CRITERIA TO SELECT OPERATING PARAMETERS OF A CO2 HEAT PUMP/REFRIGERATION SYSTEM

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Abstract

CO_2 raises several difficulties in designing, building, and controlling the operating parameters. However CO_2 is a environment friendly refrigerant with excellent thermophysical properties. These are sufficient reasons to seek to identify applications and solutions to use CO_2 at a larger scale as refrigerant. CO_2 by itself and the systems using transcritical CO_2 are very different compared to the common refrigeration systems. This is why selecting an application, a system configuration, and the operating conditions is not are trivial tasks. Several criteria used in those selection processes are listed and described . Analytical results regarding the behavior of the transcritical CO_2 system is discussed when parameters experience small changes from the design conditions. Experimental results are presented to validate the importance of the analytical studies. Two and single stages compressors are considered for analysis and during experiments

Introduction

Refrigerants are used in a multitude of industrial, commercial, and residential applications. Almost every residence or business has an air conditioning unit and a refrigerator. Among those, the refrigerators are the smaller consumers. Barely the power of an electric bulb is what a refrigerator is using today. Do not imagine that those low power consumption applications are not subject to energy regulations requiring to reduce even further energy consumption. Extensive work is done in that direction and advanced technology is used for that purpose(Jacobsen 1995).

Although the CO_2 applications are just a few niches at this time, the power consumption of each CO_2 systems and the electric bill is not negligible at all. The energy consumption of the CO_2 systems is expected to be high. Optimization work done at the level of system component seems to be the method to be applied for CO_2 systems also.

What I suggest in this study is a different approach to optimization. One ought yet to analyze if the system itself is worth to be optimized at component level. Before working on tuning a system we need to ensure that the system is reliable and that the application is safe.

It is simple to list the factors and the objective function for a common domestic refrigerator. The factors would be the efficiency and the production cost. The objective functions are the customer investment and the electrical bill.

Is the electrical bill an objective function anymore in the case of CO2 as refrigerant? Maybe it is not. For sure, there are expectancies that the investment is going to increase sensibly. Those simple comments motivate one to look deeper in the optimization work.

Instead of applying methods used for standard systems with freon, I will rather start from scratch and analyze the objective functions and the factors in the case of using CO2 as refrigerant?

What are the objective functions in the case of using CO2 as refrigerant?   What are the factors?   How many disciplines are involved?   To do a correct study one ought to more advanced Mathematics and learn about Multi Objective Optimization and Multi Disciplinary Optimizations Theory. For a large number factors and objective functions the fuzzy logic method is a good analysis tools.

The CO2 proves to be a very good refrigerant when calculating the system performances at certain condition (Beaver et al. 1999). This study tries to answer to the question if the CO2 systems is still efficient or if it operates at all at other conditions than the one that make CO2 look as a very good refrigerant.

Optimization Factors and Objective Functions
Of more interest is though the discussion itself about factors and objective functions. It turns sometimes to be an issue of semantics if something is a factor or an objective function. The first step in this analysis would be to list everything that is related to the use of CO2 as refrigerant.   The second step is to group those items in factors and objective functions. A third step is to organize those data and see what is the relation priority among those items.

This is a list of possible factors and objective functions:

Table 1   List of factors and objective functions

Natural refrigerant	Access to technology
Instant heat	Patent protection
Very good thermo physical properties	Pressures
Small volume	Temperatures
Profit	Weight
Creating a market	Regulations
Reliability	Academia
Environmental groups	Pressure drops
Efficiency	Control
Cost	Oil miscibility
Refrigerant availability	Service personnel and equipment
Lubricant availability	

The list could be extended.   Some of those items are in real contradiction.   For example CO2 is cheap and easy to be obtained but what about the handling of the expensive hygroscopic lubricant? CO2 could bring potential relief to the nations where food preservations is a serious survival concern. Would such nations afford special equipment for CO2 handling? What are the factors and what are objective functions in that list? What meaning would have an optimization that takes in account only a limited number of factors?   The final goal is to have CO2 used as refrigerant at a large scale in air conditioning, heat pump, and refrigeration application.

