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ABSTRACT

How to deal with the heat exchange inside compressors is one of the main challenges on its design nowadays. The heating of the gas entering the compressors decreases its density reducing directly the mass flow and compressor efficiency, so should be avoid. To study this topic it was developed a full three-dimensional model of the gas and the solid parts inside a compressor. Prescribed steady state mass flow rate is imposed at suction and discharge tubes and heat sources due electrical and mechanical inefficiencies of the compressor are imposed. The portion of the suction and cavity gas are separated from the portion of the discharge gas and an isentropic compression from the suction muffler outlet temperature is assumed to define the temperature of the gas flowing through the discharge port. The main advantage of this approach is the directly calculation of the heat transfer coefficients from the solid parts to the fluid, however with a high computational cost. Comparison with experimental data shows good agreement is some parts, but some disagreement in other, probably due the non-consideration of the oil flowing inside the compressor, which are topics for future investigations.

1. INTRODUCTION

Understanding and predict the heat transfer phenomena and temperature distribution is crucial in a reciprocating compressors design. The heating of the gas entering the compressors decreases its density reducing directly the mass flow and compressor efficiency. Also, excessive increase of the temperature can degrade oil lubrication properties causing catastrophic fail of the compressor.

Several studies on modeling the heat transfer process inside the compressor has been conducted along the years. They can be characterized according to Tab. 1, with different levels of accuracy and complexity, making the choice of the method dependent on the accuracy and time required for the analysis. Some works falling in each category are presented, but there are more in current literature.

The steady state lumped models (0D) are based on energy balance equations. They are computationally inexpensive but are not able to describe the effect of drastic changes in the compressor layout. Besides, heat transfer coefficients and thermal resistances between solid parts must be obtained experimentally or using general heat transfer correlations that may lead to differences in the comparison with experimental results. However, even if this kind of model was used in the early stages of the studies (Meyer and Thompson, 1988; Todescat *et al.*, 1992 and Padhy, 1992), it is still widely used due to ease of integration of more physics into the model, as showed by Dutra and Deschamps (2015).

Advances were obtained with the combination of lumped models with differential models (0D & 3D), also known as hybrid models. Being less computationally demanding than full 3D models, they are more used in the combination of 3D discretization for solid parts and lumped formulation for flow parts. Thus, the temperature gradients present in solid parts are better captured, affecting positively the results. This kind of procedure was used for part of the compressor by Almbauer *et al*. (2006) and for the entire compressor as in the work of Lohn *et al*. (2015).

A third class of methods are those full 3D models for the fluid flow and heat transfer phenomena. They can consider both fluid and solid parts and are computationally expensive, so limited to perform few cases analysis or consider just part of the compressor as performed by Pereira *et al*. (2010) and Lacerda and Takemori (2014). On the other hand, there is no need to input heat transfer coefficients and strong modifications in compressor layout can be better evaluated with this procedure. Full compressor 3D models were performed by Chikurde *et al*. (2002) and Raja *et al*. (2003).

Table 1: Map of works on heat transfer according to a modeling classification

		0D (lumped model)	**0D & 3D**	**Full 3D**
Parts of the Compressor	**Steady State**		- Almbauer *et al.* (2006) - Kara and Oguz (2010)	- Birari *et al.* (2006) - Colmek (2014)
	Transient		- Abidin *et al.* (2006)	- Morriesen *et al.* (2009) - Pereira *et al.* (2010) - Lacerda and Takemori (2014)
Full Compressor	**Steady State**	- Meyer and Thompson (1988) - Todescat *et al*. (1992) - Padhy (1992) - Ooi (2003)	- Ribas Jr. (2007) - Schreiner *et al.* (2009) - Sanvezzo Jr. and Deschamps (2012)	- Chikurde *et al.* (2002) - Raja *et al.* (2003)
	Transient	- Dutra and Deschamps (2015)	- Lohn *et al.* (2015) - Rigola *et al.* (2014)	

In this work is presented a study of a compressor using a full compressor 3D model with a steady state mass flow rate. The compression process is not included, but the temperature of the discharged gas into the cylinder head is calculated considering an isentropic compression from the temperature of the gas leaving the suction muffler. More details on the model are further discussed. Comparisons with the experimental data is showed, as well as an energetic balance and heat transfer coefficient predictions.

