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OPTIMUM PERFORMANCE OF EXTERNAL INTERCOOLING TWO STAGE CYCLES WITH REAL COMPRESSOR CURVES

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ABSTRACT

Two stage cycles have been used traditionally in the case of high temperature difference between heat sink and heat source in order to overcome the high pressure ratios that deteriorate compressor volumetric and isentropic efficiency thus needing huge swept volumes for giving the required cooling capacity and increasing the power consumption.

Normally, intermediate cycle pressure is determined using geometric mean between evaporation and condensation pressure which gives optimum COP in the case of perfect gas assumption and same inlet vapor temperature to both compressors. These assumptions are not fulfilled in real conditions, so the objective of the paper is to study optimum intermediate pressures in two stage cycles with external subcooling.

A model for two stage cycles with external subcooling is developed that takes into account volumetric and isentropic efficiency of compressors dependent on the pressure ratio. Compressor curves are taken from general expressions provided by Pierre and are tested against actual compressor performance from manufacturer catalog.

First, COP dependence on intermediate pressure is calculated for standard low temperature conditions (-35ºC evaporation temperature, 40ºC condensation temperature) working with refrigerants R-404A and R-717 (Ammonia). Results show that optimum intermediate pressure is close to the arithmetic mean in the case of R-404A but there is a significant difference in the case of Ammonia. Also, it is compared the case of using constant compressor efficiencies vs. pressure ratio dependent compressor efficiencies. Results show that there are higher differences in the determination of optimum intermediate pressure,

Next, the model is used in order to calculate optimum intermediate pressure dependent on evaporation and condensation temperature with a constant superheat and subcooling of 10 K and 5K respectively. Contour surface maps and analytical expressions giving optimum intermediate pressure dependent on evaporation and condensation temperature are presented. Results show that there is less than 1K equivalent saturation temperature difference between optimum intermediate pressure and geometric mean pressure for R-404A in all the map; for ammonia there are significant differences that depends mainly on the evaporation temperature.

However, comparison between COP calculated at optimum intermediate pressure and at geometric mean pressure shows very low differences (less than 2%) for both refrigerants and therefore validating the use of the simple geometric mean equation for this type of cycles. Second main conclusion of this paper is that significant errors can be made when calculating optimum conditions in two stage cycles when using constant compressor efficiencies as used by many authors in the literature.

1. INTRODUCTION

Traditionally, one of the improvements to the vapour compression cycle was the use of two stage compression. The use of two stage cycles is justified when the temperature differences between the heat sink and heat source are high; the compressor is working at high pressure ratios that deteriorate its volumetric and compressor efficiency thus needing huge swept volumes for giving the required cooling capacity and increasing the power consumption. Using two stage designs with two compressors make that each compressor is working at a reduced pressure ratio and giving good values of volumetric and compressor efficiency. Normally, the use of two stage is restricted to refrigeration at low temperature conditions

(Granryd et al. 2003), although there are studies in the last years applying the concept to heat pump applications working at low Temperatures (Bertsch and Groll 2008) and in supercritical cycles using R-744 (Carbon Dioxide) (Cavallini et al. 2005), (Cecchinato et al. 2009).

The designer of a two stage cycle has to decide the working intermediate pressure given cooling capacity, temperatures of the heat sink and heat source, superheat and subcooling of the cycle in order to optimize the power input to the system (thermodynamic optimization) or the total cost of the system including first cost and operation cost (thermoeconomic optimization). The normal practice for thermodynamic optimization is to use an intermediate pressure corresponding to the geometrical mean between evaporation pressure and condensation pressure, $p_{opt} = \sqrt{p_{evap}\, p_{cond}}$ (GMP criteria). This expression is based on the assumptions of perfect gas, ideal compressors and inlet temperature to the High Pressure (HP) compressor equal to the Low Pressure (LP) compressor.

There are several authors that have investigated the optimum intermediate reaching to different conclusions. (Arora and Dhar 1971) used the discrete maximum principle, a mathematical tool described by (Katz 1962) to minimize compression work optimizing the intermediate pressure for ammonia using perfect gas relationships. Results show that optimum intermediate pressure is near geometrical mean pressure in the case of external intercooling to the saturation temperature between stages and higher optimum intermediate pressure than geometrical mean pressure for the case of internal flash intercooling.

(Kueh, Ramsey, and Threlkeld 1998) used gas real properties for a two stage cycle working with ammonia and found that optimum intermediate pressure differs from the geometrical mean criteria.