One can optimize the discharge pressure taking in account the special profile of the isotherms on top of the critical point. Analysis can be made on the suction line heat exchanger (Yahia et al 2000). However, what is the outcome of that kind of optimization? Is any of those results going to motivate manufacturing companies to switch to CO2 as refrigerant? What prevents the manufacturing companies from using the CO2 are the following factors instead: weight, cost, high discharge temperature, low efficiency, high operating pressures, difficulty to control the operating parameters. We can consider these as factors also and the scale of using CO2 as refrigerant as the objective function.

The optimization of CO2 systems proves out to be not a deterministic analysis. The evolution itself of the CO2 as a refrigerant is not a simple succession of events. A similar situation was presented about the case of variable capacity refrigeration [Manole 2002]. There are three major forces that are providing pushing toward using CO2 as refrigerant. The Thermodynamics show that CO2 has some advantages and that some disadvantages can be overcome by a refrigeration system of a configuration more complex. The technology advances in heat exchangers and telecommunications made possible to build systems that operate at high pressures and to build complex systems to control the pressures and temperatures. Then there are Governmental Regulations and Environmental Groups indicating what efficiency, refrigeration fluids, and performances are acceptable. When it happens that those three forces are acting in same direction opportunities occur for new applications as illustrated in Figure 1. The Case of Variable Capacity resulted in numerous applications accepted by manufacturers and market as described in (Manole 2002).

However, the case of CO2 as refrigerant is not that clear yet. The 'three forces' are acting in same directions but with either not enough force or our of phase. This is why there are still only niches where a CO2 application seems to be an opportunity at this time and the situation is in continuous change. This is why items listed in Table 1 can be in conflict at times.

Heat pumps and water heaters appear to be applications for which cost and high temperatures concerns can be overcome. The heat rejection for the CO2 in supercritical state exhibits a temperature glide. This feature is an advantage compared to the normal refrigerant that has a constant temperature process during condensation heat rejection process (Groll and Cohen 2000).

We have designed, built, and tested a compressor for a specific application – mobile AC/HP. We specified and participated in designing the thermodynamic cycle for that application. The next step in our Engineering work was to generalize our findings. This is when we encountered the difficulties because of discontinuities in the realm of CO2 applications as suggested in Figure 1.
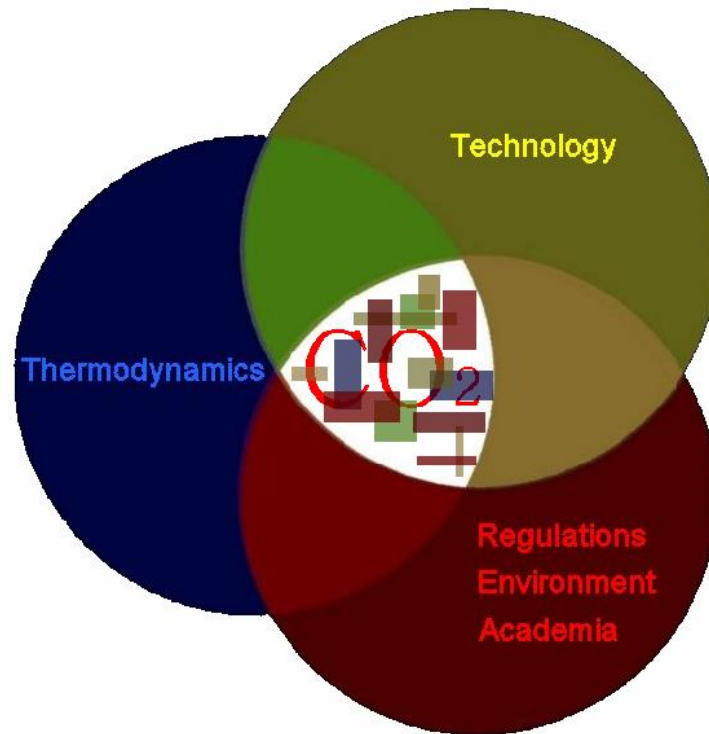
Figure 1. CO2 as a Refrigerant - Driving Forces Paradigm

Our compressor and parameter settings and control does not transfer as easy to another applications as in the case of more familiar freons. It turned out that it is more important to better define and to share the method of designing and setting the operating parameters than the values of the optimum parameters per say.

