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ABSTRACT

Comprehensive knowledge about the heat transfer mechanisms and the temperature field inside hermetic compressors is very important for the thermal management and thus their performance. A numerical model to predict the temperature field in a hermetic reciprocating compressor for household refrigeration appliances is presented in this work. The model combines a high resolution three-dimensional heat conduction formulation of the compressor's solid parts, a three-dimensional computational fluid dynamics (CFD) approach for the gas line domain and lumped formulations of the shell gas and the lubrication oil. Heat transfer coefficients are determined by applying CFD to the gas line side and correlations from the literature on the shell gas and oil side, respectively. The valve in the gas line simulation is modelled as a parallel moving flat plate. By means of an iterative loop the temperature field of the solid parts acts as boundary condition for the CFD calculation of the gas line which returns a cycle averaged quantity of heat to the solid parts. Using an iteration method which is based on the temperature deviation between two iteration steps, the total number of iterations and consequently the computational time can be reduced. The loop is continued until a steady-state temperature field is obtained. Calculated temperatures of the solid parts are verified by temperature measurements of a calorimeter test bench. The numerical results show reasonable agreement with the measured data.

1. INTRODUCTION

Thermal management is one of the main topics in the development process of modern hermetic reciprocating compressors for household refrigeration application. Due to the hermetic design of the compressor, electrical and mechanical losses influence the thermodynamic efficiency of the compressor. Heat transfer mechanisms like convection and conduction determine the temperature field inside the compressor and consequently, the compressor performance. The knowledge of the temperature field inside the compressor is essential to quantify loss mechanisms like superheating.

One possibility to obtain the heat transfer and the temperature field inside the compressor is the experimental investigation using heat flux sensors (Dutra and Deschamps, 2010) or thermocouples (Kara and Oguz, 2010). The spatial resolution of these methods is low, they are not suitable for compressors in the design phase and the usage of sensors may affect the compressor behaviour. Another possibility to investigate the thermal performance of a hermetic reciprocating compressor is the use of simulation tools. Several strategies to model the temperature field and heat transfer mechanism inside the compressor have been developed and can be found in the open literature.

Simple approaches split the compressor into several lumped volumes using the first law of thermodynamics to calculate the temperature field (e.g. Meyer and Thompson, 1988; Todescat *et al.*, 1992). Such models use either experimental data or correlations for the convective heat transfer formulation. The lumped volumes are connected via thermal conductance adjusted to experimental data. The low spatial resolution of these models and

the dependence on experimental data yield a very rough and inflexible estimation of the temperature field in a hermetic reciprocating compressor.
A more flexible modelling strategy is the Thermal Network approach (TNW). The usage of TNW for thermal modelling of a hermetic reciprocating compressor for refrigeration application can be found in e.g. Sim *et al.* (2000) and Ooi (2003). TNW uses mass points to model the considered compressor parts and the heat transfer between the mass points is represented with the Lumped Conductance Method. Convective heat transfer is modelled with correlations based on forced or natural convection Nusselt number. Due to the reduction of geometrical information, the heat transfer modelling is characterized by a high level of uncertainty especially for regions with transient 3d flows like in the suction or discharge line. Also the validation of TNW with experimental data can cause problems because it is not clear if the chosen measuring points represent the temperature of the lumped mass.
The use of Computational Fluid Dynamics (CFD) in the development process of hermetic reciprocating compressors leads to another approach for thermal modelling, the so called hybrid simulation models. Although the performance of CPUs increased significantly over the last years, an overall 3d simulation of a compressor is still not possible within a reasonable time. Hybrid models use different combinations of complex 3d formulations and simple correlations for convective and conductive heat transfer, respectively. Almbauer *et al.* (2006) applied 1d flow simulation of the gas line, 3d formulation of the cylinder solid domain and lumped formulation of the remaining compressor parts. Ribas (2007), Sanvezzo and Deschamps (2012) and Lohn *et al.* (2015) combined 3d heat conduction formulation for the solid parts of the compressor and lumped formulation of the gas path. The authors used either experimental data or correlations from the literature to model the heat transfer.
The model in the present study is based on the hybrid approach. Compared to the hybrid models found in the literature, this work contains 3d formulation of the full gas line and 3d formulation of solid components of the entire compressor in combination with lumped volume formulation of the gas inside the compressor shell and the lubrication oil. The simulation of the fluid flow in the gas line considers 3d phenomena in the suction and discharge mufflers, the flow in the cylinder as well as interactions with the compressor valves. Special focus is laid on the high geometric resolution of the solid compressor parts. A simulation algorithm is presented to combine the transient flow calculation and the steady-state heat conduction calculation to model the thermal behaviour of a hermetic reciprocating compressor running at calorimeter test conditions. The simulation results are verified by experimental data of a calorimeter test bench.

