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Research Publications

NUMERICAL ASSESSMENT OF EFFICIENCY IMPROVEMENT BY IMPLEMENTATION OF A CONTROL TECHNIQUE IN TRANSCRITICAL CO_2 AIR CONDITIONING SYSTEMS

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ABSTRACT

An optimum pressure that maximizes the COP exists for vapor compression cycles where heat rejection occurs under transcritical pressure, e.g. CO_2 transcritical cycles. The paper presents a performance comparison between two transcritical split air conditioning units operating under typical subtropical conditions of Pune city (India). First system of comparison has a control technique to set the gascooler pressure to its optimum value while second system does not have such a control. The comparison has been carried out by numerical simulations as function of indoor and outdoor conditions. The parameters analyzed are the COP improvement and power input reduction by using a control procedure. The results revealed COP improvements up to 15% and power reductions at most of 22%. The operating conditions to find the largest improvement correspond to large indoor humidity and low outdoor temperature.

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, either theoretical or experimental, on the performance of transcritical CO_2 refrigeration systems covers automobile air conditioning applications where ambient temperature is less than 35°C.

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, which results in the non-monotonic variation of COP with the gascooler pressure. Liao et al. (2000) proposed a correlation for optimum gascooler pressure as function of evaporator temperature, gascooler outlet temperature and compressor performance for transcritical CO_2 split air conditioning system using theoretical thermodynamic models. 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) 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 a water-to-water combined heating and cooling transcritical CO_2 system. The researchers observed that for fixed evaporation temperature, the decrease in the gascooler outlet temperature decreases the optimum gascooler pressure and sharply enhances the system

COP. Cabello et al. (2008) developed a correlation for optimum gascooler pressure based on experimental results. They observed that inclusion of the theoretical model for the compressor proposed by Liao and Sarkar gives close prediction of optimum gascooler pressure to experimental results. Ge and Tassou (2009) studied the CO_2 cycle for medium temperature food retail refrigeration applications and concluded that it is possible to keep constant the approach temperature of the gascooler by using variable frequency drive fans for high ambient conditions. The research proposed a correlation for optimum gas pressure as function of ambient temperature. At the evaporating temperature of 10 °C, when the gas cooler refrigerant outlet temperature increases from 33 °C to 45 °C, the optimum heat rejection pressure increases by 23.06%.

The main goal of the present work is to evaluate, with simulation studies, the improvements on performance when there is a control of the gas cooler pressure, to keep it equal to the optimal pressure, with respect to a system which does not have such a control of the gas cooler pressure. The present research is focused on quantifying the efficiency improvements that a control procedure of a transcritical CO_2 split air conditioner can achieve when it is operated in typical subtropical conditions observed in India. The first task to this end is to assess the steady-state performance of a cooling unit with a control procedure to work always with the optimal gas cooler pressure. Present authors (Padalkar et al., 2010) analyzed in detail two ways to keep gas cooler pressure in its optimal value for a split air unit air conditioning. The best system proposed in that work has been chosen for the present work to be analyzed in detail.

2. MODEL DESCRIPTION

The simulation study has been performed by means of IMST-ART software (Corberán et al., 2002). A short description of the main characteristics of the model in what matters the present paper is given in this section. For a full description of its characteristics and capabilities the reader is referred to IMST-ART (2010).

The global model of the whole system is divided in submodels: compressor, heat exchangers, expansion device, accessories, and piping. Each submodel involves a series of non-linear equations and in the case of the heat exchangers, a system of ODEs, which is discretized with a finite volume technique. Then, the submodels are coupled to form a global model of the heat pump. The global set of equations forms a complex system of non-linear equations AEs and DAEs, which is solved globally by a Newton-like solver. The global system of equations is solved using a standard solver based on the MINPACK subroutine HYBRD1, which uses a modification of M.J.D. Powell's hybrid algorithm.

Calculation of the refrigerant thermodynamic and transport properties is performed by REFPROP subroutines from NIST for each refrigerant. The corresponding properties are then conveniently stored in a refrigerant data library. The required properties during the simulations are estimated by interpolation from the corresponding data file. Additionally, built-in tables allow the calculation of the properties of any usual secondary fluid, i.e. dry air, humid air, water and common brines.

