore, the COP values for the R134a and the R-1234yf are quite similar, but the R-134a COP is slightly better, it is 2.1% to 4.2% higher. In the experimental work carried out by Lee and Jung (2012) the COP of R-1234yf was 0.8% to 2.7% lower than R-134a in a similar analysis for a mobile air conditioner. Aprea *et al.* (2016) compared a performance of a domestic refrigerator using R-1234yf and R-134a and the COP of R-1234yf changed from 0.4% lower to 2.6% higher than COP of R134a.

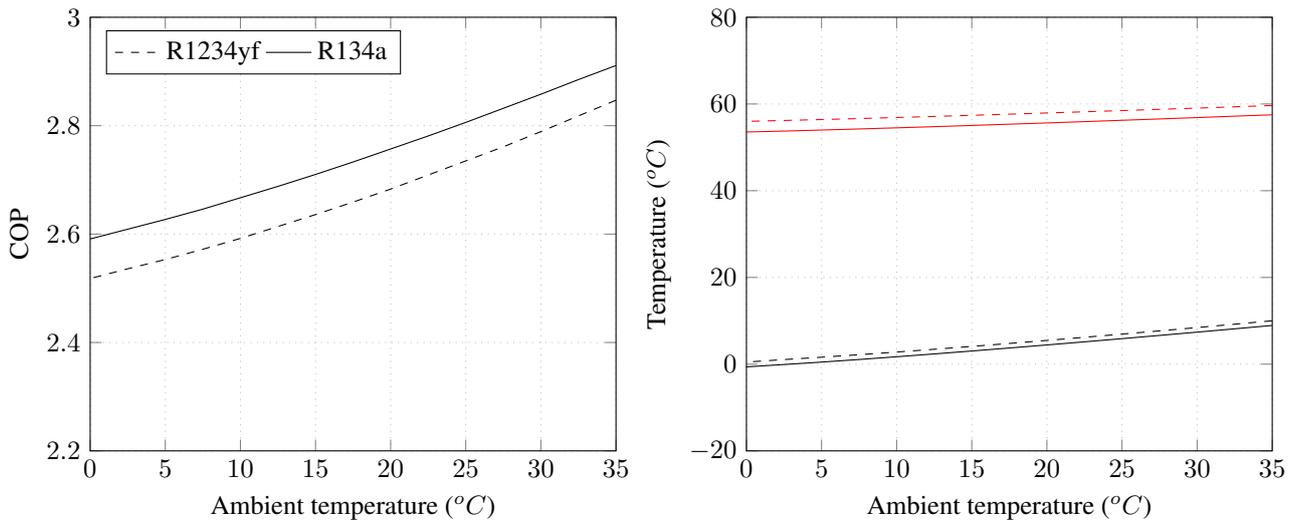


Figure 3. (a) COP for different ambient temperatures. (b) Evaporating and condensing mean temperatures for different ambient temperatures.

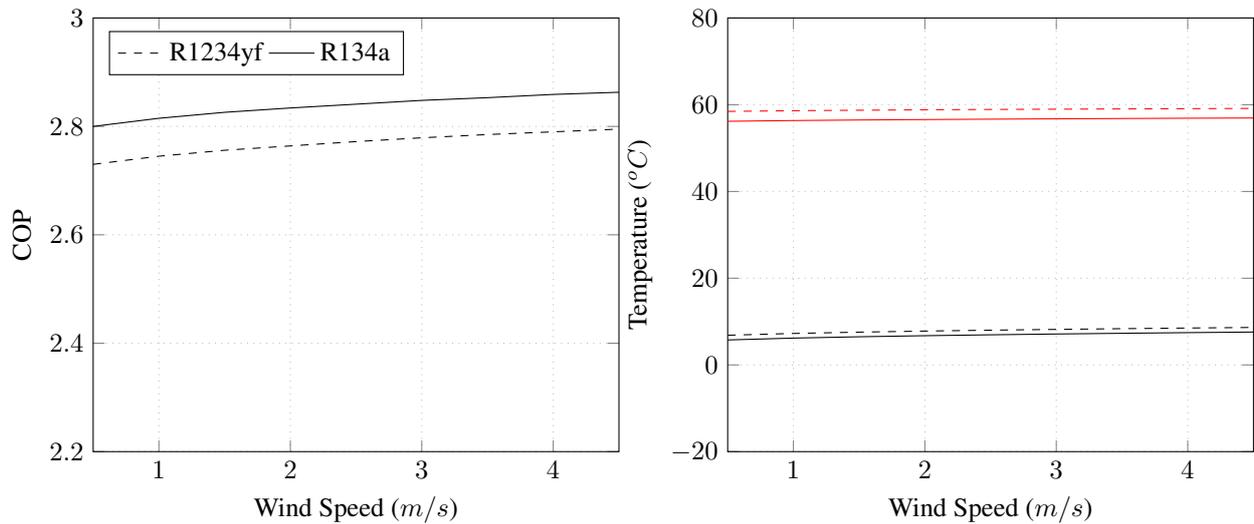


Figure 4. (a) COP for different wind speed. (b) Evaporating and condensing mean temperatures for different wind speed.

## 5. CONCLUSIONS

In this work a mathematical model of a R-134a DX-SAHP for producing domestic hot water is used to compare the performance of the system if the refrigerant is replaced by R1234yf. The refrigerant R1234yf has lower environment impact with lower GWP and ODP equal to zero. Many studies have been done considering the R1234yf as drop-in replacement for R134a due the similar properties.

The mathematical model presented in this work is based in lumped model for the heat exchangers. The model was validated using 10 experimental tests performed in different environmental conditions. The COP of DX-SAHP was measured with average uncertainty of 4.9%. The difference of experimental and predicted COP is 1.7%

The results show a slightly better performance for the R-134a compared to R-1234yf. The trend of results are the same found in the literature about DX-SAHP to produce domestic hot water. In addition, the results present that increased of the solar radiation, the ambient temperature or wind speed produce an increase in the COP.

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