Intermediate pressure

Figure 2 shows the heat pump system used for thise current analysis. The system consists of an evaporator, a suction line heat exchanger (SLHX), a compressor with two stages, an intermediate pressure heat exchanger (IPHX), a gas cooler, and an expansion valve.
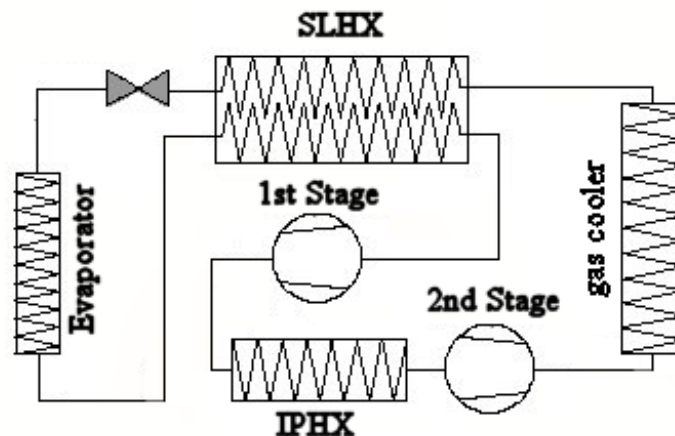

Figure 2   Two compression stage heat pump

One can optimize the heat pump illustrated in Figure 2 for a set of operating parameters (Yahia et al. 2000). We have studied analytically and experimentally what happens if during the operations the ambient temperatures and heat load are changing. The heat exchange surfaces, the compressor displacement for each stage, and the amount of CO2 charged in the system are constant. When the heat rejection load need varies in a heat pump using a common refrigerant, the condenser pressure would change and that tunes the heat transfer between the freon in the condenser and the ambient air. In the CO2 system represented in Figure 2 a reduction of the heat load causes the average temperature in the cooler to increase. The increased average temperature causes the density of the gas to decrease. The effect of the increase in average temperature in the cooler is compensated by a trade off between a migration of the CO2 into other parts of the system and a pressure increase in the cooler.

Please note that the pressure increase at the cooler means also more work for the compressor and higher discharge temperature, thus increasing even more the average temperature in cooler. The higher the value of the polytropic coefficient is, the hotter the discharge gas gets thus the more the pressure in the cooler needs to increase to accommodate the hot CO2.
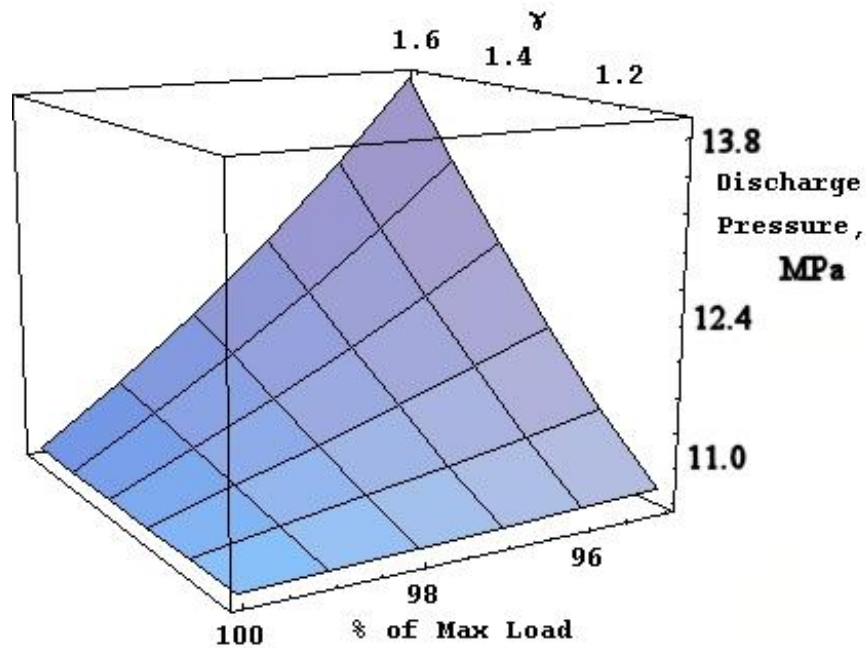

