F-Gas regulation– Possible solutions for the retrofit dead end

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Abstract. The EU F-gas regulation of 2006 and the recast of 2014 are forcing the market to reduce the use of refrigerants with high global warming potential (GWP). As a result, the production of hydrofluorocarbon (HFC) refrigerants with higher GWP decreased, making the prices of all HFC gases to increase. Any maintenance problem in a refrigeration system asks to evaluate the retrofitting of the gas making necessary to know the expected behaviour of the system. This paper aims to discuss the gases that can substitute the now-a-days HFCs, and the impact the retrofit will cause in a real air-conditioning systems. Many studies on retrofit address the behaviour of the refrigeration cycle, but usually, do not take into account the behaviour of the system as a whole. This paper models a water-to-water air-conditioning system taking into consideration the evaporator and condenser heat exchangers, the refrigeration cycle, the air-conditioning loop and the heat exchanger to the acclimatized area. Moreover, the paper studies the performance of the system when subjected to high condensing temperatures. The paper concludes that all possible retrofit solutions need to use flammable gases that make the refrigeration power of the equipment to reduce.

1 Introduction

By the end of the twenty century, the global warming made many authorities to discuss the need of stopping the production of gases with high potential to warm the planet. The impact of the CO_2 molecule over 100 years is the unitary reference to define the global warming potential (GWP) of a molecule of refrigerant. Most refrigerants in use in the EU are hydrofluorcarbons (HFC) with high GWP forcing the market to ask for new retrofitting solutions.

On 2014 the European Parliament and the Council launched the regulation on Fluorinated greenhouse gases or F-Gas Regulation. Chapter IV of the regulation forces a reduction in the amount of HFC placed on the market, needing to reduce the HFC use by 89% until 2030, based on the average HFC amounts on the period 2009 to 2012. According to the mentioned regulation, the average GWP of the refrigerants putted in the market are dropping from 2000 on 2015 to 420 on 2030. On 2018 occurs a major cut forcing the market to reduce the value the average GWP of refrigerants placed in the market to the

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value of 1260. It means that the weighted GWP by mass of all the refrigerants putted in the market on 2018 needs to be less than 1260. On 2018 the average values of the GWP dropped from 93% to 63% of the average GWP occurred on the period above [1]. The reduction of HFC placed in a market in expansion caused the price of refrigerants to increase by more than 500%. Therefore, any refrigeration cycle subjected to a major intervention in the coming years may need to change the refrigerant and undergo a retrofit process.

The behaviour of systems undergoing a retrofit is becoming easier due to the capability of platforms to simulate models. A possible approach is a semi-empirical model that uses data from the chiller manufactures to predict the behaviour of the refrigeration system. It takes into account the clearance of reciprocating compressors, uses the ε-NTU method for evaluate the heat exchange, and solves the refrigeration cycles equations [2].

There are many models regarding the evaluation of chillers. Some models evaluate the performance of special coils, others condensers with microchannel, some take into account the oil mixed in the refrigerant. However, ASHRAE Toolkit for refrigeration cycles, the Gordon-Ng Universal Chiller Model, and the DOE-2 model are still the basic models to evaluate the performance of chillers. The DOE-2 model is a correlation model using polynomial function, and the Gordon-Ng model does not make use of the thermodynamic properties of the refrigerants [3]. Therefore, both are unable to discuss the problem of the retrofit of new refrigerants. ASHRAE Toolkit allows evaluating the retrofit of refrigerant gases, assuming isentropic compression and treating the water flow as variable according to the loads. The heat transfer coefficients are constant despite the variation of flow.

The contribution of this paper is to address the retrofit of a refrigeration cycle of an air-conditioning plant considering the system as a whole. It studies the performance of the water-to-water chiller when subject to extreme water temperature entering in the condenser heat-exchanger of 35 °C. Models of refrigeration systems usually take into account the water condenser cycle, the refrigeration cycle, and the water evaporator cycle. This study also models the air handling unit (AHU) in the cooling cycle using the psychrometric evolution. It uses the EES platform to evaluate the thermo-physical properties of each refrigerant on appraisal, makes use of the isentropic efficiency and the volumetric efficiency according to the pressure ratio of the compressor, and takes into account the pressure drop of the refrigerant on the condenser and the evaporator.

