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CONTROL OPTIMIZATION OF TRANSCRITICAL CO_2 AIR CONDITIONING SYSTEMS OPERATING IN SUBTROPICAL CLIMATE

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ABSTRACT

For vapour compression cycles where, heat rejection occurs under transcritical pressure, an optimum pressure that maximizes the COP exists. The most common application of this situation are CO_2 transcritical systems. The paper presents two different air conditioning systems operating in subtropical conditions, which have different control techniques to set the gascooler pressure to its optimum value. Simulation studies has been carried out in order to get the optimum gas cooler pressure for each system, to compare differences of both systems and for analysing the COP improvement of these systems respect to the baseline split air conditioning system without pressure control. The simulated values of the optimum gascooler pressure for both systems were compared with other correlations of literature, showing different deviations depending on the analysed system.

Keywords: transcritical, optimum gascooler pressure, EEV, COP

1. INTRODUCTION

Energy, environment and economics are the driving forces of contemporary refrigeration and air conditioning business. The interruption of man-made chemical refrigerants through Montreal and Kyoto Protocol has raised the need to find and test alternative sustainable refrigerants for wide range of weather conditions. Major research on the theoretical and experimental performance studies of transcritical CO_2 refrigeration systems covering application range from automobile air conditioning to air dries mostly for the ambient temperatures less than 35^0C. For the transcritical CO_2 system operating above the critical temperature of CO_2, the gascooler pressure is independent of the temperature and the isotherms have ‘S’ shaped orientation in the supercritical region, results in, the non-monotonic variation of COP with the gascooler pressure. The main goal of the present work is to evaluate, with simulation studies, the performance improvements when there is a control of the gas cooler pressure, to keep it equal to the optimal pressure, respect a system which does not have any control of the gas cooler pressure.

Liao et al. (2000) have proposed a correlation for optimum gascooler pressure in transcritical CO_2 split air conditioning system using the theoretical thermodynamic models developed in Engineering Equation Solver (EES). The research concluded that the optimum heat rejection pressure of the gascooler depends on the evaporator temperature, gascooler refrigerant outlet temperature and the compressor performance. Sarkar et al. (2004) have formulated the correlation for the optimum heat rejection pressure in terms of the evaporator temperature and gascooler outlet temperature neglecting isentropic efficiency of the compressor for the water-to-water combined heating and cooling transcritical CO_2 system. The evaporator temperature and gascooler outlet temperature varied between -10 and 10^oC and 30 to 50^oC respectively.

The researchers observed that for fixed evaporator saturation temperature, the decrease in the gascooler outlet temperature decreases the optimum gascooler pressure and sharply enhances the system COP. Chen and Gu (2005) investigated the relation of optimum gascooler pressure with ambient temperature (30 to 50^{o}C), suction line heat exchanger effectiveness (0 to 1) and evaporator saturation temperature (-10 to 10^{o}C) for transcritical CO_2 refrigeration system. Cho et al. (2007) have studied the opening of Electronic Expansion Valve (EEV), length of SLHX, compressor frequency and refrigerant charge on the cooling COP (Cho et al., 2007). The optimum refrigerant charge was determined for maximum COP at standard cooling test conditions. The paper recommends retaining balance between EEV opening and compressor frequency to maintain optimum gascooler pressure and suction superheat. Cabello et al. have presented an experimental study of refrigeration plant for maximum energy efficiency and gascooler pressure for subzero evaporator temperatures and different gascooler outlet temperatures and developed the better correlation for optimum gascooler pressure over the Lio, Sarkar, Kauf and Cheng and Gu correlations (Cabellp et al., 2008). They observed inclusion of theoretical model for the compressor by Liao and Sarkar have given close prediction of optimum gascooler pressure to experimental results. Ge and Tassou have studied the CO_2 cycle for medium temperature food retail refrigeration applications (Ge et al., 2009). The optimum pressure and approach temperature of the gascooler are depends on the ambient temperature. The approach temperature of the gascooler is possible to keep constant by changing the air velocity over the gascooler using variable frequency drive fans for high ambient conditions. The research has proposed the correlation for optimum gas pressure as function of ambient temperature. At the evaporating temperature of 10^{o}C, when the gas cooler refrigerant exit temperature increases from 33^{o}C to 45^{o}C, the optimum heat rejection pressure increases by 23.06%. These authors have further corrected the Lio correlation of optimum gascooler pressure for deviations in evaporator and gascooler exit temperature. Aprea and Maiorino (2009) performed an experimental optimization of gascooler heat rejection pressure for a CO_2 split air conditioner for ambient temperatures 25 - 35^{o}C (Aprea et al., 2009). They have modified Liao correlation for determining optimum heat rejection pressure. The authors have tested a Manual Back Pressure Valve (MBPV) with Electronic Expansion Valve (EEV) in series arrangement to have precise control on the cooling capacity and proposed use of Electronic Back Pressure Valve. The effect of suction superheat on the optimum gascooler pressure is investigated by Zhang et al. considering water-to-water heat pump (Zhang et al., 2010). The researchers have considered: semi-theoretical model of the compressor, two stage series throttling, zero pressure drop in the components and their connecting lines and studied the system for evaporator temperature range of 10 to 20^{o}C and gascooler outlet temperature range 33 to 45^{o}C. This research work confirms deviation of 0.397% in optimum gascooler pressure for a suction superheat difference of 15^{o}C.