(Czaplinski 1959) studied optimum interstage pressure in an internal flash intercooling two stage cycle working with ammonia using real gas equations and constant efficiencies of the compressor, concluding that optimum is giving by geometric mean of saturation temperatures, $T_{opt} = \sqrt{T_{evap}\, T_{cond}}$ (GMT criteria).

(Prasad 1981) criticized the work of (Arora and Dhar 1971) because they only minimises compression work not taking into account the cooling effect loss in the evaporator due to vapour injection in the flash intercooling cycle. An optimization of flash intercooling cycles working with R-12 is made using compressor efficiencies dependent on pressure ratio. The result of the optimization is a formula that gives optimum intermediate pressure dependent on evaporation and condensation temperature.

(Behringer 1928) studied the cycle with external intercooling between stages and concluded that the optimum intermediate pressure is given by adding 5 K to the saturation temperature of the geometrical mean pressure, $T_{opt} = T_{GMP} + 5$ (BEH criteria). The analysis is made for an ammonia cycle and using ideal compressors.

(Zubair, Yaqub, and Khan 1996) searched the optimum intermediate pressure in two different cycles working with R-134a: flash intercooling cycle and mechanical subcooling cycle using real gas properties and constant compressor efficiencies. It is found that optimum pressure is higher than intermediate pressures used by other authors cited in the literature, the nearest pressure is the arithmetical mean of temperatures followed by Behringer´s criteria, geometric mean of temperatures and geometric mean of pressures. They also found that maximum COP in an actual cycle with 65 % compressor efficiency, 20ºC of return gas temperature and 3K subcooling is produced at higher intermediate pressures than in an ideal cycle with isentropic compressor and null superheat and subcooling. From the analysis, it is concluded that differences between both cycles in term of optimum COP are given by subcooling and superheat with no effect of the compressor efficiency.

(Ouadha et al. 2005) used exergetic analysis to calculate the performance of two stage flash intercooling cycles working with ammonia and propane with constant compressor efficiency and evaporation temperature. They concluded the optimum interstage pressure is close to the AMT criterion, $T_{sat,\mathrm{int}} = \dfrac{T_{evap} + T_{cond}}{2}$.

(Cecchinato et al. 2009) studies the thermodynamic evaluation and optimization of five different two-stage carbon dioxide cycles (basic single stage, single throttling with two stage compression, split cycle phase separation cycle and single stage coupled with a gas cooling circuit. Real compressor curves are used for the calculations based on previous experiments and catalogue data. Optimal values of upper and intermediate pressure are calculated dependent on evaporation pressure.

It is clear from the authors cited that GMP criteria, valid for perfect gas and ideal compressor is not valid when used real gas properties, and dependent on the refrigerant and the two stage cycle used there are different criteria for the optimum intermediate pressure. However, all the authors except (Prasad 1981) and (Cecchinato et al. 2009) used ideal or constant efficiency compressors; that is an important simplification of real compressors. Then, the objective of this paper is to study the influence of the real compressor performance in the determination of the optimum intermediate pressure for external intercooling two stage cycles and motivate to answer the following questions: What is the optimum intermediate pressure for a two stage cycle with external intercooling?, How it depends on the cycle working conditions?, Is it accurate to use criteria used by other authors? and Has real compressor behavior influence in the optimum?.

2. MODEL DESCRIPTION

The system analyzed is a double stage cycle with external subcooling as shown in Figure 1. It is assumed that intercooling is made with the heat sink fluid used in the condenser, so there are two possible intercooling modes:

1. Outlet temperature of the LP compressor higher than the condensation temperature (T2>Tcond): This is the normal case, intercooling makes that inlet temperature to the HP compressor (T3) matches the condensation temperature.
2. Outlet temperature of the LP compressor lower than the condensation temperature: It happens when intermediate pressure is low. If refrigerant is put in contact with the coolant, there will be intermediate heating instead intercooling and will be harmful for the COP. What is considered in this case is that there is no heating of refrigerant, so inlet temperature to the HP compressor will be the same as the outlet temperature of the LP Compressor.

Refrigerants considered in the study are R-717 (Ammonia) and R-404A due they are the most used now in low temperature refrigeration applications where two-stage cycles can take advantage.

The model used is a tool written in FORTRAN language implemented in IMST-ART software (J.M. Corberán 2010) developed by the authors that allow the calculation of two Stage Cycles with External Subcooling.