2. SIMULATION MODEL

The R600a (isobutane) hermetic reciprocating compressor used in the present study has a displacement of 5.5 ccm and the COP (ASHRAE test conditions -23.3 °C/55 °C) is approximately 1.8. A schematic view of the compressor is shown in Figure 1. The simulation model can basically be split up into three main parts: (i) simulation of the gas flow using commercial CFD software, (ii) heat conduction in the solid parts also using commercial CFD software and (iii) the energy balance for oil and refrigerant by assuming lumped control volumes. The three simulation parts exchange data in terms of heat flux and temperatures.

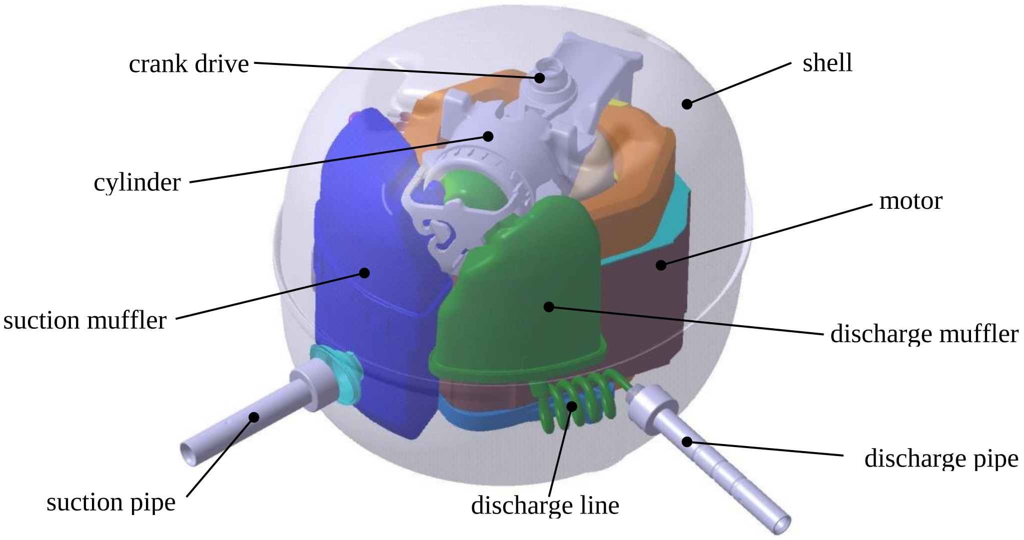

Figure 1: Schematic view of the investigated compressor

2.1 Gas flow

The simulation of the gas flow is done with the commercial CFD software package ANSYS Fluent. The simulation domain includes suction line, suction muffler, valves, cylinder, ports, discharge muffler and discharge line. To reduce the computational time the valves are assumed to be parallel moving flat plates. The valve motion is calculated via User Defined Functions solving the single-degree of freedom system considering oil stiction forces. The number of cells in the computational mesh varies between 3.1 and 5.5 million cells depending on the position of the piston. The k-ε turbulence model is used and 1^{st} order spatial discretization of density, momentum, energy and turbulence is applied. Pressure-velocity coupling is used with coupled solver setting.