Figure 3   Cooler pressure variation due to heat load requirement reduction

Figure 3 shows how the pressure varies for the system we analyzed when the heat rejection load is reduced. For a heat rejection load of 100% the cooler pressure is designed to be 11.0 MPa. If the polytropic coefficient,γ , during the compression in the second stage has the value 1.6 then a reduction of 5% in heat rejection load could lead to an increase in cooler pressure to a value of 13.8 MPa. Of course, those values depend on the system inner volume, the ratios among the inner volume of the evaporator and the other heat exchangers, and other a few parameters. Nevertheless,

one can see that the cooler pressure variation can be substantial in the case of a high value of the polytropic coefficient during the compression.

A further analysis was made on the fact that heat pump and water heater applications can take advantage of the CO2 temperature glide in the high pressure gas cooler. This temperature glide reduces the average temperature difference between the CO2 and the ambient air or water when using counter flow heat exchangers. However, if the heat load is reduced either because of the reducing the amount of water to be heated or by reducing the how water outlet temperature, then the previous gain due to the temperature glide is rapidly lost since the CO2 average temperature increases rapidly in the cooler, thus the exergy losses are increasing also rapidly.   Figure 4 shows how the exergy content of the heat delivered to the water is lost when the heat load is reduced.
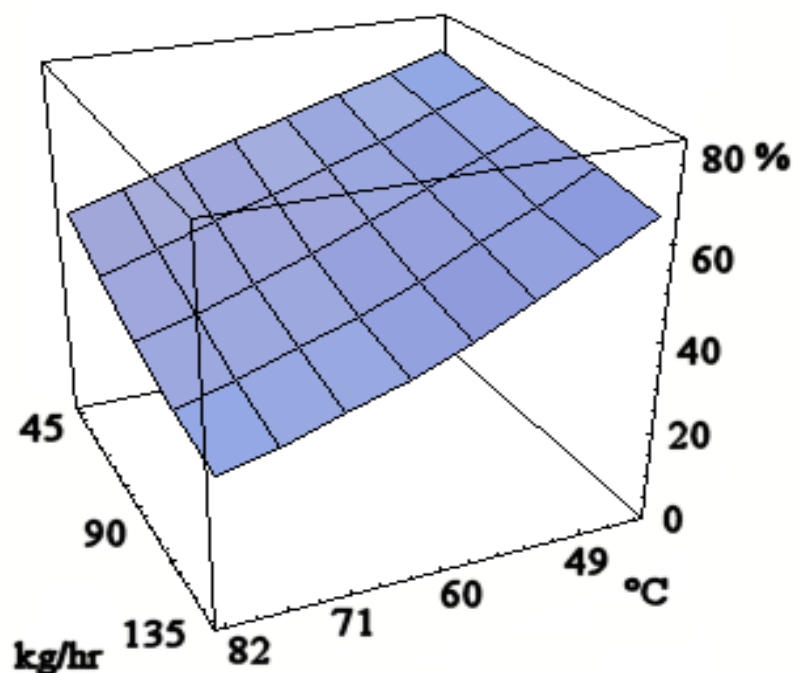

Figure 4   Exergy losses due to heat load reduction

The results presented in Figures 3 and 4 show that by varying the heat load of a heat pump or water heater the CO2 average temperature in the cooler is increasing and that causes significant losses in exergy. That means that while the temperature glide of the CO2 in the cooler is advantageous, the high exergy content of the CO2 gas at the cooler inlet could cause significant losses thus the second stage discharge temperature needs to be controlled carefully.