2. METHODOLOGY

The thermal management of the compressor consists and understand and act on the heat transfer paths from thermal sources to a thermal “receiver”. Usual thermal sources are heat generated due motor and friction losses, the compression process in the cylinder and the hot gas being discharged. All the solid parts, the oil and the gas inside the compressor can be considered as heat transfer paths and the receivers are the gas being suctioned at low temperature and the housing that exchange heat with the exterior ambient.

In the present work a full 3D discretization was adopted for the entire compressor: fluid and solid parts. The solid parts are the suction tube, the housing, the suction muffler, the valve plate, the cylinder head, the shockloop, the crankcase, the winding, the stator and the rotor. The fluid parts are the suction, the housing cavity and the discharge gas. The Tab. 2 shows a matrix of connections among all the parts in the model. The solid material properties of each part was considered, detaching that the only plastic part is the suction muffler.

A steady state mass flow rate is imposed at the suction tube considering ASHRAE-LBP conditions for a compressor that operates at 60Hz with R134a (suction pressure 97 kPa and suction temperature 32.2°C). Electric motor losses are estimated from their efficiency curves obtained in dynamometer tests and are imposed as volumetric heat sources in the winding, the stator and the rotor. Estimated losses of the bearings (Neto, 2006) are imposed locally as area heat sources. As mentioned before, the compression process is not included, but the temperature of the discharged gas into the cylinder head is calculated considering an isentropic compression from the temperature of the gas leaving the suction muffler up to the discharge pressure (discharge pressure 1,072 kPa). In the cavity gas volume is considered the buoyance effect causing some natural convection inside the housing. At the external part of the housing and the suction tube is imposed a natural convection heat transfer process considering the ambient temperature at 32.2 °C and heat transfer coefficient of 10.0 W/m².K. A graphical summary of the boundary conditions is shown in Fig. 1.

Table 2: Matrix of connections between parts

	Suction Tube	Housing	Shockloop	Cylinder Head	Stator	Muffler	Valve Plate	Crankcase	Rotor	Winding	Muffler Gas	Cavity Gas	Discharge Gas
Suction Tube													
Housing													
Shockloop													
Cylinder Head													
Stator													
Muffler													
Valve Plate													
Crankcase													
Rotor													
Winding													
Muffler Gas													
Cavity Gas													
Discharge Gas													

To perform this task the commercial code Ansys-CFX was used. Compressible transient mass conservation and Navier-Stokes equations were numerically solved. To close the problem were also solved the equation of energy balance without simplifications and a Peng Robinson real gas model for the R134a (Ansys CFX, 2012).

From the numeric point of view, the called 'High Resolution' method was employed as spatial discretization scheme. The SST turbulence model was used, which falls within the category of Unsteady Reynolds Averaged Navier-Stokes (URANS) equations and combines κ-ε and κ-ω models (Cezario, 2007), being the first used on turbulent free stream and the second on near-wall regions.