The thermodynamic model is described on section 3 and section 4 shows the results of the model using data from retrofit refrigerants shown on Section 2. Section 5 discusses the results and made some conclusions regarding how to solve the problem of the retrofit of current refrigeration equipment dedicated to air conditioning systems.

The paper shows that the most likely currently solutions for retrofitting needs flammable gases. Moreover, shows that the retrofit study needs to take into account the power of the compressor as well as the pressure at the condenser at extreme working conditions.

2 Retrofit gases

The GWP of the most used refrigerants in the air conditioning market, R407c, R410a, and R134a have GWP higher than the average value allowed on 2018 (1260). Therefore, the amount of these gases need to a reduction by using more efficient equipment, by controlling leaks and by using gases with low GWP.

A study regarding retrofitting R22, R134a and R407c with a gas with lower GWP, the R32, shows the R32 allows removing the same amount of heat with lower mass flux and lower compressor ratio [4]. For temperatures below 0 °C, the refrigerant recommended to replace the R22 have been the R507 or the R404a due to the higher transfer capacity of these refrigerants. However, the efficiency of the refrigeration cycle after retrofitting decreases [5]. Regarding the choice of new refrigerants, it needs to take into account the safety, operating pressure, gliding and the specific volume at the suction line [6]. A comparison between R22 and R404A, R407a, R407b, R407c, R507, and R410a shows that the latest makes the cycle to have energy consumption close to the R22. All other gases make the refrigeration system to use 5% to 15% more energy than a R22 cycle. Moreover, in the comparison states that CO_2 is one of the most likely refrigerants of the future [7].

The substituting of the most common gases in the air conditioning market, R134a, R410a, and R22 or R407c needs an evaluation of the Physical Hazard Evaluation (PHE) the GWP and the ODP. Regarding the PHE all proposed gases nowadays are non-toxic (A type), with classification A1 non-flammable, A2L mildly flammable and A3 flammable. R1234yf, R1234ze, R32, are classified as A2L and R290 (propane) and R600a (butane) as A3. R1234yf, R1234ze are hydrofluoro-olefins (HFO), R32 is a HFC and the latest two hydrocarbons (HC), being the most probable choices to substitute the former refrigerants. Other possible refrigerants are blends of the substitution gases having properties and behaviours in between. Carbon dioxide can be the refrigerant of the future although it has a lower critical temperature and a higher critical pressure compared to the commercial refrigerants as R22 or R404a. A possible solution to avoid very high pressures in the CO_2 cycle can be a cascade cycle using CO_2 and NH_3. Both refrigerants are natural, and the cascade cycle uses less mass of ammonia, a toxic gas, than a NH_3 cycle between the same temperatures [8]. Withouse compared R22 with bends of refrigerants and with R404a at two arrangements of temperatures at the evaporator and condenser of -28 °C to 38 °C and -6 °C to 38 °C. The latest arrangement has application on air conditioning systems, and the study shows that R22 has lower mass flux, lower liquid rate, but similar vapour rate than the R404a, due to the changes of density of the refrigerants [9].

3 Refrigeration model

The model presented in this paper uses data from a R22 water to water chiller with a reciprocating compressor with six cylinders. Table 1 shows the data from the chiller, showing the evaporator and condenser nominal powers, the maximum working pressure on the condenser, the displacement flow and the water flow in the evaporator and condenser. The water flow is steady in the evaporator and condenser, which is at 35 °C; the refrigerant flow varies according to the volumetric efficiency that affects the displacement flow. Regarding the AHU, the model studies the system at maximum external conditions of 32°C and 80% of relative humidity, maintaining the indoor ambient at 24 °C and 50% of relative humidity. The AHU supplies 74000 m^3/h of airflow, which results from a mixture of 30% of outdoor air and 70% of return air at indoor conditions. The cooling coil of the AHU cools the supply air with a power depending on the contact factor (*cf*) and the coil temperature. The *cf* is made constant due to a steady airflow, and the coil temperature depends on the cooling water temperature.