2.2 Conduction in the solid parts

Heat conduction in the solid parts of the hermetic compressor is also solved with the commercial software package ANSYS Fluent. The whole domain is meshed with about 2.4 million cells with refinements at the interfaces between the single parts. Heat flux boundary conditions are applied on the surfaces of the solid parts depending on the flow conditions. Surfaces that are in contact with the gas flow use area-weighted average heat flux values determined by the CFD calculation of the gas flow. Heat flux values for the convective heat transfer between the solid parts, the compressor oil and the gas inside the shell are calculated with correlations from the literature (e.g. flow over flat plates, flow inside pipes). The heat flux between the compressor shell and the ambient air is modelled by means of a natural convection correlation. Electrical power losses are considered as volumetric source terms in the energy equation of the corresponding part. Mechanical power losses are considered as heat input to the compressor oil (Zach, 2013).

2.3 Lumped control volumes

To complete the thermal modelling of the entire hermetic compressor, the refrigerant and the oil inside the compressor shell have to be taken into account. The gas caught inside the shell is regarded as one control volume. Leakage mass flow rates are neglected and the gas is assumed to be in thermal equilibrium. The sum of the heat fluxes between the shell gas and the adjacent parts $\dot{Q}_{SG}$ are calculated with the results of the solid part conduction simulation. The temperature of the gas can be calculated by applying the energy balance to the control volume:

$$T_{SG}^{n+1} = T_{SG}^{n} + \frac{\dot{Q}_{SG} \cdot \Delta t}{m_{SG} \cdot c_{v\,SG}} \tag{1}$$

The oil inside the compressor shell is divided into three control volumes, namely the oil in the sump, the oil transported by the oil pump and the expelled oil on the oil covered walls. The oil mass flow rate at steady-state operating conditions is determined by experiments (Posch *et al.*, 2015). Applying a first order upwind scheme for the calculation of the outlet enthalpy flow, the energy balance of the considered control volume yields the following equations for the oil temperatures:

$$T_{OS}^{n+1} = \frac{\dot{Q}_{OS} + c_{oil}\left(\dot{m}_{oil} \cdot T_{OW} + m_{OS}\dfrac{T_{OS}^{n}}{\Delta t}\right)}{c_{oil}\left(\dot{m}_{oil} + \dfrac{m_{OS}}{\Delta t}\right)} \tag{2}$$

$$T_{OC}^{n+1} = \frac{\dot{Q}_{OC} + c_{oil}\left(\dot{m}_{oil} \cdot T_{OS} + m_{OC}\dfrac{T_{OC}^{n}}{\Delta t}\right)}{c_{oil}\left(\dot{m}_{oil} + \dfrac{m_{OC}}{\Delta t}\right)} \tag{3}$$

$$T_{OW}^{n+1} = \frac{\dot{Q}_{OW} + c_{oil}\left(\dot{m}_{oil} \cdot T_{OC} + m_{OW}\dfrac{T_{OW}^{n}}{\Delta t}\right)}{c_{oil}\left(\dot{m}_{oil} + \dfrac{m_{OW}}{\Delta t}\right)} \tag{4}$$

2.4 Solution algorithm

At the beginning of the simulation process an initial temperature field in all solid parts, refrigerant and oil in the compressor is guessed. The solid part temperatures are used as boundary conditions for the following CFD simulation of the gas flow. The results of the CFD simulation are averaged over one rotation of the compressor and are considered as heat flux boundary conditions for the simulation of the solid part heat conduction. Since a steady-state temperature field of the compressor should be calculated, the solid part heat conduction model in ANSYS Fluent is set to steady. After calculating the temperature distribution in the solid parts, the energy balance for the refrigerant in the shell and the respective oil volumes are carried out using the CFD heat fluxes. The lumped control volumes are modelled as quasi-transient using a time step of 1s. The boundary conditions for the solid part conduction simulation are updated with the gas and oil temperatures. Lumped volume energy balance and solid part conduction forms an iteration loop which is continued until heat flux deviation between two consecutive iteration steps is below a certain limit. Using the new solid part temperature field the CFD simulation is carried out again. The iteration loop between CFD simulation of the gas line and the solid part heat conduction calculation is executed until heat flux deviations between two consecutive iteration steps are also below a certain convergence criteria.