A two stage compression can help reducing the cooler pressure variation and exergy losses in the cooler. The discharge temperature is lower and the temperature control is easier when two stage compression is used.

Intermediate Pressure

Setting the optimum intermediate pressure for a CO2 compressor is also dependent on the system design, displacement ratio, and heat load.

The compression work for an open thermodynamic system is calculated with the equation

$$\frac{\gamma}{\gamma - 1} \, m \, R \, T_i \left[\left(\frac{p_i}{p_s} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

for the first stage and with the equation

$$\frac{\gamma}{\gamma - 1} \, m \, R \, T_s \left[\left(\frac{p_d}{p_i} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

for the second stage. The meaning of the variables in the equations is self explanatory. By adding the two equations and further simple calculations one obtains a minimum of total compression work. The intermediate pressure is the geometric mean of the evaporator pressure and cooler pressure if the mass flow rate thru both stages are the same and the suction pressures are the same for both stages.

$$p_i = \sqrt{p_d \, p_s}$$

Once this optimum intermediate pressure is calculated the displacements of the two stages can be calculated also.

It happens in practice though that the suction temperature would not be constant during the system operations. Even more, the polytropic coefficient is not the same for the compression in the two stages. A reason is that CO2 adiabatic coefficient of CO2 varies with temperature and pressure as shown in Figure 5.

Also in Figure 5 are plotted results calculated from the experimental data obtained by testing a double stage compressors with CO2. The results show that the average adiabatic coefficient of CO2 has values in the range 1.7 to 2.0.   The two stage compressor polytropic coefficient has smaller values that the adiabatic coefficient but the variation curves have same profile.
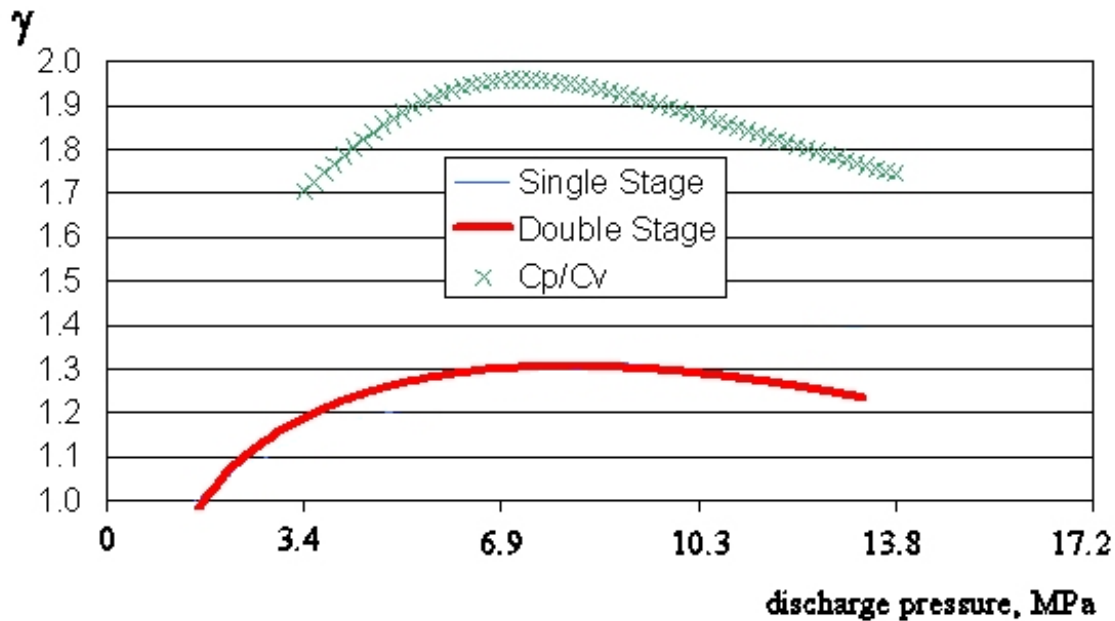
Figure 5.