IMST-ART models the compressor performance as function of the volumetric efficiency, the compressor efficiency and the fraction of power input which is lost to the environment from the outer shell of the compressor. For this simulation study the volumetric and compressor efficiencies were obtained from catalogue date as a function of the pressure ratio. Alternatively, IMST-ART allows definition of compressor by using the ARI performance polynomials provided by the compressor manufacturer. A fraction of 5% was assumed for the het losses.

The model of heat exchangers is the most important model of the whole system in order to get accurate results, and it is also the most complicated. Heat exchangers are modeled applying a segment-by-segment approach. In the case of the evaporator or condenser, a 2-phase flow with phase change occurs. A steady

2-phase flow is considered to occur along the refrigerant path. The separated two-phase flow model is assumed. At a fluid cell, the number of equations to be considered is two: energy and momentum.

The numerical method employed for the heat exchangers solution is called SEWTLE (Semi Explicit method for Wall Temperature Linked Equations). Basically, this method is based on an iterative solution procedure, which consists in an iterative series of explicit calculation steps. For further description, see (Corberan et al. 2001). Correlations from literature are employed for the evaporation and condensation heat transfer and friction coefficients.

For the air side, the dehumidification phenomenon is modeled by applying the approach recommended by Threlkeld (1970).

3. CASE STUDY DESCRIPTION

The paper compares the performance of two systems, depicted in Fig. 1, for a wide range of indoor/outdoor conditions.

System A consists of: a reciprocating-hermetic compressor; a gas cooler of finned tubes; an internal heat exchanger (IHX); a thermal back pressure regulator (TBR); an evaporator of finned tubes; and a receiver at the compressor inlet. The receiver imposes that the refrigerant at the inlet of the IHX is always saturated vapor. A great advantage of the liquid receiver is that allows the system to be easily used as reversible cycle. On the other hand, disadvantages of the liquid receiver are: possible oil build-up in the receiver and the cost of an additional component.

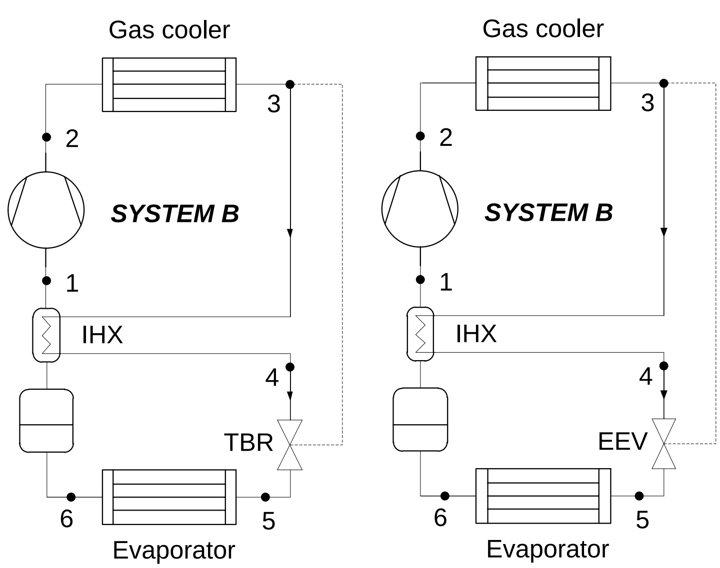

Figure 1. Schematics of the two systems studied

The design of these components has been carried out for IS 1391 Part 1 (1992) standard conditions: indoor dry bulb temperature (DBT) of 27 °C, indoor wet bulb temperature (WBT) of 19 °C and outdoor DBT of 35 °C and outdoor WBT of 24 °C by using the software IMST-ART (Corberán et al, 2002). Main data used to define all the components was reported by present authors in Padalkar et al. (2010). Table 1 presents a summary of the most important parameters.

Table 1. Description of system components (Padalkar et al., 2010)

Evaporator/Gas Cooler		***Compressor***	
Tube Length [mm]	550/550	Maker and model	Danfoss, TN1416
Depth [mm]	36/50	Displacement capacity [cm3]	2.46
Height [mm]	180/375	***Internal Heat Exchanger (IHX)***	
Width [mm]	500/700	Overall Thermal Resistance (1/UA) [W/K]	4
Tube diameter [mm]	4.75/4.75		
Fin spacing [mm]	1.37/1.37		

The CO_2 air conditioning system A controls the system to keep gas cooler pressure in a constant value regardless the indoor/outdoor conditions. Thus, it cannot assure the optimal gas cooler pressure when indoor/outdoor conditions change. The gas cooler pressure is obtained by means of gas cooler approach, which is analogous to fix a gas cooler pressure given indoor/outdoor conditions. Temperature approach is defined as the temperature difference between gas cooler outlet temperature and air inlet temperature.

