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A Heat Transfer Model Combining Differential and Integral Formulations for Thermal Analysis of Reciprocating Compressors

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ABSTRACT

Heat transfer in hermetic compressors is a complex phenomenon strongly affected by fluid flow inside and outside the compression chamber and by the efficiency of both the electrical motor and the crankshaft mechanism. This paper presents a heat transfer model for thermal analysis of reciprocating compressors formed by combining a lumped formulation for convective heat transfer in some selected regions and a three-dimensional differential formulation for heat conduction in the solid components. Convective heat transfer coefficients were evaluated from correlations available in the literature so as to avoid calibration based on experimental data. The model was applied to a reciprocating compressor operating in a fully periodic condition and predictions for temperature distribution, heat transfer and volumetric efficiency were seen to be in reasonable agreement with measurements.

1. INTRODUCTION

One of the main sources of thermodynamic inefficiency in compressors is associated with heat transfer that takes place as the gas flows throughout the suction system and enters the compression chamber. This process is usually referred to as superheating and its main effect is the decrease of both the volumetric efficiency, due to reduction of the gas density, and the isentropic efficiency, since the specific work of compression becomes greater as the gas temperature is increased. For a suitable analysis of gas superheating and proposals to reduce its negative effects, one must be able to verify the compressor temperature distribution according to different design alternatives. However, the temperature of each one of the compressor components is affected by several phenomena that act simultaneously, making the temperature distribution particularly difficult to predict.

The simplest modeling approaches to predict the temperature distribution in compressors apply the first law of thermodynamics to conveniently chosen regions of the compressor via lumped formulation (Meyer and Thompson, 1990; Todescat *et al.*, 1992). Meyer and Thompson (1990) made use of convective heat transfer correlations to characterize heat transfer between the components and their surroundings. Todescat *et al.* (1992) developed a numerical model in which control volumes are linked through equivalent thermal conductances that are calibrated with reference to experimental data for the compressor thermal profile at a certain operating condition. The numerical model is able to predict several complex phenomena inside the compressor, but it is not flexible enough to allow tests of variations of compressor layout with the same set of experimentally calibrated conductances.

Padhy (1992), Sim *et al.* (2000) and Ooi (2003) proposed a thermal network model (TNW), in which heat transfer coefficients in each component are evaluated from classical correlations available in the literature. Therefore, such models do not require experimental data and, in principle, are more flexible regarding the analysis of modifications in the compressor operating conditions, since heat transfer can be correlated to mass flow rate. However, this modeling approach requires a good understanding of the main heat transfer mechanisms in each component in order to select the most adequate heat transfer correlation.

A third simulation alternative for thermal analysis of compressors adopts a differential formulation to solve both the conduction heat transfer and the gas fluid flow in the compressor components. For instance, Chikurde *et al.* (2002)

numerically analyzed the fluid flow and heat transfer phenomena in a 1.5 ton A/C hermetic compressor using a commercial CFD code. A steady-state condition was used for the simulation, with the mass flow rate and suction gas temperature being prescribed at the inlet. A similar approach was used by Birari *et al.* (2006) to predict the temperature at various positions of a refrigeration compressor operating with R404a, by considering a steady state incompressible fluid flow. Heat sources inside the compressor like copper and iron losses in the motor, frictional losses and heat due to gas compression were considered. The main difficulty of such simulation methods is the modeling of the lubricating oil flow inside the compressor, which plays an important role in the heat transfer at several components of the compressor.

Hybrid simulation models have been proposed as a combination of differential and integral formulations so as to solve a conjugate heat transfer problem, usually for isolated components, such as suction and discharge mufflers, cylinder head and compression chamber. For instance, Almbauer *et al.* (2006) presented a model to numerically solve the temperature field of a compressor cylinder-piston system, by combining three approaches: i) a one-dimensional differential formulation for the fluid flow conservation equations; ii) a three-dimensional formulation for the conduction heat transfer in the cylinder-piston solid domain; iii) a lumped formulation for the thermal energy balance along the compressor solution domain, subdivided into mass elements. Ribas Jr. (2007) developed a method to solve the three-dimensional conduction heat transfer in the compressor crankcase, combined with an experimentally calibrated lumped model, similar to that of Todescat *et al.* (1992), to take into account the convective heat transfer between the gas and crankcase. Although the model is partially dependent on experimental data, it allows a thermal optimization for the components because the interaction between the components by conduction is accurately described.

