2010

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ABSTRACT

This paper aims to present a simplified thermal model to predict the thermal profile of a CO_2 reciprocating compressor. The methodology was firstly presented by Todescat (1992), considering a standard reciprocating compressor. This paper applied this thermal model for a CO_2 compressor which has a different thermal network. The governing equations and numerical methodology for the CO_2 platform are detailed. The numerical results are validated through comparisons with experimental data for different operating conditions.

1. INTRODUCTION

In recent years environmental aspects are increasingly becoming an important issue in the design and development of refrigerant systems. The banning of CFCs and HCFCs in vapor compression systems, because of their negative environmental impacts, has made way for the HFCs refrigerants.

In this situation, HFCs seem to be only a temporary solution and taking into account regulations for greenhouse gas emissions, natural fluids are viable alternatives as refrigerant fluids. If non-toxicity and non-flammability are required, the focus comes to carbon dioxide.

The transcritical cycle, shown in Figure 1.a, is that one in which the compressor discharge pressure is above the refrigerant critical point, while the suction pressure is located below it. In other words, there is no condensation at the high side of the heat exchanger and the refrigerant vapor and liquid phases co-exist in only one distinguishable phase. Since usual ambient temperatures can exceed 31ºC (CO_2 critical temperature), a CO_2 refrigerating system does not exhibit condensation at the high side of the heat exchanger, as shown by the heat rejection process 2-3 in the figure, and temperatures and pressures are not linked as in saturation (dome) region. In a refrigeration cycle where temperature conditions range from subcritical to supercritical, CO_2 reaches high pressures, as much as 100 bar (10MPa) or even more at high ambient temperature and/or extreme operating conditions.

The use of R744 implies in high operating pressures as shown in Figure 1b. The differential pressure between discharge line and suction line for the R744 transcritical cycle is around four times higher than R22 Rankine cycle and up to twelve times higher than R600a. As reflect of high operational pressures, the

CO_2 compressor present high temperatures which have strong influence in the thermal efficiency and, by consequence, in the performance.

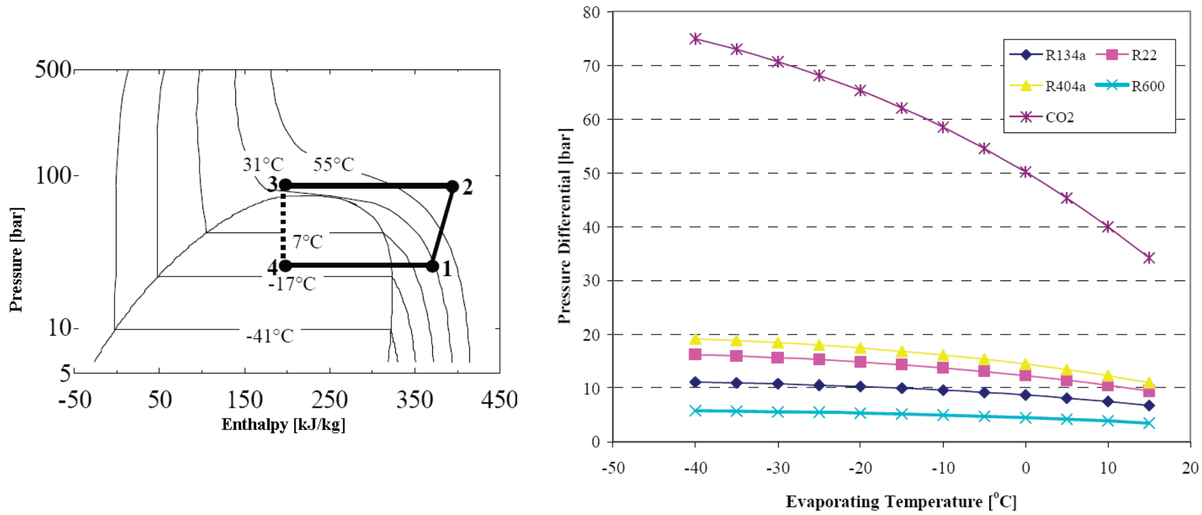

Figure 1: (A) CO_2 pressure-enthalpy diagram. (B) Pressure difference for common refrigerants and CO_2.

The compressor thermal efficiency is a result of a global thermal network that involves almost all the compressor components (Ribas Jr., 2007). Thus, for improving the thermal efficiency, it is needed to know thermal profile and heat flow paths, aiming to reduce the superheating effects.

Then, based on the HFC compressor thermal model (Todescat, 1992), a numerical model was developed to predict the thermal profile of a CO2 compressor, regarding the constructive differences and their influences in the global thermal networks.