Thus, to define completely the system A, an additional parameter is needed: the temperature approach. This approach value will be the setting point for the TBR. The TBR will control the system to keep the approach always to this set point. Since only one setting point is possible to be defined, it will be chosen the approach that for standard conditions leads the system to work with the the optimal gas cooler pressure. A parametric study for the temperature approach was carried out by using IMST-ART in order to identify the approach that maximizes COP. An approach value of 3.3 K resulted from that parametric study, corresponding to a gas cooler pressure of 9775.1 kPa. If correlation of Liao et al. (2000) is used it results a gas cooler pressure of 9521 kPa, which agrees well with the value obtained with IMST-ART simulation.

System A was run for the whole range of indoor/outdoor conditions and it was found that an approach of 3.3 K cannot be assured for all the conditions. Conditions where such an approach could not be satisfied corresponded to extreme conditions. These conditions were characterized by large values of outdoor DBT and low values of indoor WBT. For example, for DBT for DBT=41 °C and WBT=18 °C the maximum approach which is possible to get with system A is 2.85 K. This maximum value decreases as the WBT is increased or DBT is reduced. Reasons for this maximum approach are based on the system and the compressor coupling, therefore this maximum approach will be different depending on the system and TBR will be able or not to maintain the set point.

For this reason, authors decided to fix a superheat of 2 K for System A, in order to satisfy a stable operation of the system A for the whole range of indoor/outdoor conditions. This is the first disadvantage of a non-controlled system: some parameters have to be chosen just to satisfy extreme conditions, which are not optimum for standards conditions.

System B, which is the control-equipped system, is based on system A and it uses the same components with the exception of the electronic expansion valve (EEV) instead of the TBR. The EEV allows to this system to modify the set point to keep the gas cooler pressure always equal to the optimal value. The EEV analyzed in this work operates with the temperature approach as control variable. To define the EEV in IMST-ART it is necessary to define the control function of this component, which will be presented in next section.

4. SIMULATION STUDIES

Simulations of systems 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: from 18 °C to 23 °C and from 33 °C to 45 °C respectively. The indoor DBT was always kept to 27 ºC.

The first studies correspond to analyse the performance of system A when the gas cooler pressure varies. This study allows knowing the sensibility of system A to the gas cooler pressure variation, consequence of variation of the indoor/outdoor conditions when there is no control. Fig. 2(a) 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. This figure shows that a system such as A could suffer large detriments in COP for applications with large indoor/outdoor variations. Thus it is worth assessing the potential of a controlled system such as B for energy saving purposes.

Fig. 2(b) shows the corresponding approach for the same cases plotted in Fig. 2(a). Fig. 2(b) allows defining easily the control function for the EEV of system B, which has not been defined yet, for WBT=19 °C. This control function can be defined as the curve that crosses the maximum of each curve. This curve should be obtained for each WBT, obtaining a map for the approach as function of DBT and WBT. This map was obtained by simulation, which is plotted in Fig. 3, and the EER was characterized with it in IMST-ART.

Fig. 3 shows the influence of indoor/outdoor variables on optimum approach, where it can be noticed that the optimum approach is more sensible to the outdoor DBT than indoor WBT. Optimum approach ranges between 1.5 and 5 K, and more approach is needed as the indoor humidity rises and outdoor DBT decreases.

Temperature approach for system A was set to 2 K, which does not optimizes the system operation for standard conditions, but it does for another conditions, in fact in Fig. 3 it is deducible that an approach of 2 K optimizes the COP for low WBT values and DBT about 44 °C.