Overall, the aforementioned three modeling approaches possess advantages and disadvantages. A limitation of the models proposed by Todescat *et al.* (1992) and Ooi (2003) is the impossibility of properly predict conduction heat transfer between the components. Although, currently available hybrid models can resolve conduction heat transfer in solid components, they need to be calibrated with reference to experimental data. Finally, fully differential models offer the greatest flexibility for analysis of layout modifications, since fluid and solid domains are solved without the necessity of prescribing heat transfer coefficients. However, the high computational processing time make such models unsuitable for optimization purpose. Moreover, some physical phenomena inside the compressor are difficult to solve even with this type of modeling, such as the flow of lubricating oil inside the compressor and its influence on the compressor thermal profile.

This paper presents a simulation model based on the hybrid model of Ribas Jr. (2007) that combines three-dimensional formulation for conduction heat transfer in solid components and a lumped formulation for the gas. However, instead of adopting equivalent thermal conductances derived from experimental data, the model described herein characterizes convective heat transfer coefficients with reference to correlations available in the literature in a similar manner put forward by Padhy (1992), Sim *et al.* (2000) and Ooi (2003). The new model approach is devised to offer greater flexibility for thermal analysis of compressor designs, including the use of different materials, operating conditions and geometric configurations.

2. SIMULATION MODEL

Correlations are adopted to prescribe convective heat transfer coefficients in different components of the compressor in contact with gas and lubricating oil. Such coefficients are boundary conditions for the three-dimensional conduction solution domain of solid components and adopted in the balance of energy of lumped control volumes adopted for the gas and lubricating oil. Heat released during to the cylinder wall during the compression cycle affects the compressor temperature distribution which in turn affects the amount of gas admitted to the cylinder and the energy used to compress the gas. Simulation of the compression cycle, including the modeling of mass flow rate through valves and energy consumption, is carried out with the model described in Deschamps *et al.* (2002).

As previously mentioned, the model adopts a lumped formulation for energy balances in control volumes specified for regions of gas and oil, whereas a differential formulation to solve conduction heat transfer in solids components. As far as the lumped formulation is concerned, each control volume is analyzed separately and encompasses the entire path of the gas inside the compressor. The energy balance for a typical control volume of fluid considers advection and convection of energy assuming steady state, i.e.:

$$\sum (mh)_{in} - \sum (mh)_{out} = Q_{CV} \quad (1)$$

where $\sum (mh)_{in}$ and $\sum (mh)_{out}$ are the total input and output of energy by advection and Q_{CV} is the convection heat transfer at the surface of the control volume. Figure 1 depicts a simplified scheme of the control volumes included in the model. It should be noticed that the simulation model adopts the actual geometry, such as volume and surface areas, of all compressor components.

Convection heat transfer rate in each control volume is written in the following form:

$$Q_{CV} = H_i A_i (T_{gas} - T_{wall}) \quad (2)$$

where H_i is the convective heat transfer coefficient determined from literature correlations, A_i is the heat transfer area, T_{gas} is the gas temperature inside the component and T_{wall} is the surface temperature obtained by solving conduction heat transfer in the solid components via the finite volume method.