2. CO_2 COMPRESSOR

The CO_2 compressor has a particular design when compared to households ones using HFC, CFC and HC, as showed in the Figure 2. As it is possible to notice, a expressive difference is already verified in the shell shape and it is extended to the layout of components inside compressor.

(a) (b)

Figure 2: HFC compressor (a) present expressive differences when compared to CO2 compressor.

In both type of compressor, the manifold head is the main heat source due to the high temperatures generated during the discharge process. Considering HFC compressor, the suction and discharge chambers are not close like in CO_2 compressor, where the both chambers are separated by the same wall. In this case, superheating effect increases due to the intense heat exchange between suction and discharge chamber. Then, as the thermal behavior is function of geometry and components layout, it is important to adjust the thermal model for HFC compressor to CO_2 compressor.

3. THERMAL MODEL

For complete understanding of proposed thermal model, the HFC thermal model is explained and CO_2 ones is presented, where the main differences are analyzed.

3.1. HFC Compressor Thermal Model

The methodology (Todescat, 1992), can be divided in two parts: the thermal behavior of the gas inside the cylinder and the global heat transfer to compressor components. The compression chamber is solved using a simulation methodology developed by Ussyk (1984). Considering geometric relations, the variation rate of cylinder volume and the gas properties are solved using the first law of thermodynamics, in a transient regime. The model estimates the valves dynamics, the mass flow through them and cylinder-piston leakage. When the compression process reaches steady-state, some interest properties are calculated and considered in the second part of the simulation.

Applying energy conservation equation to compressor components, they are used to evaluate the global heat transfer coefficient, UA [W/K]. Thus, the non-linear system, formed by these energy balance equation, are solved through its coupling to thermodynamics model of compression chamber.

In the simulation procedure, firstly, it is necessary to load data base of interest compressor, according Schreiner (2008). Thus, the compressor is analyzed in a determined condition, aiming to obtain associated energies and the mass flow. The temperatures are obtained through an experimental measurement and they are used in the energy balance equation, resulting in the global heat transfer coefficients (UA's).

After the coefficient definition, it is possible to estimate the compressor thermal profile by an iterative numerical process.

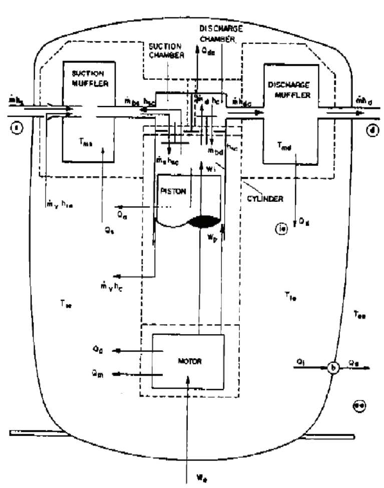

Figure 3: Schematic view of a HFC compressor for the energy balance, according Todescat (1992).

3.2. CO_2 Compressor Thermal Model

Initially, based in the HFC compressor thermal model, it was developed two simplified thermal model of CO_2 compressor, aiming to obtain a non-linear energy balance equation system. In the both models, six main heat fluxes are considered and the difference between them is defined by cylinder heat flux direction. In the first model, the heat from cylinder is exchanged to the internal environment, as it occurs in the HFC model. Considering the second one, the cylinder heat is transferred to the shell which occurs in CO_2 compressors due to its components layout.

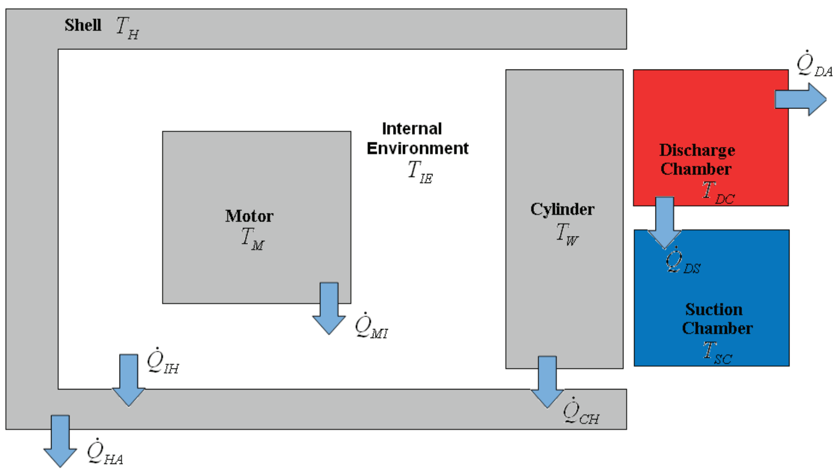

Figure 4: Second thermal model configuration and the heat fluxes direction.