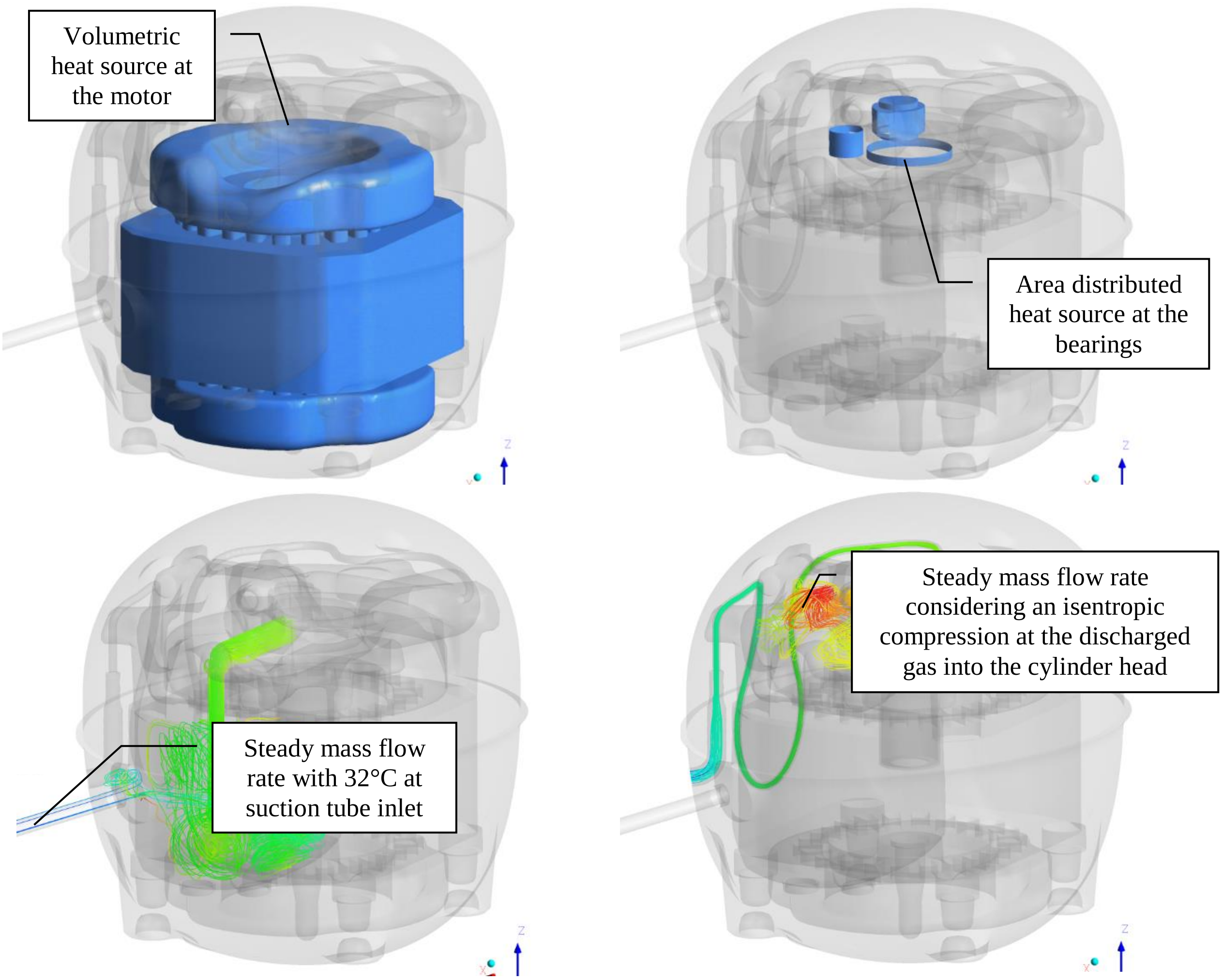

Figure 1: Boundary conditions used in the numerical simulation

The simulation was started on a coarser mesh and then finalized on a refined one with 840 thousand nodes. Around 2000 iterations was run taking around 24 hours using parallel simulation on a 12 Gb of RAM with four 3.33GHz computer processing unites.

3. RESULTS AND DISCUSSIONS

The same compressor model was instrumented with thermocouples in several parts to measure the temperature distribution inside the compressor and compare with numerical results. The compressor was mounted on a calorimeter facility and a test was carried out under ASHRAE-LBP conditions.

Table 3 and Fig. 2 shows a comparison between experimental and numerical data of the temperature distribution. The temperature are normalized by the maximum observed temperature for the solid and fluid parts, respectively. As can be noticed, in some parts it was observed a very good agreement as in the suction muffler wall close to the motor (S04, -2%) or the discharge muffler fluid (F06, 0%). However, in other parts the quantitative comparison was not so good, as for the crankcase close the cylinder (S08, 13%) or for the suction muffler outlet (F04, -12%). It was considered that most of these differences was obtained because of the absence of the lubricant oil in the model or the assumption of an isentropic compression process. The oil plays a major hole in the compressor temperature distribution because due its higher specific heat, compared to the gas, it is capable of absorb and reject a higher amount of heat wherever it goes. That can be the explanation of the temperature difference in solid parts as the motor, the crankcase, the shockloop and the housing, where a big amount of oil is splashed in during compressor run. The inclusion of the oil presence seems to be a great advance for future works using this technique.