This research is focused on the comparative study of two different systems with the baseline system for the COP improvement and optimal gascooler pressure for typical subtropical conditions observed in India. These systems have application in split air conditioning module. The research in the open literature has studied systems like one of the presented systems for obtaining a correlation for optimal gascooler pressure. The paper presents the results for COP sensitivity of both systems for indoor WBTs and outdoor DBTs and compared the results for optimal gascooler pressure with correlations developed by Sarkar et al. (2004) and Aprea and Maiorino (2009).

2. SYSTEMS DESCRIPTION

The paper studies the performance of two systems, depicted in Fig. 1, for a wide range of indoor/outdoor conditions. Each system has a different regulation methodology to control the gas cooler pressure, whose advantages and disadvantages are described below.

The baseline CO_2 air conditioning system, which does not have any control, consists of: a reciprocating-hermetic compressor; a gas cooler of finned tubes; an internal heat exchanger (IHX); a thermostatic expansion valve; and an evaporator of finned tubes. The two system analysed are based on the baseline system, and both of them use the same components, excepting the electronic expansion valve, and the receiver at the compressor inlet that is only in the system B. The design of these components has been carried out for IS 1391 Part 1 (1992) standard conditions: indoor DBT 27^{o}C, WBT 19^{o}C and outdoor DBT 35^{o}C, WBT 24^{o}C using the software IMST-ART (Corberán et al, 2002). This software is a tool for simulation of any vapour compression system, which will be introduced in next section. Table 1 contains a summary of the data used to define all the components.

Finally, to define completely the baseline cycle, two additional parameters are needed: refrigerant charge and evaporator superheat. These values were determined using IMST-ART for the standard conditions as follows: a temperature approach equal to 2^{o}C was imposed at the gas cooler outlet, and then the superheat was varied until maximum COP was reached. The results were: a refrigerant charge for the system of 0.143 kg, and a temperature superheat of 7 ^{o}C for the thermostatic expansion valve.

The baseline system will work with a fixed refrigerant charge and superheat, so the gas cooler pressure will vary when outdoor/indoor conditions change, resulting a different pressure from the optimal pressure, due to no existence of control. The systems with control on the gas cooler pressure, analyzed in this paper, are described below.

Table 1. Description of baseline split air conditioner

Evaporator			
Tube Length [mm]	550	Height [mm]	180
Depth [mm]	50	Width [mm]	500
Number of tube rows	4	Tube diameter [mm]	4.75
Number of tubes per row	10	Type of fin	Louvered
Number of circuits	5	Fin spacing [mm]	1.37
Gascooler			
Tube Length [mm]	550	Height [mm]	375
Depth [mm]	36	Width [mm]	700
Number of tube rows	2	Tube diameter [mm]	4.75
Number of tubes per row	15	Type of fin	Louvered
Number of circuits	5	Fin spacing [mm]	1.37
Compressor			
Maker and Model	Danfoss, TN1416	Displacement capacity [cm^3]	2.46
Internal Heat Exchanger (IHX)			
Overall Thermal Resistance (1/UA) [W/K]	4		
Connecting tubes/lines			
Pipe lines length [mm]	1000		