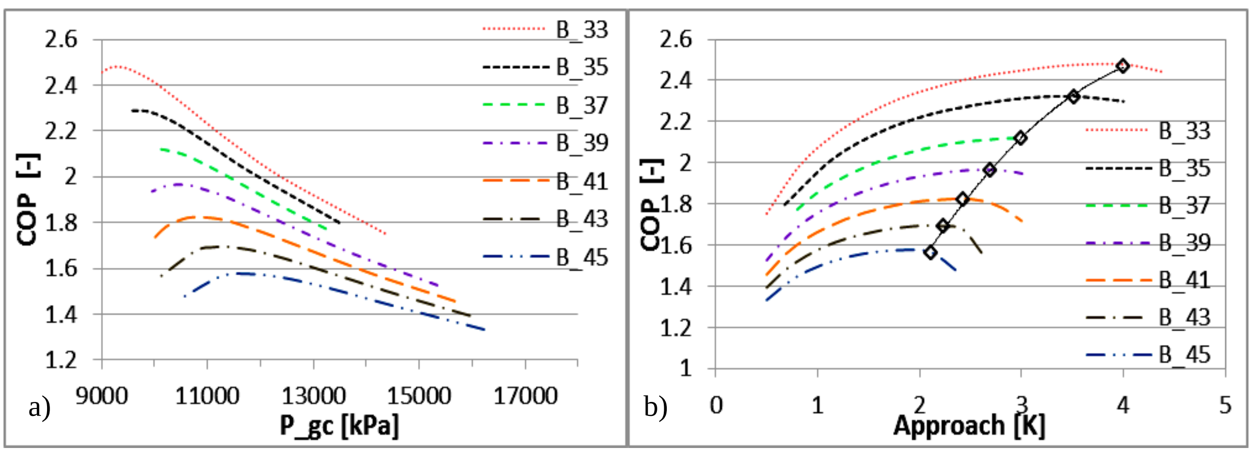

Figure 2. COP variation in system A for different values of outdoor DBT and WBT=19 °C as function of: (a) gas cooler pressure, (b) temperature approach.

In order to assess the benefits obtained by controlling the gas cooler pressure in system A with an EEV, COP improvement and power consumption reduction has been studied. The COP improvement has been defined as the COP of system B minus COP of system A, while power input reduction was defined as power input of system A minus power input of system B working in same conditions. Both parameters are relative differences with regard to the parameter value of system A.

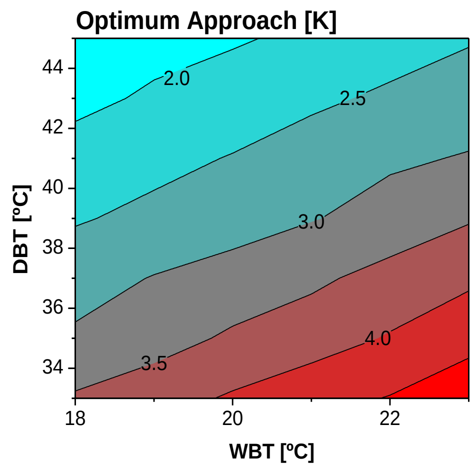

Figure 3. Map of optimum approach for system B as function of indoor WBT and outdoor DBT for characterisation of EEV.

Results for COP improvement are plotted in Fig. 4(a), where can be noticed that a large COP improvement, being up to 14%, can be achieved for large humidity values and low outdoor DBTs. This climatic region is also the most critical region for a non-controlled system operating with variable indoor/outdoor conditions, since the COP improvement raises drastically there. The most dominant variable is the outdoor DBT.

In the corner for low indoor WBT and high DBT, %COP improvement is reduced to almost a value of zero because system A has a set point of 2 K, which is close to the optimum value shown in Fig. 3. Furthermore, this corner shows a flat region for the COP improvement. This region is really characterized by large DBTs. Fig. 2(a) shows that gas cooler pressure impact on COP is much more sensible for low DBT than for high values.