The heat global heat transfer coefficient in the evaporator and condenser heat exchangers depends mostly on the water side heat transfer rather than on the refrigerant side. Moreover, the fouling factor tends to increase mostly on the water side of the heat exchanger, making the global heat transfer to be close to the water heat transfer coefficient. In the refrigerant side, the heat resistance is very small as there are high heat transfers coefficients due to the evaporating or condensing phenomena. The heat transfer coefficients of R22, R134a, R507, R404a and R410a at two-phase zone increase with pressure. Values on the evaporator varies from 2000 to 4000 $W/m^2 \cdot °C$ and the heat transfer coefficients on the condenser from 8000 to 12000 $W/m^2 \cdot °C$ [10]. The model uses a constant global heat transfer coefficient (U) of 1000 $W/m^2 \cdot °C$ on the condenser and the evaporator heat exchanger, making the AU to be 35300 W/K on the evaporator and 21600 W/K on the condenser.

Table 1. Chiller data.

Evaporat. power [kW]	Condenser power [kW]	Maximum pressure Condenser [bar]	Electric motor [kW]	Displacement Flow [m^3/s]	Water flow Evaporat. [kg/s]	Water flow Condenser [kg/s]
240	293	22	60	0.0817	13.9	10

The model is highly coupled, what made the authors to decouple the equations fixing one of the loops [11]. The equations of the condensing water loop and the ones of the refrigeration loop are computed in an EES platform allowing to obtain a curve of the refrigeration power for each evaporating temperature. Because the cooling water flow is constant, it makes possible to assume the coil temperature at the AHU, which is set to be 2 °C higher than the superheat temperature leaving the evaporator. Linking the coil temperature to the superheat temperature allows solving the water cooling loop and decoupling it from the AHU loop. Given the coil temperature of the AHU, the outdoor and indoor thermo-hygrometric conditions and the outdoor airflow and supply flows, it is possible to compute the cooling power of the AHU. The intersection of the equation of the AHU cooling power with the mentioned equation of the refrigeration power allows obtaining the working conditions of the system as a whole and the evaporating temperature. The following equations summarize the method not showing the details of implementation to increase readiness.

The model uses the *Ɛ-NTU* method to describe the heat exchange in the condenser. Assuming the condenser is at a constant temperature the efficacy of the heat exchanger is:

$$\varepsilon = 1 - \exp(-NTU) \tag{1}$$

NTU is the number of thermal units. The isentropic efficiency of a reciprocating compressor [12] is on the average of the range of working pressure ratios set at 0.774. The volumetric efficiency of the reciprocating compressor [13] is computed by Equation 2 being r_p the pressure ratio between the suction and the discharge pressures:

$$\eta_v = 0.94 - 0.046 \cdot r_p \tag{2}$$

The pressure drop in the condenser and the evaporator in the two–phase region is a correlation of the pressure-drop “a” of the liquid fluid and the pressure drop “b” of the gas fluid in a tube of the same dimension, given by:

$$\left(\frac{dP}{dz}\right)_{frict} = \left(a + 2\left(b - a\right)x\right) \times \left(1 - x\right)^{1/3} + bx^3 \tag{3}$$

The superheat provided by the thermal expansion valve is 5 °C controlling the mass flow rate in the circuit. The mass flow rate m_f depends on the specific volume v_c of the gas entering into the compressor according to the following equation, where V is the displacement flow defined at Table 1.