2.1. System A

The main characteristic of this system is that it has a fixed refrigerant charge. System A consists of the same components as baseline system, the only difference is the expansion valve. System A has an electronic expansion valve, which controls the superheat at the evaporator outlet. Since the system has a fixed refrigerant charge, varying the superheat, for specific indoor/outdoor conditions, the gas cooler pressure will change. Thus the electronic expansion valve allows keeping the gas cooler pressure at its optimal value by controlling the superheat, when the indoor/outdoor conditions change.

The main advantages of this system are a lower cost than system B, since it does not have the receiver and a total refrigerant charge in the system lower than system B. On the other hand the charge is uniform so it is not suitable for reversible systems because for heat pump mode it would have a worse performance.

2.2. System B

System B consists of the same components as System A, but it includes a receiver at the evaporator outlet and uses an electronic expansion valve to control the gas cooler pressure instead of the superheat. The receiver imposes that the refrigerant at the inlet of the IHX is always saturated vapour. In these conditions, in order to fix a gas cooler pressure for different indoor/outdoor conditions, a variable refrigerant charge in the system is necessary. It is the function of the receiver in the system. The main advantage of this system is the presence of this component, because it allows having in the system a non-uniform refrigerant charge in the circuit, so that this system can be used as reversible cycle.

Some disadvantages of this system are: possible oil build-up in the receiver and the cost of an additional component (receiver).

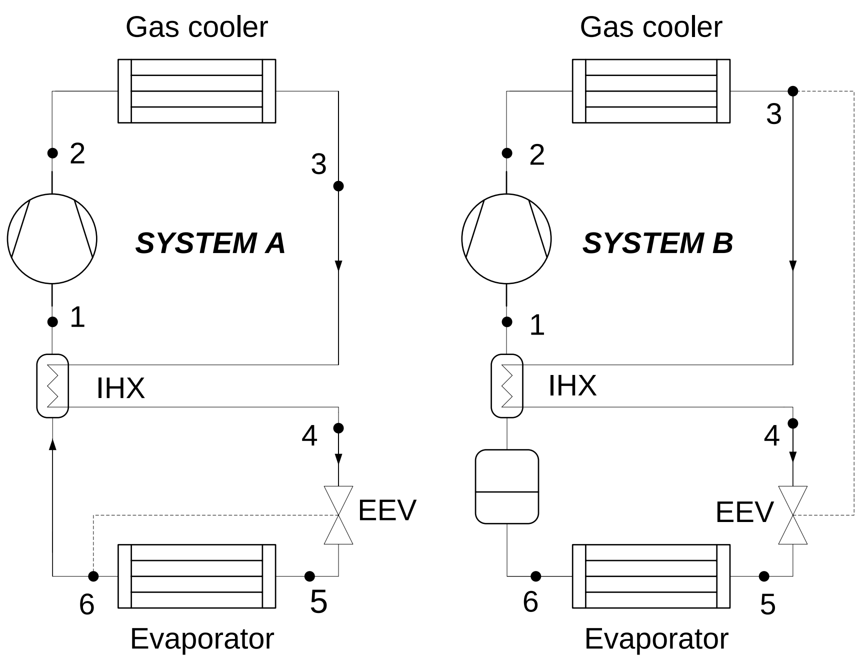

Figure 1. Schematics of the two systems studied

3. SIMULATION STUDIES

The simulations of the system A and B are performed for the forty two sets of combinations of six indoor WBTs and seven outdoor DBTs observed in the subtropical conditions. The ranges of the indoor WBTs and outdoor DBTs considered are 18 to 23°C and 33 to 45°C respectively. The indoor DBT was always kept to 27 ºC.

The simulation tool used for this work is IMST-ART. It is software for modelling vapor-compression refrigeration systems, allowing also modelling each of the components such as compressor, suction line heat exchanger, valve expansion... with a large variety of models and accuracy. It uses a detailed segment-by-segment model for the heat exchangers.