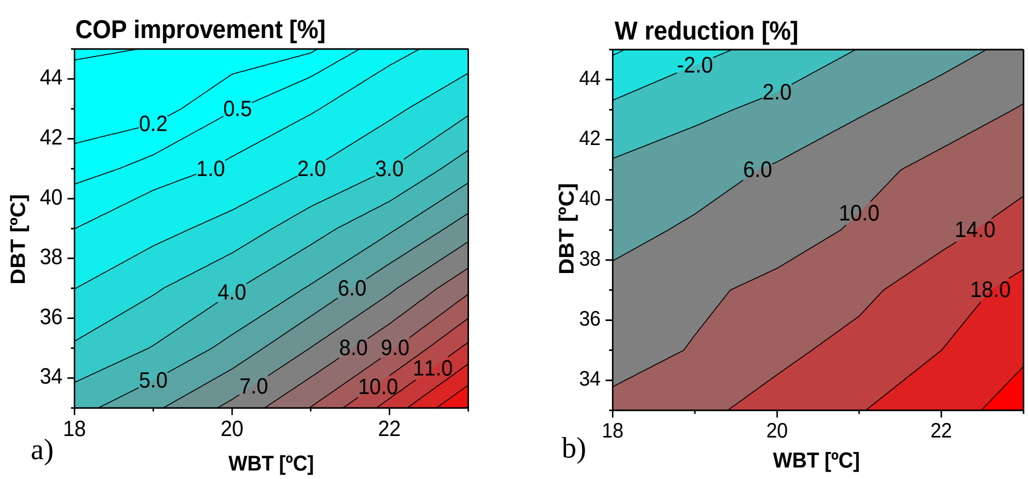

Figure 4. Performance parameters for comparison of system A and B: (a) COP improvement, (b) Compression power reduction.

Power consumption reduction is showed in Fig. 4(b). Dependence on DBT and WBT is quite similar to that of the COP improvement. The figure shows interesting power savings, being up to 22% for low DBT and large humidity conditions. Only for large DBT and low WBT the compression power of system B is larger than for system A, but anyhow less than 6%, despite COP is still larger for system B.

5. CONCLUSIONS

The impact of controlling the gas cooler pressure, to keep it in the optimal value, was numerically assessed for a transcritical CO_2 split air conditioning system operating in subtropical conditions. To this end a system with a fixed approach was compared in terms of COP and power input against the same system but equipped with an EEV which allowed to lead the gas cooler pressure to the optimal value for any indoor/outdoor conditions. The following conclusions can be made after analyzing the simulations studies presented:

- COP improvements up to 14% were found by controlling the gas cooler pressure by means of temperature approach. For high outdoor DBT and low indoor humidity, there is a flat and almost negligible improvement.

- Power input is reduced up to 22% by controlling the gas cooler pressure by means of temperature approach. Only for high outdoor DBT and low indoor humidity the power input is increased (less than 6%).

- The most critical region for a non-controlled system operating with variable indoor/outdoor conditions corresponds to large indoor humidity and low outdoor DBT, since improvements rise drastically there.

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REFERENCES

Bureau of Indian Standard IS 1391: Part 1: 1992 Room Air Conditioners - Specification - Part 1 : Unitary Air Conditioners

Cabello R., Sanchez D., 2008, Llopis R., Torrella E., Experimental evaluation of the energy efficiency of a CO2 refrigerating plant working in transcritical conditions, Applied Thermal Engineering, 28, 1596-1604.

Corberán, J.M., Fernández de Córdoba, P., Gonzálvez, J., Alias, F., 2001. Semiexplicit method for wall temperature linked equations (SEWTLE): a general finite-volume technique for the calculation of complex heat exchangers. Numerical Heat Transfer Part B-Fundamentals 40 (1), 37-59.

Corberán J.M., Gonzálvez J., Montes P., Blasco R., 2002. 'ART' a Computer Code to Assist the Design of Refrigeration and A/C Equipment. International Refrigeration and Air Conditioning Conference at Purdue, IN, USA

IMST-ART, 2010.Simulation tool to assist the selection, design and optimization of refrigeration equipment and components, http://www.imst-art.com , UniversitatPolitècnica de Valéncia, Instituto de IngenieríaEnergética, Spain.

Liao S.M., Zhao T.S., Jakobsen A., 2000, A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles, Applied Thermal Engineering, 20, 831-841

NIST. Reference Fluid Thermodynamic and Transport Properties. Database (REFPROP): Version 8.0.

Padalkar, A., Martinez-Ballester S., Kadam, A., 2011, Control optimization of transcritical CO_2 air conditioning systems operating in subtropical climate, IIR's International Congress of Refrigeration, Prague, 2011

Sarkar J., Bhattacharyya S., M. Ram Gopal, 2004, Optimization of a transcritical CO_2 heat pump cycle for simultaneous cooling and heating applications, Int J of Refrig, 27, 830-838

Ge, Y.T, Tassou, S.A., 2009, Control optimisation of CO_2 cycles for medium temperature retail food refrigeration systems, Int J of Refrig, 32, 1376-1388

Threlkeld, J. L, 1970, Thermal Environmental Engineering. Prentice Hall, Englewood Cliffs, NJ