$$m_f = V \cdot \eta_v / v_c \tag{4}$$

Regarding the psychrometrics of the AHU, the mixing point *M* is on the line linking the indoor (*RA*) and outdoor (*OA*) condition, at the temperature T_M:

$$T_M = T_{OA} \times 0,3 + T_{RA} \times 0,7 \tag{5}$$

The supply temperature T_{SA} occurs at point (*SA*) on the line that links *M* to the *coil* temperature T_{coil}:

$$T_{SA} = cf \times \left(T_M - T_{coil}\right) \tag{6}$$

Finally, Q_{total} is the heat removed by the AHU requiring the computation of the enthalpy *h* at the *SA* and *M*, using the air flux of mass supply m_{sp}:

$$Q_{total} = m_{SP} \times \left(h_{SP} - h_M\right) \tag{7}$$

4 Results

Table 2 shows the results of the model for each refrigerant. It shows the GWP, the pressure and temperature on the suction (1) and discharge (2) of the compressor, the pressure drop at the condenser (cond) and on the evaporator (evap). Moreover, it shows the cooling load (Q_{ref}) and the compressor needs (W_c). The table also shows the data of R404a because it has been used to retrofit R22 in refrigeration premises. All values in bold regards exceed the maximum working conditions regarding the motor power or the pressure at the condenser.

Table 2. Data from the refrigeration model at 35 °C condensing temperature.

Refrig. ***(GWP)***	P_1 **[bar]**	T_1 **[°C]**	ρ_1 **[kg/m³]**	P_2 **[bar]**	ΔP_{cond} **[bar]**	ΔP_{evap} **[bar]**	Q_{ref} **[kW]**	**Wc** **[kW]**
R22 *(1810)*	5,05	0,4	21,5	19,85	0,31	0,18	194	61,1
R404a *(3940)*	6,24	1,2	31,7	23,63	0,38	0,25	187	68,6
R407c *(1620)*	5,29	3,0	21,8	21,23	0,28	0,17	190	61,4
R134a *(1300)*	3,49	4,9	17,1	12,28	0,27	0,15	152	37,8
R410a *(1920)*	7,95	-0,1	30,4	32,32	0,21	0,15	206	77,8
R1234yf *(<1)*	3,73	5,0	20,8	12,17	0,35	0,18	148	37,5
R1234ze *(<1)*	2,92	6,9	15,1	8,887	0,27	0,14	128	28,1
R32 *(650)*	8,04	-0,3	21,9	32,98	0,13	0,09	210	74,2
R290 *(5)*	5,07	2,1	11,0	16,91	0,19	0,09	180	53,1

Table 2 shows that the chiller with R22 gas at extreme conditions exceeds the maximum power. It shows that the equipment with R407c or with R290 have a similar behaviour to the R22, both with slight reduction of the cooling power. R404a, R410a or R32 are not solutions to retrofit the R22, because the pressure at the condenser needs to increase too much. The compressor being unable to attain such pressure crash or makes the condenser not to be able to transfer the total amount of heat. In the last condition, the fluid goes out of the condenser at a two-phase state and leaves the evaporator with high superheat. The results also show the possible retrofit of R134a by R1234yf with a slight reduction on the refrigeration power. Moreover, R32 has similar data to the one of the R410a.

5 Discussion and conclusions

The results are in accordance to the market trends to achieve possible solutions to retrofit commercial gases. R290 (propane) can retrofit installations using R22 or R407c reducing the cooling power by less than 5%, but having the disadvantage of being flammable (A3). R32 is a possible solution to retrofit R410a being mildly flammable (A2L), but having a GWP of 650 will be a transition solution. Blends of R32 (HFC) with other gases may help extending the working life of equipment designed to work with R410a. Equipment formally supplied with R134a may use R1234yf (HFO) as a retrofit gas, a gas having classification of A2L. Therefore, blends of R1234yf or R1234ze and HFCs may produce gases for the coming years. As a result, all new retrofit gases have a PHE classification of A2L or A3. Blends of HFO gases with HFC gases can be a transition trade-off solution targeting to achieve an A1 gas with low GWP. Maybe, the future will be back with CO_2 and NH_3.

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