The model is made up of a set of equations based on mass and energy balances, evaluating the properties of the refrigerant as a real fluid. Refrigerant properties are calculated using subroutines provided by REFPROP (E.W., M.L., and M.O. 2007).
Figure 1 describes the system analyzed and the representation on a pressure-enthalpy diagram.

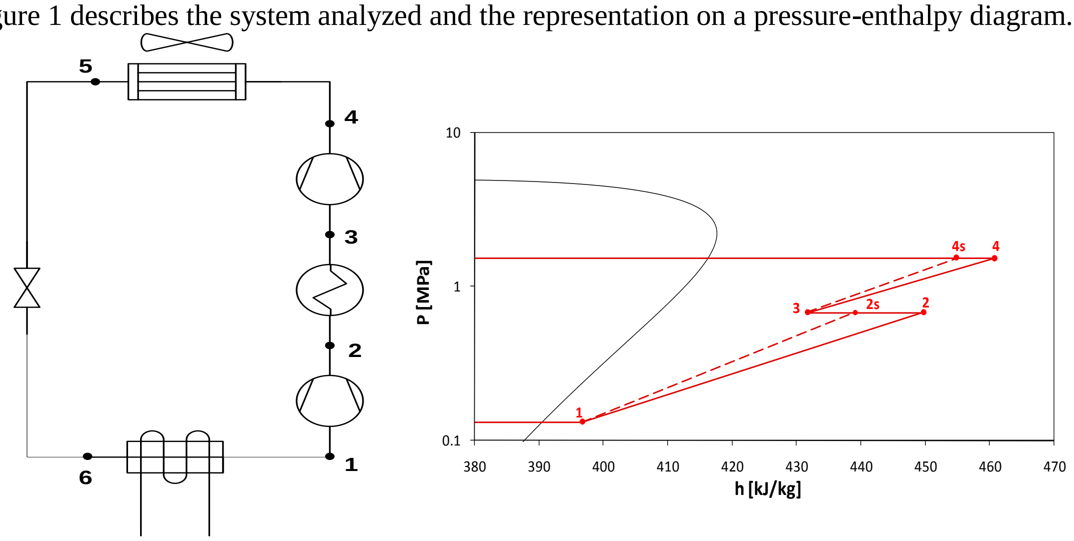

Figure 1 Two stage cycle with external subcooling and evolution in a pressure-enthalpy diagram

The set of unknown variables for the problem are pressure and enthalpy in each node P_i, h_i (except intermediate pressure, P_i) the circulating mass flow rate $\dot{m}$ and the displacement of LP and HP compressors $\dot{V}_{LP}$ and $\dot{V}_{HP}$.

Boundary conditions of the problem are the cooling capacity $\dot{Q}$, evaporation temperature T_{ev} , condensation temperature T_{cond} , intermediate pressure P_{int} , superheat SH and subcooling SC .

Low pressure compressor performance is considered with its isentropic efficiency.

$$\eta_1 = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

Intercooler outlet is defined with its enthalpy and pressure:

$$h_3 = f(P_3, T_3) \tag{2}$$

$$P_3 = P_{\text{int}} \tag{3}$$

After intercooling, second compressor outlet is calculated as:

$$\eta_2 = \frac{h_{4s} - h_3}{h_4 - h_3} \tag{4}$$

While the removed energy per unit mass in evaporator is calculated as:

$$\Delta h_{evap} = h_1 - h(P_4, T = T_{sat}(P_4) - SC) \tag{5}$$

Mass flow rate can be calculated as:

$$\dot{m} = \frac{\dot{Q}_{evap}}{\Delta h_{evap}} \tag{6}$$

And finally, compressors consumption are calculated as:

$$\dot{E}_1 = \dot{m}(h_2 - h_1) \tag{7}$$

$$\dot{E}_2 = \dot{m}(h_4 - h_3) \tag{8}$$

Finally, COP is obtained.

$$COP = \frac{\dot{Q}_{evap}}{\dot{E}_1 + \dot{E}_2} \tag{9}$$

Intercooler can works in two operation modes:

If $T_2 > T_{cond} : T_3 > T_{cond}$

If $T_2 \leq T_{cond}$

This mode operation will cause an incorrect behaviour of the facility because of intercooler is heating refrigerant instead of cooling down, consequently, the COP decreases. So, it is decided to model the situation that at this time intercooler does not work.

$$T_3 = T_2$$

Compressors are assumed adiabatic and its volumetric and isentropic efficiency are calculated using the expressions proposed by Pierre (Granryd and Department of Energy Technology 2002):
The expressions are valid for open compressors and for given refrigerants not including R-404A. In order to use these expressions with semihermetic or hermetic compressors working with R-717 and R-404A and adjustment factor of F=0.89 for the compressor efficiency has been adopted not requiring any adjustment in the case of volumetric efficiency. Adjusted Pierre´s expressions have been tested against different actual commercial compressors correlating results within 10% of maximum error for both refrigerants.