The smaller values of the polytropic coefficient during the compression compared to the adiabatic coefficient are caused by the cooling of the CO_2 gas during the compression and by internal gas leaks.

The analysis of the discharge pressure showed the importance of controlling the value of the discharge temperature of the second stage. Discharge temperature can be easily controlled by the suction temperature. However, once an optimum intermediary pressure is calculated and the displacement is calculated for each compressor stage, a change of the suction temperature of the second stage would have an impact on the intermediate pressure. Figure 6 shows the results of a parametric study of the effect of varying the second stage suction. This effect is studied for several polytropic coefficients since the value of the polytropic coefficent varies with the discharge pressure and can also vary with the compressor volumetric efficiency and internal heat transfer specific design. The pressure coefficient plotted in Figure 6 correct the value of the intermediate pressure as calculated as a geometric mean of the evaporating and cooler pressures.

Figure 6 shows that by increasing the second stage suction temperature the intermediate pressure would decrease by 40% compared to the value calculated by the geometric mean of the evaporator and cooler pressure. A temperature reduction does not cause that much of a change in intermediate pressure. An isotherm compression has a polytropic coefficient with value 1. These results show that for a polytropic coefficient closer to 1 the intermediate pressure experiences larger variation when the suction temperature changes.
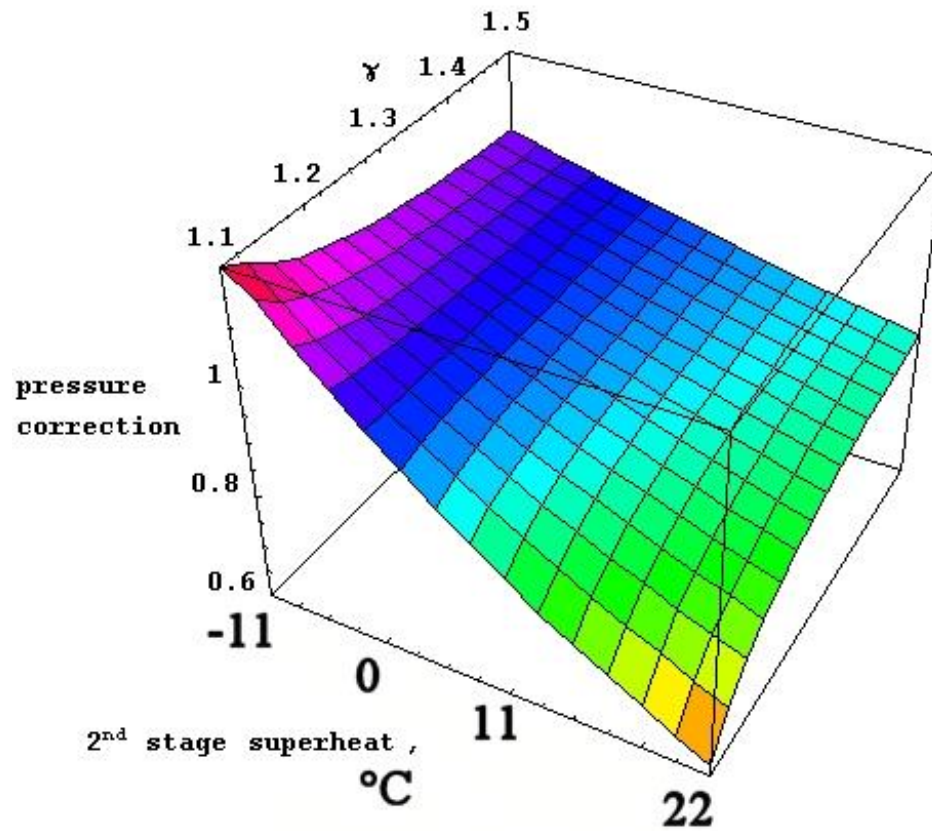
Figure 6   Intermediate pressure correction coefficient

Figure 7 shows the contours of the surface plotted ion Figure 6. The dashed line in Figure 7 outlines the points corresponding to the optimum intermediate pressure when the second stage suction temperature has the value as the first stage suction gas.