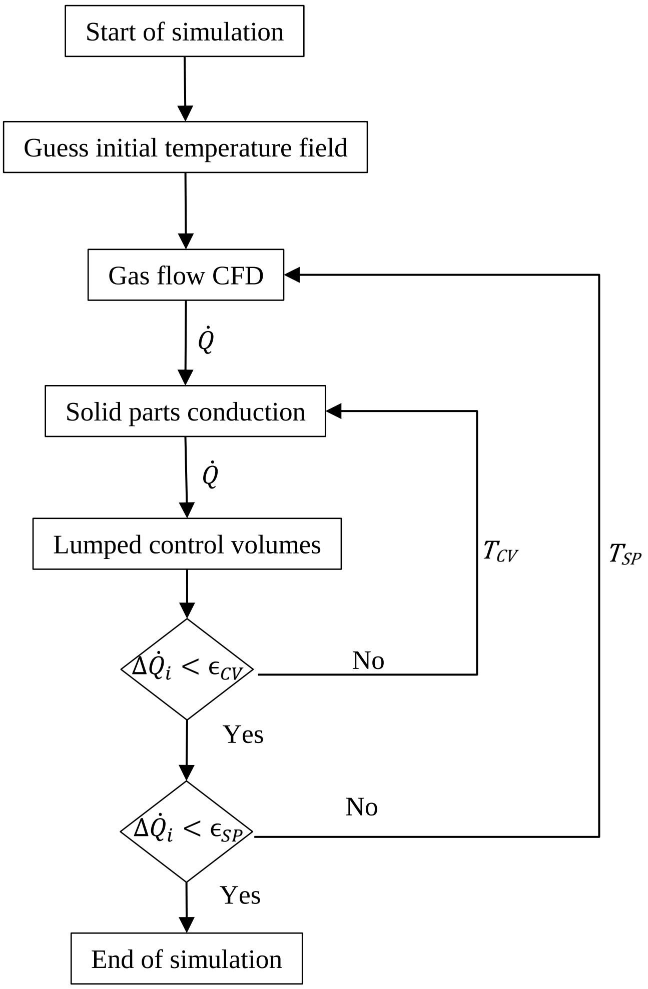

Figure 2: Flow chart of the solution algorithm

3. EXPERIMENTAL WORK

To validate the thermal modelling of the hermetic compressor, temperature measurements on several positions in the compressor are carried out. 22 thermocouples type T are distributed inside the compressor, on the shell and in the gas line. Furthermore, two pressure sensors are placed in the suction and discharge muffler. A special sealed feedthrough is used to pass the Teflon®-insulated sensor wires through the hermetic shell. Table 1 gives an overview of the thermocouples which are used for the validation of the present thermal model. A schematic overview of the sensor positions is illustrated in Figure 3.