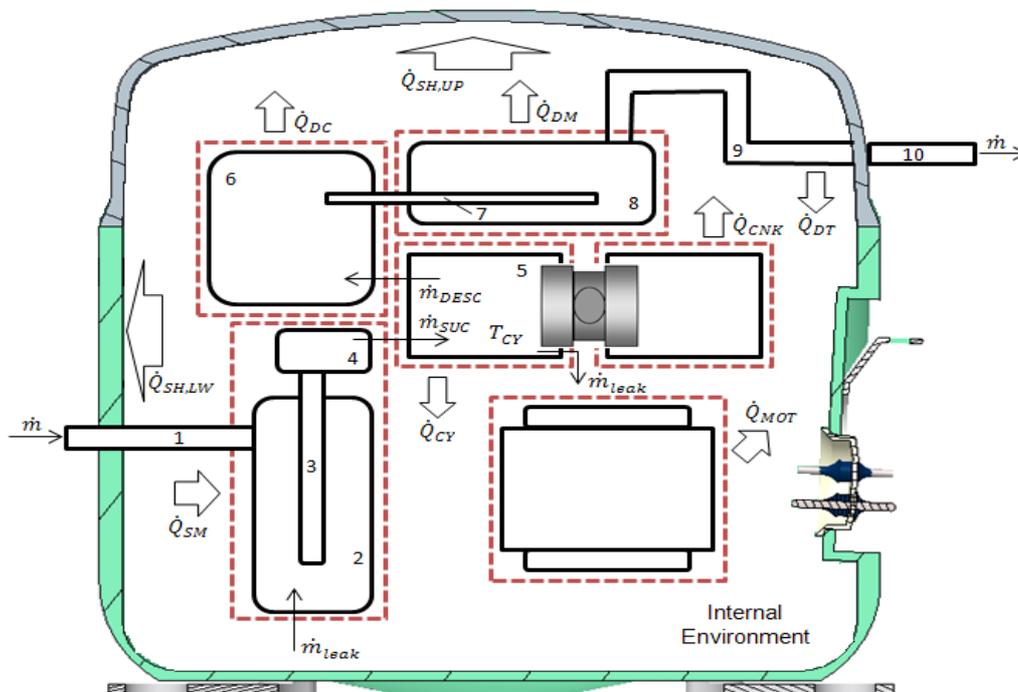


Figure 1: Schematic view of the compressor and the control volumes adopted in the analysis.

The path of the gas inside the compressor occurs in the following order: (1) suction tube, (2) suction muffler volume, (3) suction muffler tube, (4) suction chamber, (5) cylinder, (6) discharge chamber, (7) communication tube, (8) discharge muffler, (9) discharge tube and (10) outlet tube. All such components were considered in energy balances for the gas. Moreover, the presence of lubricant oil was also taken into account in the thermal modeling of the inner shell surface and other mechanical components, such as the crankcase and stator.

2.1 Convective Heat Transfer Coefficients

Several components of complex geometry are present in any design of reciprocating compressor. The first task in the modeling is to select convective heat transfer correlations available in the literature for simple geometries (flat plate, disk, cylinder, straight pipe, etc.) that can be suitable to predict convection heat transfer at the surface of the different compressor components. Some of the geometries were subdivided into simpler ones to allow a better description of the heat transfer process by using a combination of different correlations.

a) Flat plates: Correlations for this geometry are adopted for several components inside the compressor, such as the crankcase, stator, winding, cylinder head, external wall of the suction muffler, inner surface of the compressor shell for gas/oil interface, among others. The flow is considered to be turbulent due the random character of the gas flow inside the shell caused by the shaft motion and the presence of intricate component geometries. The Nusselt number is as follows:

$$\overline{Nu} = 0.037 Re_L^{0.8} Pr^{1/3} \quad (Pr \geq 0.5) \quad (3)$$

where Re_L is the Reynolds number and Pr is the Prandtl number. The characteristic length in the Reynolds number is the length in the flow direction. The characteristic velocity is assumed to be a fraction of tangential velocity of the shaft in all analyzed cases.

For the external surface of the compressor shell, a laminar convective heat transfer is assumed:

$$\overline{Nu} = 0.664 Re_L^{1/2} Pr^{1/3} \quad (Pr \geq 0.5) \quad (4)$$

b) External flow over cylinders: This type of flow is applied to represent heat transfer around the suspension spring and the crankpin where the corresponding correlation is given by:

$$\overline{Nu} = 0.3 + \frac{0.62 Re_D^{1/2} Pr^{1/3}}{\left[1 + (0.4 / Pr)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re_D}{282000}\right)^{5/8}\right]^{4/5} \quad (5)$$

where the Re_D is based in diameter of cylinder.