The change in intermediate pressure has a direct effect on the compression ratio for each stage. Figure 8 shows the variation of the pressure ratio for each stage with the second stage suction temperature and the polytropic coefficient variation.   Both stages are assumed to have same value for the polytropic coefficient for the results plotted in Figure 8.
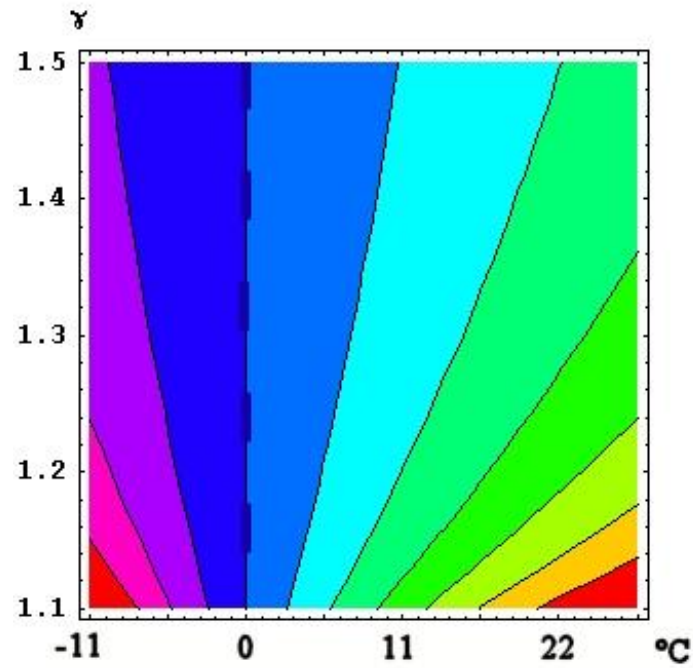
Figure 7   Contours of intermediate pressure correction coefficient variation
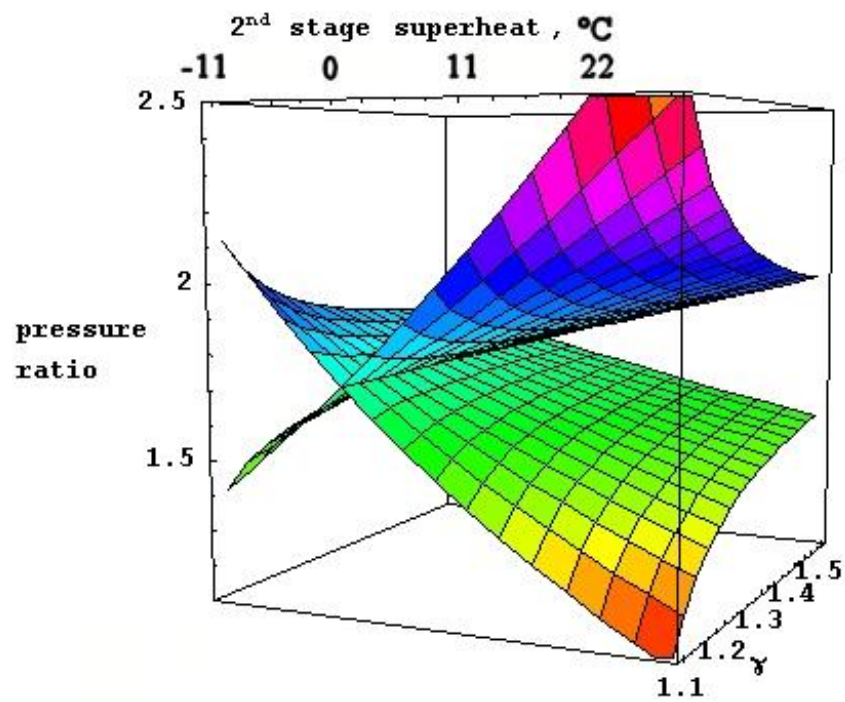

Figure 8   Pressure ratio variation with second stage suction gas superheat

These results show that a compression with a polytropic coefficient closer to 1 experiences large changes in intermediate pressure with suction temperature variation.