The first studies correspond to analyse the performance of each system when the gas cooler pressure varies. This study allows knowing the sensibility of each system to the gas cooler pressure variation, consequence of variation of the indoor/outdoor conditions when there is no control. Fig. 2 presents the COP obtained when the gas cooler pressure is changed in scenarios with different outdoor temperature values. The indoor WBT was equal to 19 ºC and the indoor DBT was 27 ºC.

In Fig. 2, it is noticeable that the System A is more sensible than System B to variations on the gas cooler pressure, since its slope near the maximum COP is more flat than for system A. This fact indicates that a system such as B is more suitable than system A for applications with large indoor/outdoor variations. The System A, would need a control system, otherwise the COP could suffer large detriments.

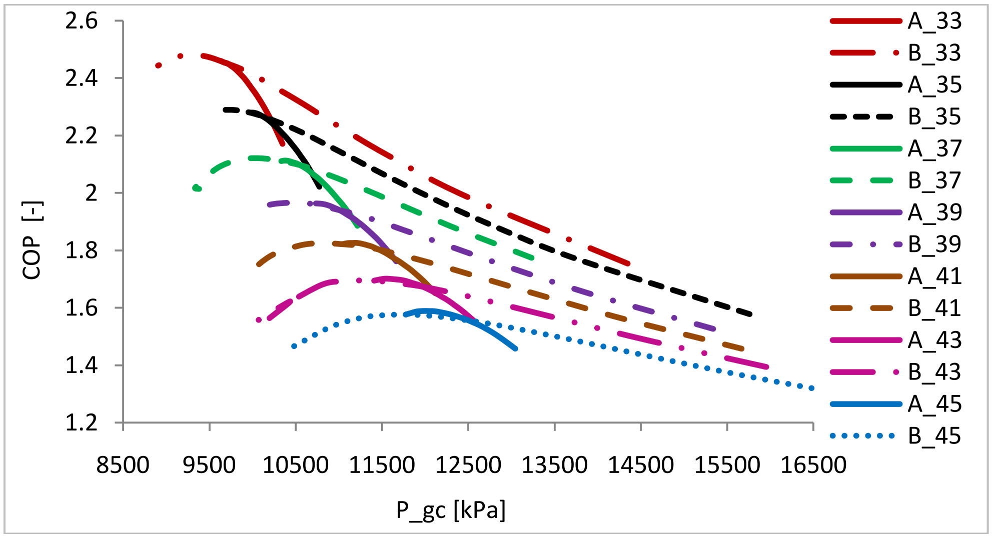

Figure 2. Influence on COP, in systems A and B for different values of outdoor DBT.

3.1. System A results

The optimum gas cooler pressure for this system is plotted in Fig..3 (a) as function of indoor WBT and outdoor DBT. The figure depicts that indoor WBT has very negligible influence on the optimum gas cooler pressure. The optimum gas cooler pressure is increasing with increase of outdoor DBT irrespective of any value of indoor WBT.

The COP improvement has been defined as the relative difference between the COP of the controlled system and the system without control working in same conditions. The COP for the system without control has been obtained in the baseline system for each indoor/outdoor condition. This parameter has been studied for system A and the results are plotted in Fig. 3 (b). At high WBT and low DBT, %COP improvement for system A is maximum (around 5.5%) over the baseline system. The COP values of system A working in such a conditions were ranging from 1.5% to 2.75%.

3.2. System B results

In system B, the EEV controls the gas cooler pressure acting directly on that pressure. Fig. 4 (a) shows the optimum pressure in system B, for outdoor and indoor temperatures ranging in the values described above. The trend is quite similar to the system A, the pressure has a negligible dependence with the WBT, and the greater the DBT is the larger the optimum pressure is located. The values of this pressure are also similar to those showed for system A, though for system B are slightly lower.