c) Rotating disc: This flow geometry was adopted to characterize heat transfer at the rotor surfaces, where an angular fluid velocity is induced, with the Nusselt number being expressed as:

$$\overline{Nu} = 0.33 Re_\Omega^{1/2} \quad (Pr \geq 0.7 ; 10^3 \leq Re \leq 2 \cdot 10^5) \quad (6)$$

where Re_Ω is the rotational Reynolds number, given by:

$$Re_\Omega = \frac{\Omega R^2}{\nu} \quad (7)$$

d) Gas flow through tubes: This type of flow was adopted to represent the convective heat transfer in the discharge system, represented by the discharge chamber, discharge muffler and communicating tubes. The correlation proposed is by Petukhov, which assumes moderate turbulence intensity, is employed:

$$\overline{Nu}_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{1/3} - 1)} \quad (Pr \geq 0.5; 3 \cdot 10^3 \leq Re \leq 5 \cdot 10^6) \quad (8)$$

where f is the friction factor:

$$\frac{1}{\sqrt{f}} = -2 \log \frac{\varepsilon / D}{3.7} \quad (9)$$

with ε being the surface roughness. In the presence of non-circular sections, the characteristic length in the Reynolds number is represented by the hydraulic diameter, D_h .

e) **Heat exchanger:** The concept of logarithmic temperature difference is adopted to evaluate the convective heat transfer coefficient in the suction muffler:

$$\overline{H} = \frac{mc_p}{PL} \ln \frac{(T_w - T_{out})}{(T_w - T_{in})} \quad (10)$$

where m is the flow mass rate, c_p is the specific heat at constant pressure and T_{out} and T_{in} are the outlet and inlet temperatures of the gas, respectively.

f) **Thin oil film:** An oil film is present over many components of the compressor, such as the internal surface of the compressor shell, crankcase and stator. The basic flow configuration is show in Fig.2, where one can notice an interface between oil and gas. The heat transfer coefficient for the gas/oil interface is evaluated via Eq. (3), whereas for the oil/solid interface the Nusselt number is estimated from a correlation developed for flow of thin oil film:

$$\overline{Nu}_\delta = \frac{H\delta}{k} = 0.029 \text{Re}_\delta^{0.533} \text{Pr}^{0.344} \quad (11)$$

where δ is the film thickness oil, with is calculated by an expression for descendants flux under the action of gravity, as:

$$\delta = \left(\frac{3\mu_o^2}{4\rho_o^2 g} \right) \text{Re}_\delta^{1/3} \quad (12)$$

The film Reynolds number is defined as:

$$\text{Re}_\delta = \frac{4\Gamma}{\mu_o} \quad (13)$$

where Γ is the oil mass flow rate per unit width.

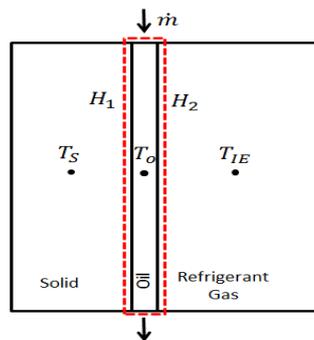


Figure 2: Control volume for oil flow along the shell and other solid components.

2.2 Energy Balance for Control Volumes

The control volumes considered in the integral formulation adopted for the gas are: internal volume of the compressor shell, suction muffler, suction chamber, discharge chamber, discharge muffler and several tubes (compressor inlet tube, muffler inlet tube, communicating tube, discharge muffler tube, discharge system tube and compressor outlet tube). The energy balance applied to each one of such control volumes bring about a system of equations described below.

a) Suction muffler: Applying the first law of thermodynamics to this control volume yields:

$$m_1(h_{SM} - h_{ST}) + m_2 h_{IE} = Q_{SM} = H_{SM} A_{SM} (T_S - T_{SM}) \quad (14)$$

where Q_{SM} is the convective heat transfer rate at the volume wall. Equation (14) includes the variation of energy caused by leakage of gas in the suction muffler.

b) Suction chamber: The main source of heat transfer at the wall of the suction chamber, Q_{SC} , is associated with the cylinder head.

$$(m_1 + m_2)(h_{SC} - h_{MT}) + m_3 h_{CY} = Q_{SC} = H_{SC} A_{SC} (T_S - T_{SC}) \quad (15)$$

Energy transfer due to backflow in the suction valve is also included in the energy balance.