Considering the second thermal model, main six heat fluxes are expanded in terms of global heat transfer coefficient and temperature differential. So, they are:

- Heat exchanged between motor and internal environment:

$$\dot{Q}_{MI} = UA_{MI} \cdot (T_M - T_{IE}) \tag{1}$$

- Heat exchanged between cylinder and shell:

$$\dot{Q}_{CH} = UA_{CH} \cdot (T_W - T_H) \tag{2}$$

- Heat exchanged between internal environment and shell:

$$\dot{Q}_{IH} = UA_{IH} \cdot (T_{MS} - T_H) \tag{3}$$

- Heat exchanged between discharge and suction chamber:

$$\dot{Q}_{DS} = UA_{DS} \cdot (T_{MD} - T_{MS}) \tag{4}$$

- Heat exchanged between discharge chamber and external environment:

$$\dot{Q}_{DA} = UA_{DA} \cdot (T_{MD} - T_{EE}) \tag{5}$$

- Heat exchanged between shell and external environment:

$$\dot{Q}_{HA} = UA_{HA} \cdot (T_H - T_{EE}) \tag{6}$$

These heat fluxes are used to determined global heat transfer coefficient when the compressor thermal profile is given, in a specified condition. Then, it is possible to use them to predict thermal profile in other condition.

4. METHODOLOGY

After model defined, it is necessary to validate the results comparing to experimental data. So, the validation method is composed by: numerical model calibration, numerical results and experimental comparison.

In the first step, the compressor model database must be loaded by the EMBRACO internal program. An experimental thermal profile is used to obtain each global heat transfer coefficient, for a determined condition. Thus, it is possible to initiate the next step.

The global heat transfer coefficients are kept the same, while the condition is changed and the thermal simulation is initiated. In the end of the simulation, the model gives whole compressor thermal profile for that specified condition and the numerical accuracy can be evaluated.

The last step will show how much accurate are the numerical results when they are compared with experimental one, as will be showed in the following section.

5. RESULTS AND ANALYSIS

To evaluate the results, it was considered a reference experimental thermal profile and, based on that, other conditions was numerically simulated. In the table below, it is showed the experimental thermal profile for the condition -23,3ºC/85bar.

Table 01: Experimental Thermal Profile in different condition

Reference Condition Experimental Thermal Profile	
Evaporating Temperature (ºC)	-23.3
Discharge Pressure (bar)	85
Temperatures (ºC)	
Suction Line	32.4
Internal Environment	71.1
Suction Chamber	56.5
Cylinder Wall	83.8
Discharge Chamber	148.7
Shell	49.7
Motor	76.3
External Environment	32.5
Bearings	72.6
Oil	70.7

In the first simulation, it was used the global heat transfer coefficients obtained by the condition -23.3ºC/85bar, and thermal simulation was done for evaluating the condition -10ºC/85bar and 0 ºC/85bar. Thus, the numerical results were compared with respective experimental one.

Table 02: Comparison between numerical and experimental results for each condition

Numerical-Experimental Results Comparison (ΔT)		
Evaporating Temperature (ºC)	-10	0
Discharge Pressure (bar)	85	85
Temperature Difference (ΔT=Numerical - Experimental)		
Internal Environment	5.4	8.7
Suction Chamber	5.0	5.2
Cylinder Wall	1.8	2.5
Discharge Chamber	6.8	10.8
Shell	1.8	5.1
Motor	5.0	9.3

Analyzing the Table 02, there is an evaporating temperature range where numerical results are closer than experimental ones. In another words, the thermal profile prediction is more accurate when the simulated condition is located near reference condition. The figure 05 shows the effect of evaporating temperature increase in the thermal profile prediction quality.