Table 3: Temperature comparisons between experimental and numerical data

			Experimental	Numerical	% Difference
Solid Parts	S01	Lower Housing	0.52	0.49	-7
	S02	Upper Housing	0.61	0.53	-12
	S03	Suction Muffler (Housing Side)	0.49	0.45	-7
	S04	Suction Muffler (Motor Side)	0.56	0.55	-2
	S05	Suction Muffler Outlet	0.57	0.61	7
	S06	Cylinder Head	0.98	1.00	2
	S07	Discharge Muffler	0.97	1.00	3
	S08	Crankcase close to cylinder	0.73	0.82	13
	S09	Stator and Winding	0.72	0.79	10
Fluid Parts	F01	Suction Tube Outlet	0.28	0.28	-1
	F02	Suction Muffler Inlet	0.37	0.35	-4
	F03	Suction Muffler Volume	0.45	0.41	-11
	F04	Suction Muffler Outlet	0.48	0.42	-12
	F05	Cylinder Head	0.99	1.00	1
	F06	Discharge Muffler	0.97	0.97	0
	F07	Compressor Outlet	0.83	0.89	8
	F08	Oil	0.55	0.58*	7

* The numerical model did not considered the oil. It is used the gas temperature instead

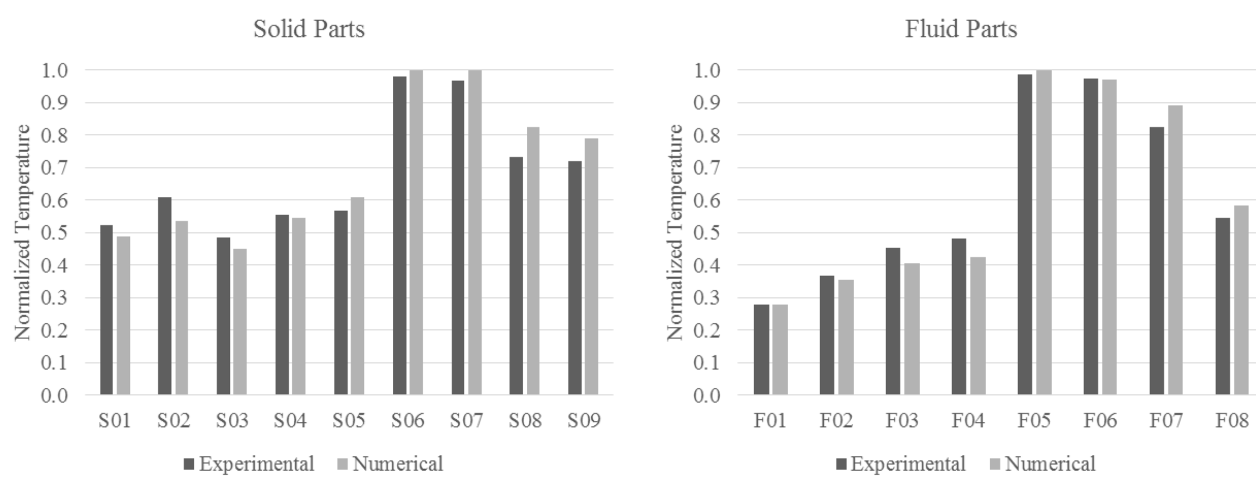

Figure 2: Normalized temperature distribution comparison between experimental and numerical results

Despite the observed differences in the temperature distribution, qualitatively it was observed a good agreement between numerical and experimental results. Therefore, an important information to understand how is the heat transfer process inside the compressor is the heat exchange from part to part that can be monitored numerically. For example, Fig. 3 shows the normalized heat exchange into and from the cavity gas (the maximum heat exchange value is used to normalize). Is noticed that a great amount of heat is coming from the motor (rotor, winding and stator) and from the crankcase. Other interesting aspect to point is how the heat coming from the discharge process is distributed, being a higher amount coming from the shockloop instead of the cylinder head. Thus, in this case, further work to decrease the contribution of the discharge process should be done in the shockloop.