c) Discharge chamber: The control volume representing the discharge chamber receives the gas at high temperature and pressure from the cylinder and releases heat at the wall, Q_{DC} :

$$m_4(h_{DC} - h_{CY}) + m_5 h_{DC} = Q_{DC} = H_{DC} A_{DC} (T_S - T_{DC}) \quad (16)$$

As for the suction chamber, backflow in the valve is also taken into account in the energy balance.

d) Discharge muffler: The energy balance for this control volume is given by:

$$m_4(h_{DM} - h_{CT}) = Q_{DM} = H_{DM} A_{DM} (T_S - T_{DM}) \quad (17)$$

e) Tubes: Several tubes are included in the analysis (suction tube, muffler tube, communicating tube, discharge tube and outlet tube) and a single formulation is adopted for the energy balance:

$$m_{gas}(h_{out} - h_{in}) = Q_{TUBE} = H_{TUBE} A_{TUBE} (T_{TUBE} - T_S) \quad (18)$$

For the discharge tube, an overall conductance, UA , is adopted to evaluate heat transferred through the wall to gas inside the volume of the compressor shell.

f) Internal volume of the compressor shell: This control volume is probably the most complex one since it includes several contributions to the energy balance, such as heat released by the bearings and electrical motor, and different walls in which heat is released, such as the compressor shell, discharge muffler, suction muffler and crankcase. Moreover, a small portion of energy is also transferred via leakage of gas through the clearance between piston and cylinder. The resulting energy balance is given by:

$$m_2(h_{IE,LW} - h_{SM}) = Q_{IE} = \sum_i^{N_1} H_{IE,i} A_{IE,i} (T_{S,i} - T_{IE,UP}) + \sum_j^{N_2} H_{IE,j} A_{IE,j} (T_{S,j} - T_{IE,LW}) \quad (19)$$

It should be mentioned that the internal volume of the compressor shell was subdivided into two regions, one for the lower region and other for the upper region, since temperature of the gas varies significantly inside the compressor shell.

2.3 Accounting for Lubricating Oil on Heat Transfer

Besides acting as a lubricating agent for bearings, the oil inside the compressor also affects heat transfer. In the reciprocating compressor selected for the present analysis, the lubricating oil stored in the sump is directed to the upper region inside the compressor via the action of a pump that is assembled to the crankshaft. As the shaft spins, the lubricating oil flows through the pump due to centrifugal action and leaves the other extremity as an impinging jet against the compressor shell. A portion of the oil returns to the sump by flowing over the crankcase and other portion flows as an oil film along the lateral surface of the shell.

The lubricating oil flow considerably affects the compressor thermal behavior, in terms of total heat loss and heat transfer distribution among the upper, side and bottom surfaces of the compressor shell, as show in Fig. (3) and (4). In order to include the presence of the oil in the model, it has been assumed that 80% of its mass flow rate is directed to the compressor shell and 20% over the crankcase.

Pizarro (2009) solved the flow and heat transfer associated with the lubricating oil by using a one-dimensional differential formulation. He observed that the oil temperature at different regions can be expressed as a percentage of the stator temperature, as follows: i) upper wall of the shell: 90%; ii) lateral wall of the shell: 85%; iii) oil sump: 80%; iv) wall of the crankcase: 90% and v) wall of the stator: 95%. These percentages were used in the present model to prescribe the oil temperature in such regions and are physically consistent with the energy balance for the oil domain.

3. SOLUTION PROCEDURE

The mathematical model is solved through an iterative procedure represented by tree steps: i) evaluation of the gas temperature by solving the equations associated with energy balances, ii) solution of the compression cycle and iii) solution of conduction in the solid components via the finite volume method. The system of equations are solved in a coupled manner, with convective heat transfer being used as boundary conditions for the differential and integral formulations adopted for the solid components and gas/oil flow, respectively. In addition to that, the temperature distribution is also used in the simulation of the compression cycle which in turn provides the amount of heat released at the cylinder wall.