3. RESULTS

The studies are performed with an evaporation temperature of -35 ºC, condensation temperature of 40 ºC, 5K subcooling and 10 K superheat. Refrigerants considered in the study are R717 (Ammonia) and R404a. The cooling capacity of the system is 1 kW, i.e. all the studies have been made per kW of cooling capacity.

Figure 2 shows COP variation of the two stage cycle when intermediate pressure is varied. Results are obtained for both refrigerants and under different assumptions for compressor efficiency: ideal performance with 100% volumetric and isentropic efficiency, constant efficiencies of 60% and 50% for volumetric efficiency and isentropic efficiency and real performance using adjusted Pierre´s expressions. Figure also shows Relative COP calculations, i.e. ratio of the two stage cycle COP by single stage COP. Also are plotted the points corresponding to different criteria used by other authors, AMT (Arithmetic Mean Temperature), GMT (Geometric Mean Temperature), GMP (Geometric Mean Pressure) and BEH (criteria used by Behringer, add 5K to the GMP criteria).

Analyzing curves with ideal efficiency compressor for both refrigerants, it is shown that a first section where COP is constant and a second section with variation reaching to a maximum. Existence of two sections in the curve are due to the option to intercool the refrigerant depending on the LP compressor outlet temperature. First section corresponds to the case where LP outlet temperature is lower than condensation temperature; no intercooling is applied and then isentropic line of HP compressor follows the same line of LP compressor, the cycle plotted in a pressure-enthalpy diagram is the same independent on the intermediate pressure. Also, COP is exactly the same as the single stage cycle as shown in the relative COP curves. On the other hand, when LP outlet temperature is higher than condensation temperature, intercooling is applied and therefore there is an enhancement of COP compared with single stage due to the higher slope of the HP compressor isentropic line in a P-H diagram.
The constant efficiency curve shows negative performance in the section with no intercooling; this strange result is explained as the LP compressor discharge temperature increases with intermediate pressure, so increasing the inlet entropy to the HP compressor and increasing its power input. It is remarkable that the behavior of the constant efficiency curve is exactly the same (shown in the relative COP plot) as the ideal efficiency, so no improvement in the selection of optimum intermediate pressure is made when introducing compressor constant efficiencies in an ideal cycle calculation.
Finally, real efficiency curve shows a different behavior. There are two sections overlapped as the previous curves but the optimum intermediate pressure changes. Relative COP is higher than constant and ideal efficiencies curves as both compressors are taking advantage of lower pressure ratios. COP curve for R-

404A is quite flat, so different criteria proposed has little differences in the optimum; however, R-717 shows a high influence of intermediate pressure on COP but different optimum criteria are near the maximum.