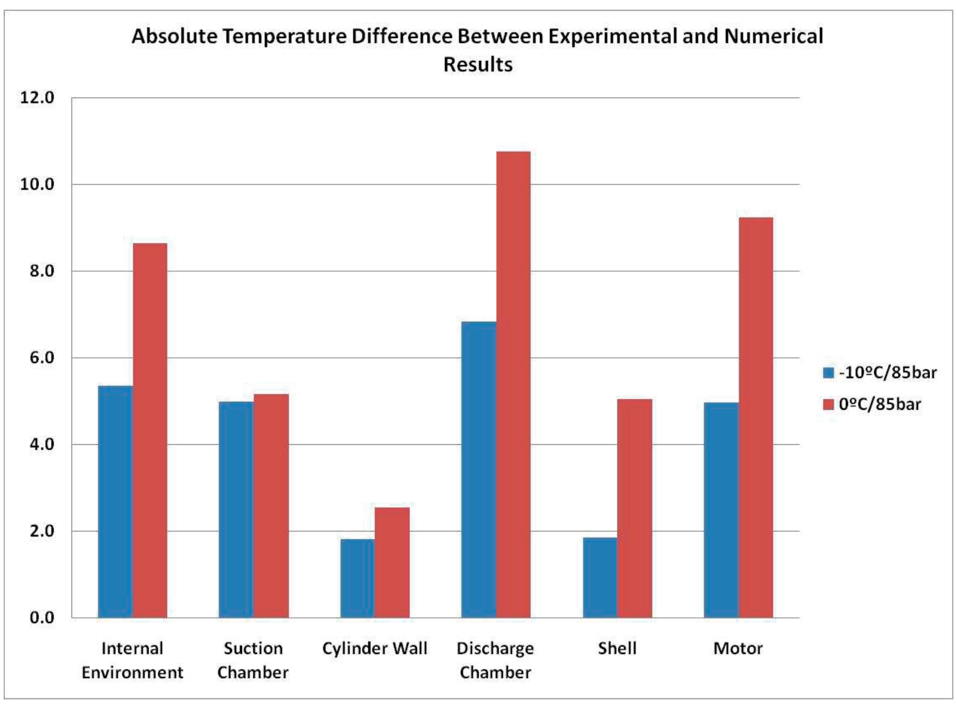

Figure 05 : Comparison results showing the temperature differential behavior.

In the results, the discharge chamber presented an higher temperature difference compared to experimental data. On the other hand, the other temperatures differences are smaller than 10ºC, which indicates the methodology accuracy for thermal analysis involving LBP and MBP condition. Thus, it is possible to use thermal profile in LBP, as reference, and simulating the thermal profile in MBP condition.

Then, this methodology allows a set of analysis, like: heat transfer area of specific component and heat transfer variation externally. The global heat transfer coefficients calibration, obtained through the experimental temperatures, considers any heat transfer mechanism inside the compressor globally and this compensates the heat diffusion effect, which it is not considered in the numerical model.

However, it is evident that material changes or flow pattern changes are not sensible in this kind of model, as well as local temperature gradients analysis. For this last case, it is recommended other methodologies where the spatial temperature effect is better represented by the heat diffusion simulation considering the real geometry. In the Figure 06, from Ribas Jr. et al (2008), is showed the thermal profile of CO2 compressor shell as result of more complete methodology.

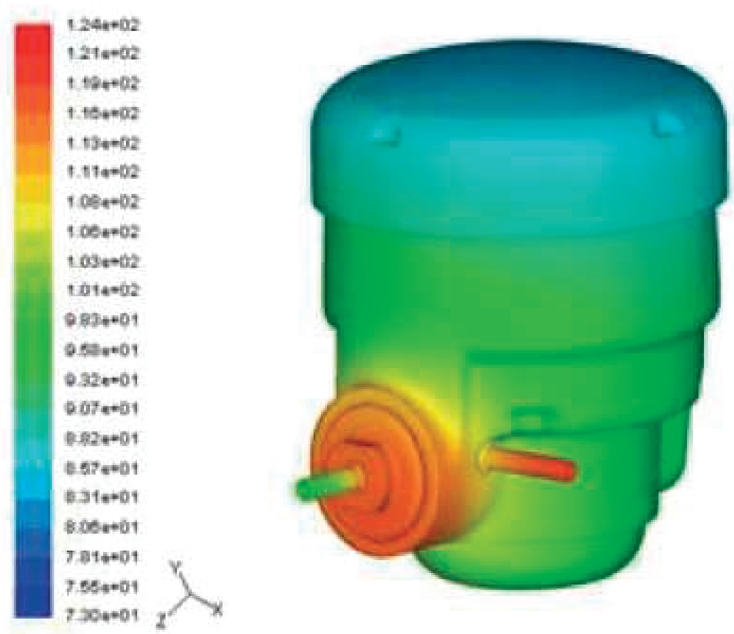

Figure 6: CO2 compressor temperature profile.

6. CONCLUSIONS

The present work presents a model to perform thermal energy analysis of reciprocating hermetic compressor adapted for CO_2 compressor platform. In the adaptation phase, the heat fluxes were evaluated regarding design differences and two thermal models were proposed. After preliminary evaluations, it was chosen one model and the comparison between experimental and numerical results is presented.

This thermal model offers good agreement with experimental temperatures, considering conditions inside an evaporating temperature range. Then, the presented model can be successfully employed for CO2 compressors evaluation.

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