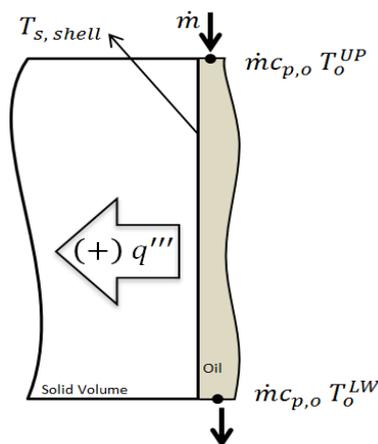


Figure 3: Scheme of energy balance for the oil along the shell.

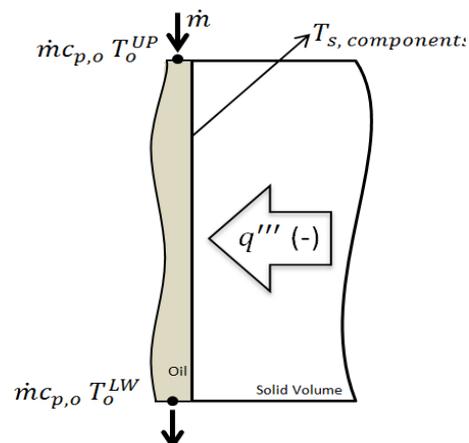


Figure 4: Scheme of energy balance for the oil over the crankcase and stator.

The only input parameters required in the model are the geometric data, evaporating and condensing temperatures. The iterative procedure includes an algorithm written with the C programming language for the integral formulation adopted in regions of gas/oil, described by equations (14) to (19), and a commercial code to solve conduction heat transfer in the solid components. The final solution provides the temperature distribution for the gas (T_{GST} , T_{GSM} , T_{GMT} , T_{GSC} , T_{SCW} , T_{GDC} , T_{GDM} , T_{GCT} , T_{GDT} , T_{GOT}), solid components (T_{CYS} , T_{OUTSH} , T_{STTS} , etc.) and lubricant oil ($T_{O,COV}$, $T_{O,LT}$, $T_{O,CRNK}$, $T_{O,MOT}$ and T_{OSP}).

4. RESULTS

A 60Hz reciprocating compressor operating with R134a was selected for the analysis, being submitted to three operating conditions, represented by the following pairs of evaporating and condensing temperatures: A (-23.3°C/40.5°C); B (-23.3°C/54.4°C); C (-10.0°C/90.0°C). The same compressor model was experimentally investigated by Dutra and Deschamps (2010) in a calorimeter, with measurements being provided for temperature in several regions inside the compressor. Such measurements are used in the present study to validate the model and to assess required adjustments.

The present model offers the capability of thermal optimization because it can predict the interaction between the components by conduction heat transfer, as the results for temperature distribution in solid components suggest (Figure 5). Therefore, the analysis of the impact of different materials and compressor layout on the compressor performance is a straightforward task.

Table 1 presents a comparison between numerical and experimental results for operating conditions A, B and C. As can be seen, there is a reasonable agreement between predictions and measurements, especially as far as the trend for temperature variation when the operating condition is modified. Nevertheless, the temperatures of some important components and regions (cylinder, stator and suction chamber) are not well predicted and deviations of up to 14.4°C are found. In fact, the temperature of the gas in the suction chamber considerably affects the compressor thermodynamic performance and the present model predicts a deviation of up to 8.4°C in relation to measurements.

The deviations observed in Table 1 do not come as a surprise considering the complexity of heat transfer inside the compressor. For instance, the geometries of several components do not comply with those assumed to specify correlations for convective heat transfer coefficients. This is the case for the suction muffler, discharge chamber and discharge muffler, where recirculating flow regions and high turbulence intensities are expected. Therefore, the model was adjusted in such regions by introducing multiplicative factors for the Nusselt number.

In order to explore the model sensitivity regarding modifications in the Nusselt number, a simple trial-and-error procedure was adopted to minimize the deviations between experimental and numerical results of temperature. Multiplicative factors for the Nusselt number equal to 2, 4 and 14 were found for the suction muffler, discharge chamber and discharge muffler, respectively. As can be seen in Table 2, such adjustments significantly improve the agreement between numerical and experimental results. For instance, the maximum deviation is reduced to 7.9°C at the external surface of the compressor shell (T_{GOS}). Considering the cylinder, stator and suction chamber, the deviations are decreased to 6.0°C, 5.4°C and 0.8°C, respectively. These results suggest that it should be worthwhile a better characterization of heat transfer in such complex geometries via numerical analysis.