Figure 9 show results that are hidden by the surfaces plotted in Figure 8. Figure 9 also helps make distinction between the first and second stage data. The second stage results are represented with dashed lines.
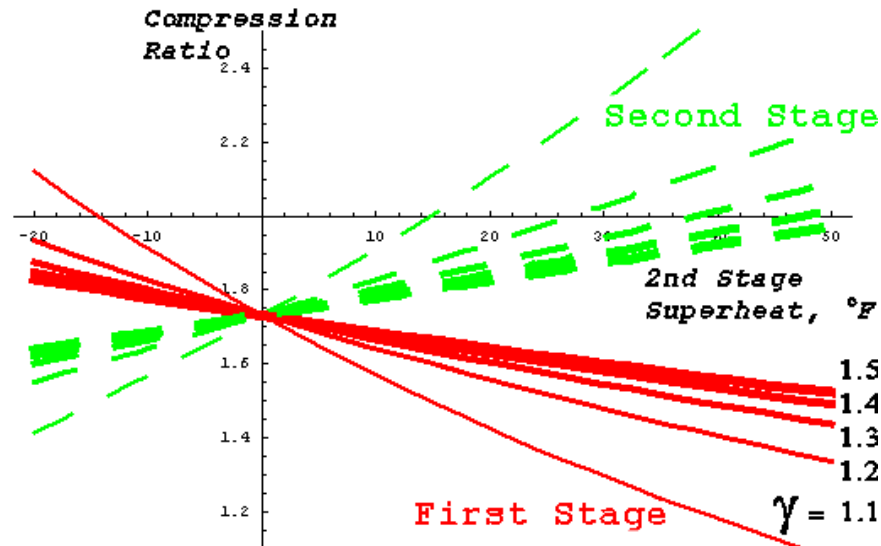

Figure 9   Pressure ratio variation with second stage suction gas superheat

The results plotted in figure 8 and 9 are important because the low pressure ratio in CO2 systems is one of the advantages of using CO2 as a refrigerant. An increase from the optimum pressure ratio would case increased internal leaks and gas recompression and cancel out exactly the reason why we used CO2 in the first place..

Discussions

Except the experimental results shown in Figure 5 all other results are calculated for specific values for the system inner volume, CO2 charge, evaporator inlet quality, etc. One should not use the values from the charts for design. The results intend to show the order of magnitude of the effect some factors have on intermediate pressure, exergy losses, pressure ratio. The present paper is presenting the method used in designing a heat pump system with CO2 as a refrigerant as well as methods to monitor and control the response of the system to ambient and load changes.

Conclusions

Until the CO2 is used at a larger scale there are only a few niches where CO2 is being used as a refrigerant. That makes the transfer of technology and test and analysis data difficult to compare. The compressor and the entire CO2 system design is very specific to each application.

The efficiency of the system is important but it is premature to work on optimizing this objective function unless other driving forces are supporting the use of that specific CO2 application.

CO2 properties have large variations about the critical point as well as in the gas domain. Load variations cause refrigerant migration in the system and pressure variations in the gas cooler. Means to control the charge inventory are important in CO2 systems.

A two stage compressor provides an additional degree of freedom that helps controlling the capacity of the CO2 system: the second stage suction temperature.

The compression polytropic coefficient is a factor to be used in the design of the CO2 system response to ambient and load changes. A thermal design of the CO2 compressor is recommended having this factor in mind.

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