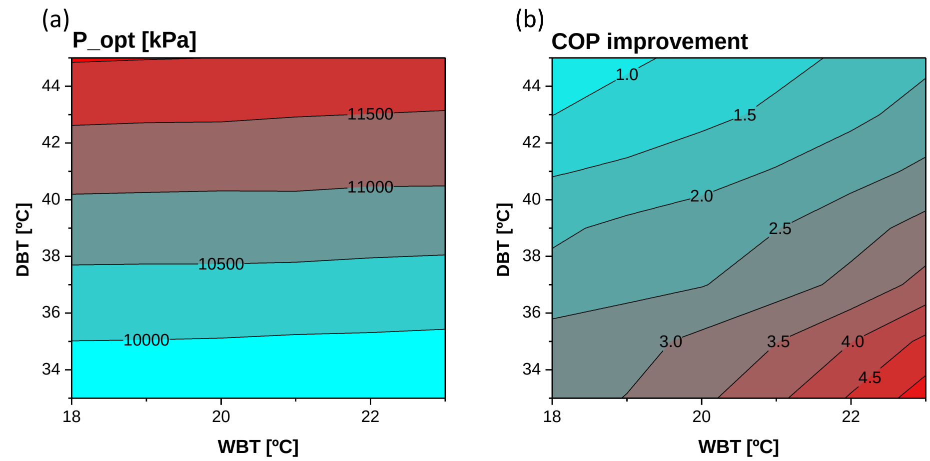

Figure 3. Optimal gas cooler pressure for system A (a) and improvement of COP in system A (b) for different values of outdoor DBT and indoor WBT.

The main difference, regarding the thermodynamic cycle, between both systems is that the system A has a superheat, at the evaporator outlet, different from the system B, which is zero. It can be seen the differences in values of optimal pressure are very small. Thus, this fact could mean that the influence of the superheat has a negligible effect on the gas cooler optimum pressure as it has been reported by several authors (Zhang et al., 2010; Aprea and Maiorino, 2009; Liao et al., 2000).

Fig. 4(b) shows the COP improvement achieved by the system B with respect to the baseline system, as it was explained in the description of system A results. The trend is quite different from the system A, now it is more independent from the WBT, excepting for low DBT where dependence on WBT appears. However, the values of improvement are almost equal for both systems. The improvement is larger for low DBT, with values up to 5.54 %. The COP of the system B ranged from 1.5 % to 2.75 %.

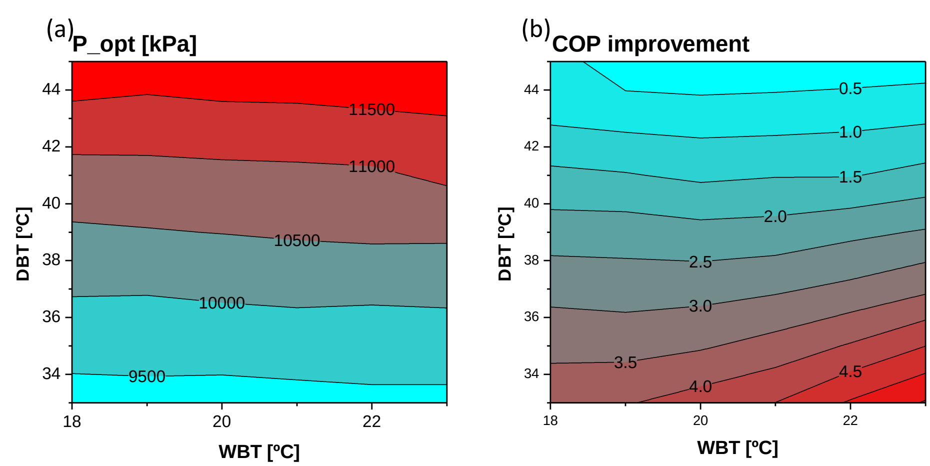

Figure 4. Optimal gas cooler pressure for system B (a) and improvement of COP in system B (b) for different values of outdoor DBT and indoor WBT.