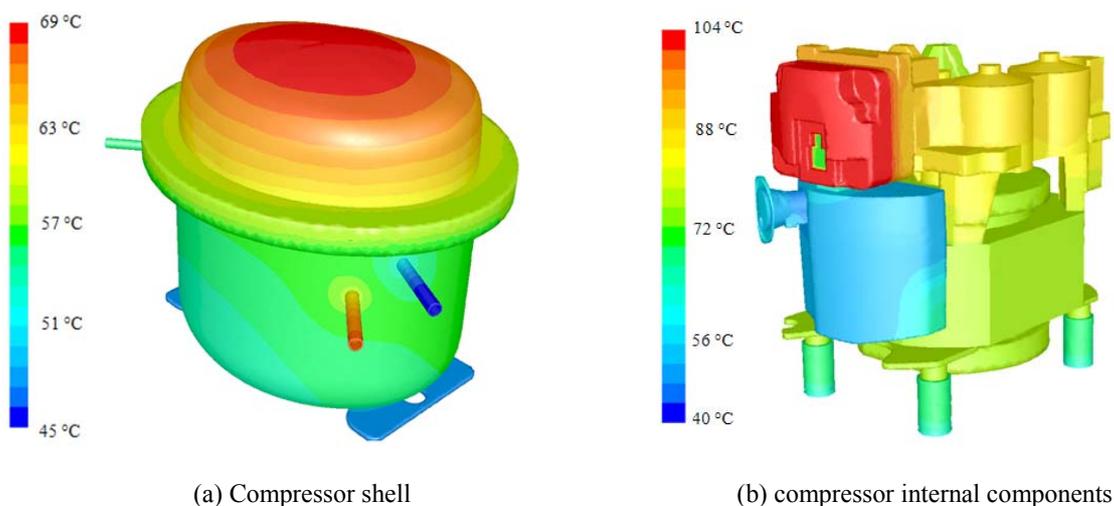


Figure 5: Temperature distribution in solid components.

Table 1: Thermal profile comparison in case of correlation forms normal.

Temperatures	Condition A			Condition B			Condition C		
	Exp.	Num.	Δ	Exp.	Num.	Δ	Exp.	Num.	Δ
Gas in Suction tube (T_{GST})	-	33.8	-	-	34.2	-	-	43.0	-
Gas in Suction muffler (T_{GSM})	-	38.4	-	-	38.6	-	-	40.5	-
Gas in Muffler tube (T_{GMT})	-	37.8	-	-	38.2	-	-	49.1	-
Gas in Suction chamber (T_{GSC})	46.6	44.8	-1.8	49.2	45.2	-4.0	57.0	49.1	-7.9
Cylinder surface (T_{CYS})	91.9	81.2	-10.7	102.4	88.0	-14.4	124.4	113.7	-10.7
Gas in Discharge chamber (T_{GDC})	116.7	119.6	2.9	129.9	130.8	0.9	157.9	156.0	-1.9
Gas in Discharge muffler (T_{GDM})	99.0	96.7	-2.3	108.9	105.4	-3.5	135.3	132.7	-2.6
Gas in Communicating tube (T_{GCT})	-	113.9	-	-	124.5	-	-	150.6	-
Gas in Discharge tube (T_{GDT})	-	92.01	-	-	92.0	-	-	103.9	-
Gas in Outlet tube (T_{GOT})	-	89.5	-	-	89.7	-	-	102.6	-
Outer shell surface (T_{OUTSH})	63.8	58.7	-5.1	68.2	62.3	-5.9	84.8	85.3	0.5
Stator surface (T_{STTS})	80.2	73.4	-6.8	86.6	79.8	-6.8	109.4	113.6	4.2
Gas in the Upper internal shell ($T_{IE,UP}$)	78.8	77.4	-1.4	81.6	83.2	1.6	108.3	113.2	4.9
Gas in the Lower internal shell ($T_{IE,LW}$)	65.2	65.5	0.3	72.4	72.7	0.3	102.1	109.6	7.5
Oil sump (T_{OSP})	65.1	58.8	-6.3	69.4	63.8	-5.6	87.1	90.9	3.8

Table 2: Thermal profile in case of correlation form modified.