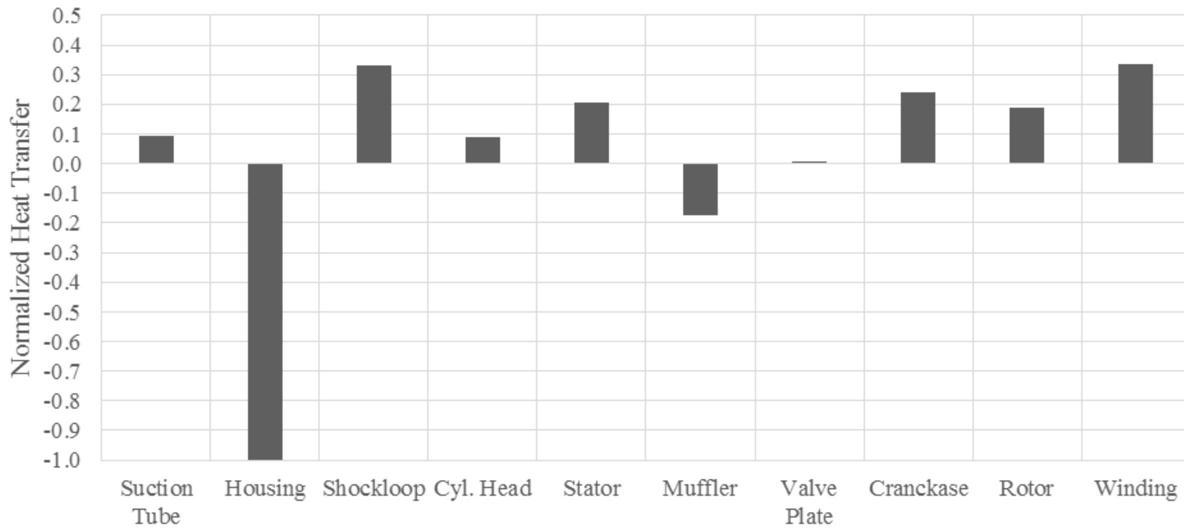

Figure 3: Heat exchange of the cavity gas

Since temperature distribution and heat flux between parts are easily accessed in this simulation, it is possible to calculate the heat transfer coefficient *h*, given by Eq. 1. This is an advantage to experimental analysis were heat flux is difficult to be measured and advantage to use of known heat transfer correlations that are suitable to more simple geometries whereas the geometries inside compressor are usually complex.

$$h = Q/A.\Delta T \tag{1}$$

In Figures 4 and 5 are presented the normalized temperature difference and the normalized heat transfer coefficients obtained for the cavity gas heat transfer. Again is detached the behavior for the shockloop, where is observed the highest value of the normalized heat transfer coefficient. This enhance in the heat transfer can be caused by an improved mixing of the gas inside the compressor due the consideration of buoyance effects.