3.3. Optimisation of gascooler pressure

The optimal gascooler pressure depends on the performance of individual components and behaviour of all the components together in the system. This performance depends on the boundary conditions of the system. Many authors have analysed boundary conditions such as evaporator superheat, compressor efficiency and gascooler outlet temperature. Fig. 5(a) and (b) gives comparison for the optimum gascooler pressure with Sarkar's and Aprea's correlations for different evaporator temperatures (Sarkar et al., 2009; Aprea and Maiorino, 2009), where the optimum gascooler pressure is plotted as function of the gascooler outlet temperature for different evaporation temperatures (for a gas cooler outlet temperature given, the optimum pressure increases as the evaporation temperature decreases). The average deviations for optimum gascooler pressure between Aprea and Sarkar correlations with the simulated results are 4.4% and 7.51% respectively for system A. These values for system B are 1.3% and 4.3% respectively for entire range of subtropical conditions considered in this study. The study of Fig. 5(a) and (b) reveals that simulated optimum gascooler pressure is very much close to referred author's correlations in the system B as compared with system A. Aprea and Maiorino (2009) developed their correlation for a system just like system B, with a liquid receiver and without evaporator superheat. Since the main difference between system A and B is the presence of a superheat at the evaporator outlet, this fact turns out as a possible reason for the differences between A and B results.

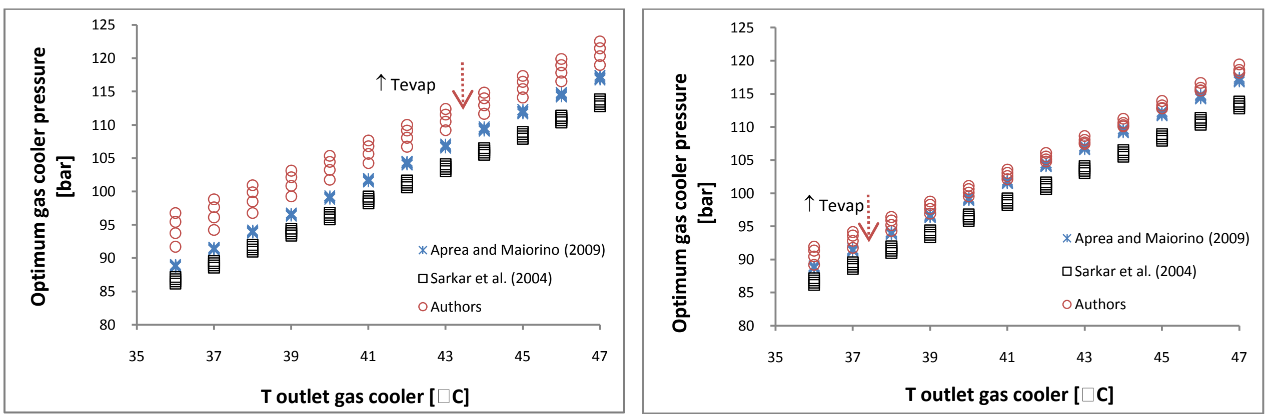

Figure 5. Optimum pressures for (a) system A and (b) system B against evaporator temperature

Another important difference is the influence of evaporation temperature on the optimum pressure value. The author's model depends on the evaporation temperature much stronger than rest of authors' models studied, even though this influence is reduced for the system B.

Another reason for differences between the presented results and the other authors' correlations could be the detailed modelling used for the present work compared with others models: an actual performance model of the reciprocating compressor was used instead of using a linear function; a segment by segment discretization of the heat exchangers; in the simulation study; lumped model for IHX; pressure losses in heat exchangers. Anyway, a deeper analysis of results with experimental data is going to be worked out for the authors to find out more clear reasons and get an accurate correlation for the optimal pressure.

4. CONCLUSSIONS

The transcritical CO_2 split air conditioning systems A and B operating in subtropical conditions are studied for COP and optimum gascooler pressure. These two systems are different in their controlling technique for the gascooler pressure. In system A, evaporator superheat is used and in system B, gascooler pressure itself is used to control the optimum gascooler pressure. The following conclusions can be made after analyzing the simulations studies presented:

- System A is cheaper than system B, but system B allows working in a reversible cycle.
- The COP of the system A is more sensible than for system B to variations of the indoor/outdoor conditions. Thus system B is more suitable to climatic conditions with large variations.
- The optimal gas cooler pressure is almost the same for both systems A and B. The reason could be in the negligible effect of the superheat in the value of the optimal pressure.
- Improvements on COP, for both systems, with respect a system without control of the gas cooler pressure are 5% at most.
- The optimal gascooler pressure is increasing with increase in gascooler outlet refrigerant temperature and evaporator temperature for system A and B. The optimal gascooler pressure for system A is 3 to 4% than system B for same conditions.

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