Temperatures	Condition A			Condition B			Condition C		
	Exp.	Num.	Δ	Exp.	Num.	Δ	Exp.	Num.	Δ
Gas in Suction tube (T_{GST})	-	33.8	-	-	34.0	-	-	35.2	-
Gas in Suction muffler (T_{GSM})	-	39.7	-	-	40.3	-	-	43.0	-
Gas in Muffler tube (T_{GMT})	-	41.0	-	-	42.4	-	-	49.1	-
Gas in Suction chamber (T_{GSC})	46.6	47.4	0.8	49.2	48.6	-0.6	54.7	54.0	-0.7
Cylinder surface (T_{CYS})	91.9	89.8	-2.1	102.4	96.4	-6.0	124.4	121.7	-2.7
Gas in Discharge chamber (T_{GDC})	116.7	117.4	0.7	129.9	127.3	-2.6	157.9	155.2	-2.7
Gas in Discharge muffler (T_{GDM})	-	112.5	-	-	121.9	-	-	150.0	-
Gas in Communicating tube (T_{GCT})	99.0	97.2	-1.8	108.9	104.4	-4.5	135.3	133.0	-2.3
Gas in Discharge tube (T_{GDT})	-	92.0	-	-	92.0	-	-	103.8	-
Gas in Outlet tube (T_{GOT})	-	89.4	-	-	89.6	-	-	102.5	-
Outer shell surface (T_{OUTSH})	63.8	57.2	-6.6	68.2	60.3	-7.9	84.8	84.1	-0.7
Stator surface (T_{STTS})	80.2	75.8	-4.4	86.6	81.2	-5.4	109.4	114.1	4.7
Gas in the Upper internal shell ($T_{IE,UP}$)	78.8	77.3	-1.5	81.6	81.9	0.25	108.3	111.9	3.6
Gas in the Lower internal shell ($T_{IE,LW}$)	65.2	65.4	0.2	72.4	71.1	-1.3	102.1	108.0	5.9
Oil sump (T_{OSP})	65.1	60.6	-4.5	69.4	64.9	-4.5	87.1	91.3	4.2

5. CONCLUSIONS

This paper presented a numerical model for thermal analysis of reciprocating compressor capable of estimating the temperature of the gas, lubricating oil and solid components, in several regions inside the compressor. The model adopts a three-dimensional formulation for conduction in solid components and convective heat transfer coefficients to characterize heat transfer between gas/oil and solid components. In principle, the model does not require calibration based on experimental data, making it possible to analyze modifications in the compressor design and a wide range of operating conditions. Predictions were compared to experimental data and reasonable agreement was verified, particularly for temperature variations due to change of operating condition.

NOMENCLATURE

Q	heat transfer rate	(W)	Subscripts	
m	mass flow rate of refrigerant	(kg/s)	L	characteristic length
m_1	inlet mass flow rate of refrigerant	(kg/s)	o	lubricant oil
m_2	leak mass flow rate of refrigerant	(kg/s)	S	solid surfaces
m_3	backflow mass flow rate of refrigerant	(kg/s)	SH	shell
m_4	discharge mass flow rate of refrigerant	(kg/s)	UP	upper region
m_5	backflow mass flow rate of refrigerant	(kg/s)	LW	lower region
h	specific enthalpy	(J/kg)	IE	internal environment
T	temperature	(K)	CY	compression cylinder
Nu	nusselt number	(-)	CRNK	crankcase
Ω	angular velocity	(rad/s)	MOT	motor
ν	kinematic viscosity	(m ² /s)	COV	inner of the cover shell
μ	dynamic viscosity	(Pa.s)	LT	lateral of the cover shell
ρ	density	(kg/m ³)	ST	suction tube
D	diameter	(m)	SM	suction muffler
P	perimeter	(m)	SC	suction chamber
L	length	(m)	DC	discharge chamber
g	gravity acceleration	(m/s ²)	CT	communication tube
			DM	discharge muffler

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