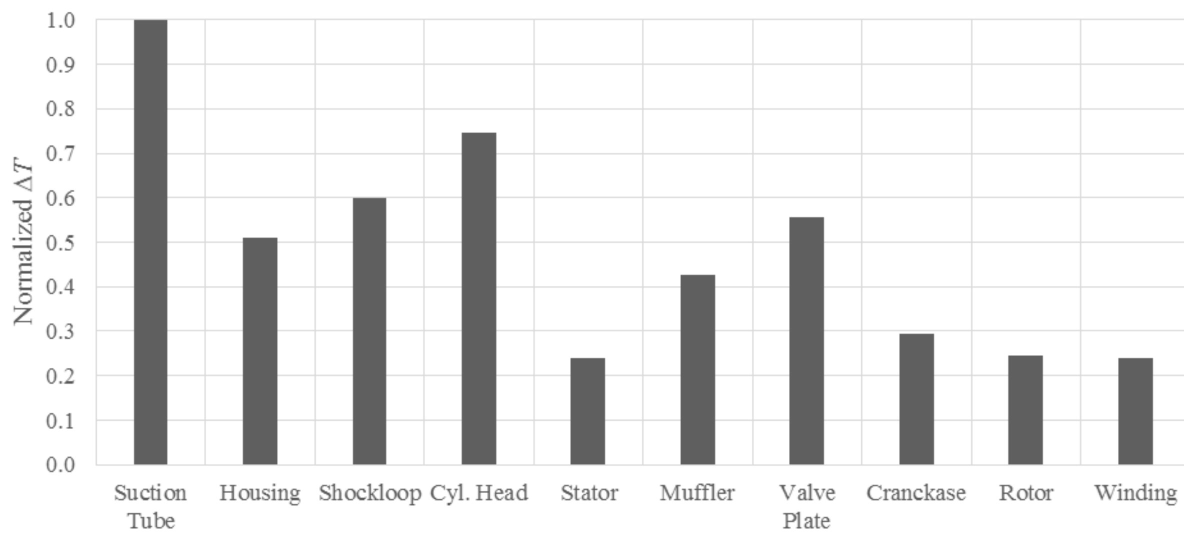

Figure 4: Normalized temperature difference between the cavity gas and solid parts

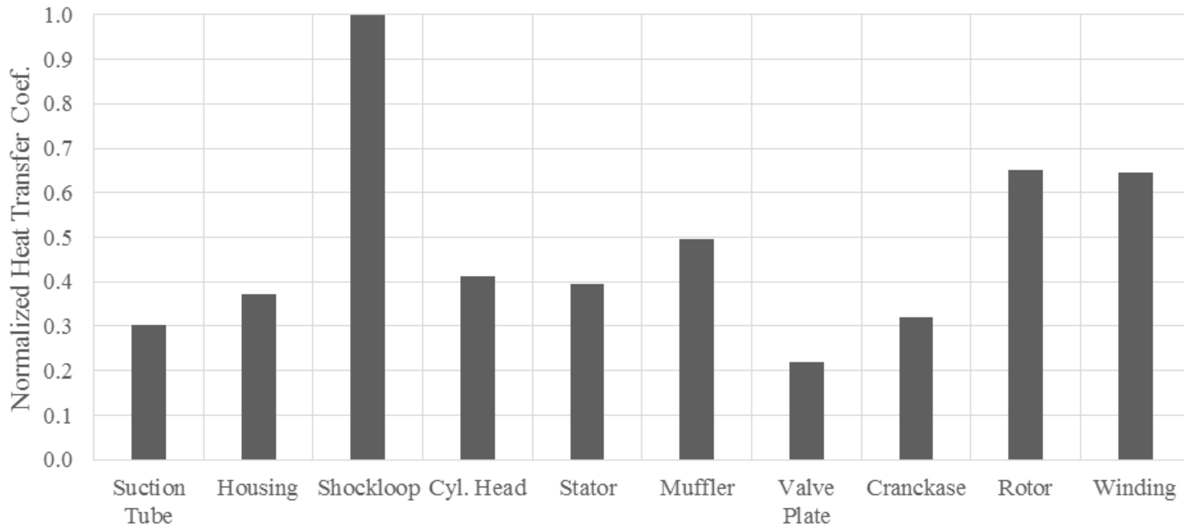

Figure 5: Normalized heat transfer coefficients between the cavity gas and solid parts

4. CONCLUSIONS

In this work the heat transfer phenomena happening inside a reciprocating compressor was investigated using a full 3D discretization model of the solid and fluid parts. This procedure is more computationally expensive than others described on literature, however it is not necessary to impose heat transfer coefficients and thermal resistances between solid parts. In fact, these information are results of the simulations. Comparison of the temperature distribution inside the compressor with experimental data showed good agreement is some parts (less than 1% difference), but some disagreement in other (more than 10%), probably due the non-consideration of the oil flowing inside the compressor or the non-consideration of transient aspects, which are topics for future investigations.

NOMENCLATURE

Q	heat transfer	(W)
ΔT	temperature difference	(°C)
A	surface area	(m^2)
h	heat transfer coefficient	($W/m^2.K$)

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