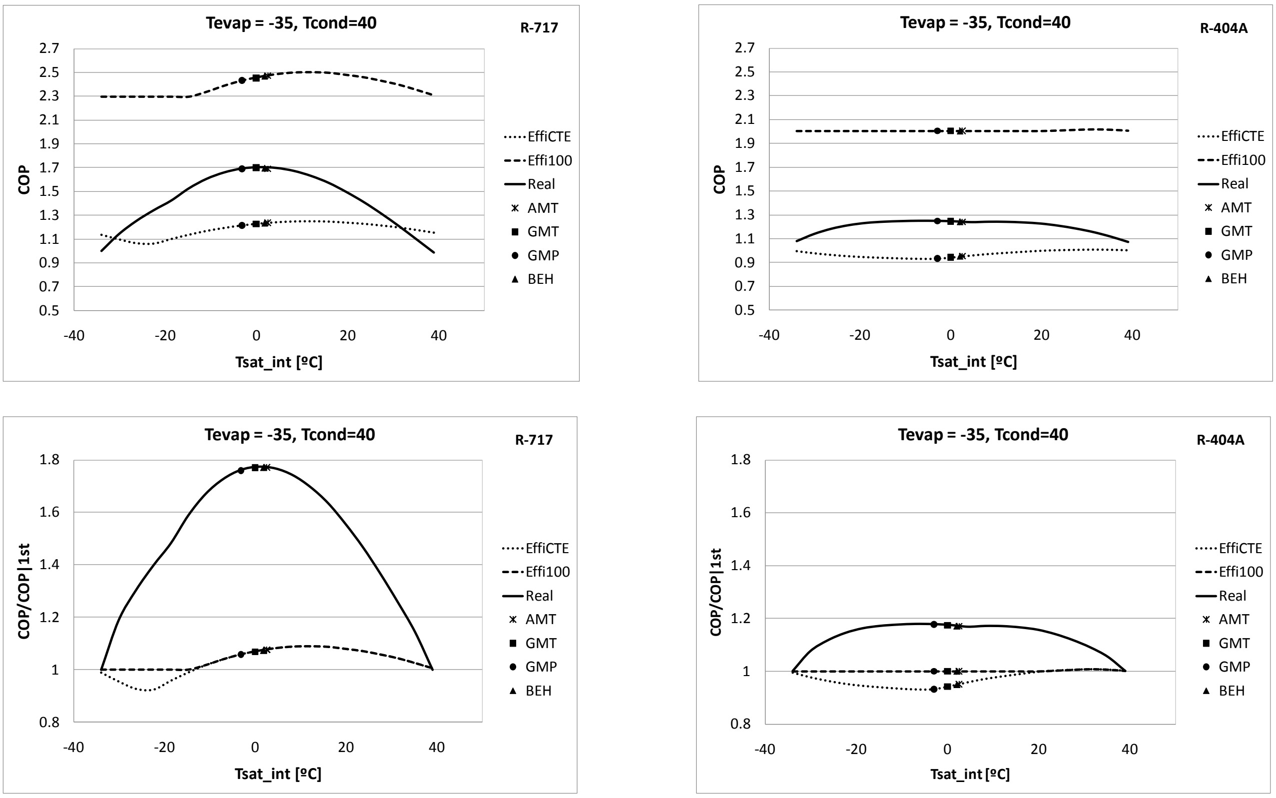

Figure 2 Influence of the intermediate pressure on two stage cycle COP

Next question to solve is how changes the optimum intermediate pressure (or saturation temperature) with changes in evaporation and condensation temperature for the case of real efficiency curves. An optimization study for the intermediate pressure has been made using commercial software EES (Klein 2010), results in terms of intermediate saturation temperature have been fitted to a polynomial giving the following values:

R-717

$$Tint_{opt} = 11.62 + 0.46 \cdot Tcond - 6.93 \cdot 10^{-4} \cdot Tcond^2 + 0.74 \cdot Tevap + 1.76 \cdot 10^{-3} \cdot Tcond \cdot Tevap + 1.78 \cdot 10^{-4} \cdot Tevap^2$$

(10)

R-404A

$$Tint_{opt} = -23.84 + 7.23 \cdot 10^{-1} \cdot Tcond - 1.3 \cdot 10^{-3} \cdot Tcond^2 - 1.35 \cdot 10^{-1} \cdot Tevap + 5.61 \cdot 10^{-3} \cdot Tcond \cdot Tevap - 5.62 \cdot 10^{-3} \cdot Tevap^2$$

(11)

The temperatures in the fitting equations are expressed in ºC and the fitting is valid for the following conditions: $Tcond \in [35,50]$ ºC $Tevap \in [-55,-35]$ ºC SC = 5 ºC SH = 10 ºC

Polynomials are plotted in Figure 3. It is shown a quite linear trend of the optimum intermediate saturation temperature with condensation and evaporation temperature. This linear dependence is also shown in the low values of coefficients that multiply quadratic terms.

Differences between the optimum intermediate temperature and optimum given by the GMP criterion are also correlated by the following polynomial fitting:

R-717

$$Dif = 11.59 - 1.98\cdot10^{-2}\cdot Tcond - 3.6\cdot10^{-5}\cdot Tcond^2 + 2.5\cdot10^{-1}\cdot Tevap - 2.44\cdot10^{-4}\cdot Tcond\cdot Tevap + 1.57\cdot10^{-3}\cdot Tevap^2 \quad (12)$$

R-404A

$$Dif = 140.39 - 3.07\cdot Tcond + 2.03\cdot10^{-2}\cdot Tcond^2 + 2.95\cdot Tevap - 2.69\cdot10^{-2}\cdot Tcond\cdot Tevap + 1.8\cdot10^{-2}\cdot Tevap^2 \quad (13)$$

Where Dif is the difference between optimum intermediate saturation temperature and temperature given by GMP criterion in ºC, evaporation and condensation temperature are formulated in ºC.

Results of these differences are shown in Figure 4. Results for R-717 show that differences do not depend on the condensation temperature chosen; differences increases with higher evaporation temperature and its value is between 2.2 ºC and 4.4 ºC. Results for R-404A shows dependence on both evaporation and condensation temperature; differences increases with higher evaporation temperatures and decreases with higher condensation temperatures.

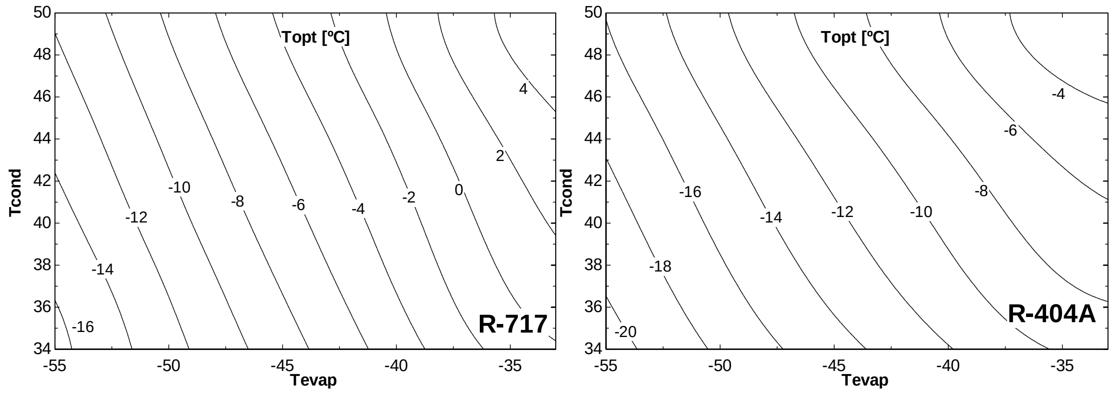

Figure 3 Optimum Intermediate Saturation Temperature Map

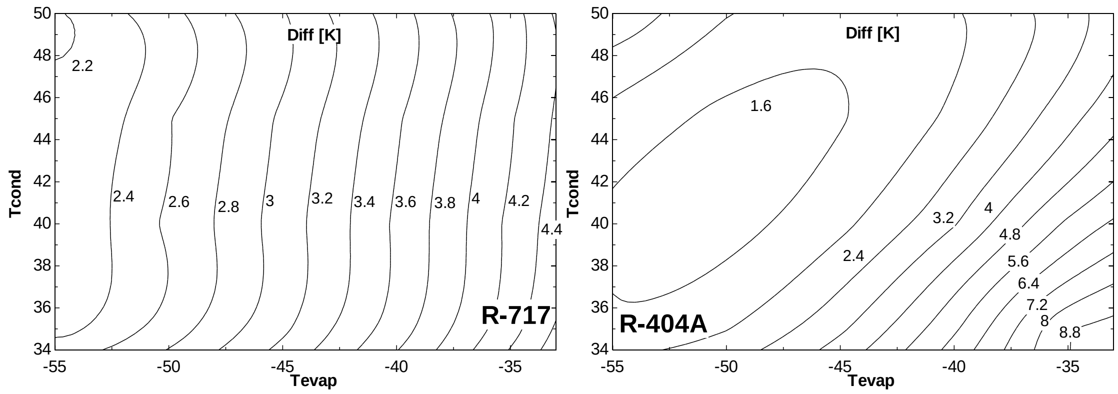

Figure 4 Saturation Temperature difference between optimum and GMP criterion

3. CONCLUSIONS

These conclusions can be drawn from the research:

- It is needed to introduce real compressor curves to know the optimum intermediate pressure in two stage cycles. Introducing constant efficiencies in the analysis does not give better accuracy or knowledge about cycle than using isentropic compression.
- Optimum intermediate saturation temperature determined for R-717 at standard low temperature conditions is very close to the different criteria proposed by other authors. Differences in COP are very low for all the points.
- Optimum intermediate saturation temperature determined for R-404A at standard low temperature conditions is higher than different criteria proposed by other authors. Differences vary between 3 ºC and 8 ºC. Nevertheless COP curve is quite flat, so COP differences at the end are very low.

* Optimum intermediate saturation temperature depends on evaporation and condensation temperature, a polynomial fitting is presented in order to know optimum for refrigerants R-717 and R-404A.

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