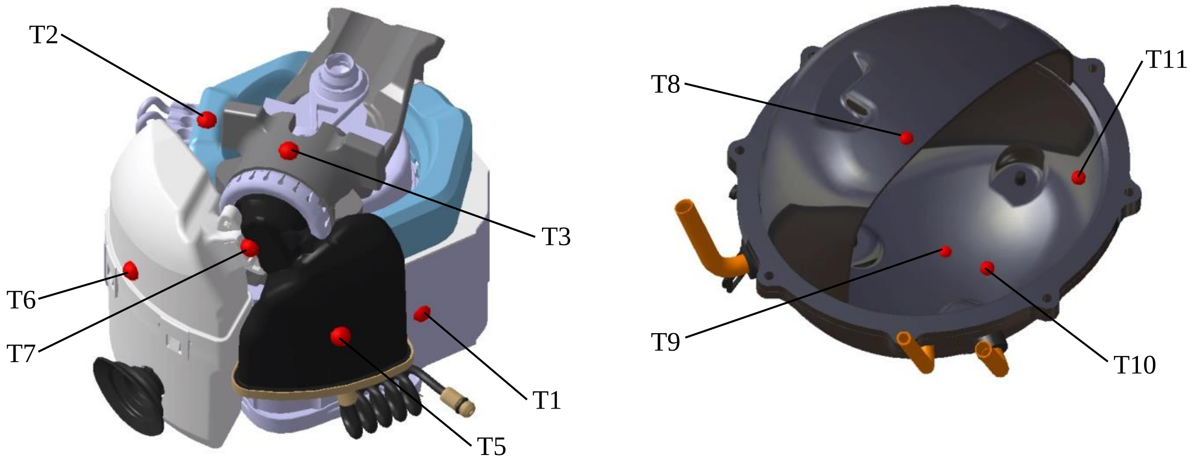

Figure 3: Schematic overview of the sensor positions

Table 1: Thermocouple measuring points

No.	Measuring point description
T1	Stator lamination (surface)
T2	Stator windings (surface)
T3	Cylinder (surface)
T4	Shell (gas)
T5	Discharge muffler (surface)
T6	Suction muffler (surface)
T7	Valve plate (surface)
T8	Shell top outside (surface)
T9	Shell bottom outside (surface)
T10	Oil sump
T11	Shell inside (surface)

4. RESULTS

The presented thermal model is used to simulate the temperature distribution in solid parts, gas and oil of a hermetic reciprocating compressor. The operating conditions of the compressor are set to -23 °C evaporating temperature, 45 °C condensing temperature and 32°C ambient temperature, respectively. The compressor works at constant speed of 2950 rpm.

Figure 4 shows exemplarily the velocity field in the cylinder of the gas flow obtained by CFD simulation. Regions with high local gradients of the heat transfer coefficient like suction and discharge port are treated as separated areas in the iteration loop between gas flow simulation and solid part conduction. Specific heat flux values for the individual regions can be seen in Table 2. The sign of the values is related to the heat flux direction from solid part to gas flow. The specific heat flux acts as boundary condition for the heat conduction simulation of the solid parts.

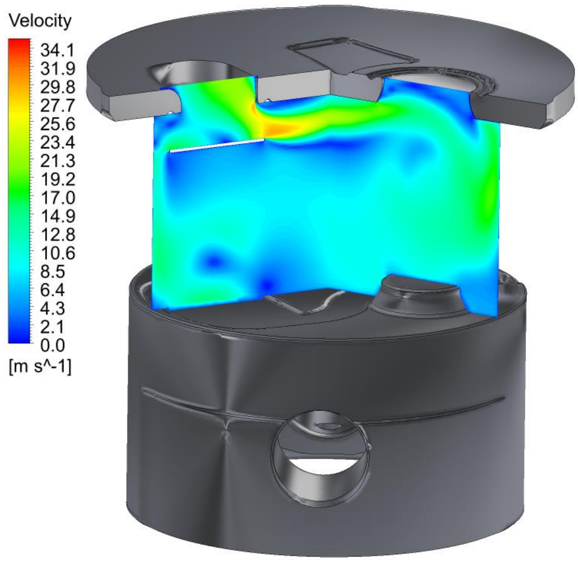

Figure 4: CFD simulation of the gas flow

Table 2: Specific heat flux from solid to gas [W/m²]

Solid	Value
piston	-7199
discharge muffler	-1289
suction muffler	351
valve plate cylinder	239
cylinder	2082
valve plate suction	414
valve plate discharge	-24421
discharge port	-10216
suction port	7485
serpentine	-3799

As described in chapter two the present model consists of two iteration loops. The outer iteration loop links the gas flow simulation with the simulation of the solid parts, shell gas and oil. The inner iteration loop links the solid part conduction with the energy balance of the lumped volumes of gas and oil in the compressor shell. The calculation time of the whole procedure is significantly depending on the outer iteration loop due to the time-consuming CFD simulation of the gas flow. To contain the number of iterations for the outer loop, the initial temperature field of the compressor has to be guessed well. In the present study, five iterations for the outer loop had to be carried out to fulfil the convergence criteria. Due to the fast calculation of the conduction in the solids and of the energy balance, the inner iteration loop has no significant impact on the overall computation time. The number of iterations for the inner loop in the present study is between 50 and 60. Figure 5 shows the temperature development over the number of iterations of the lumped volumes during the inner iteration loop.

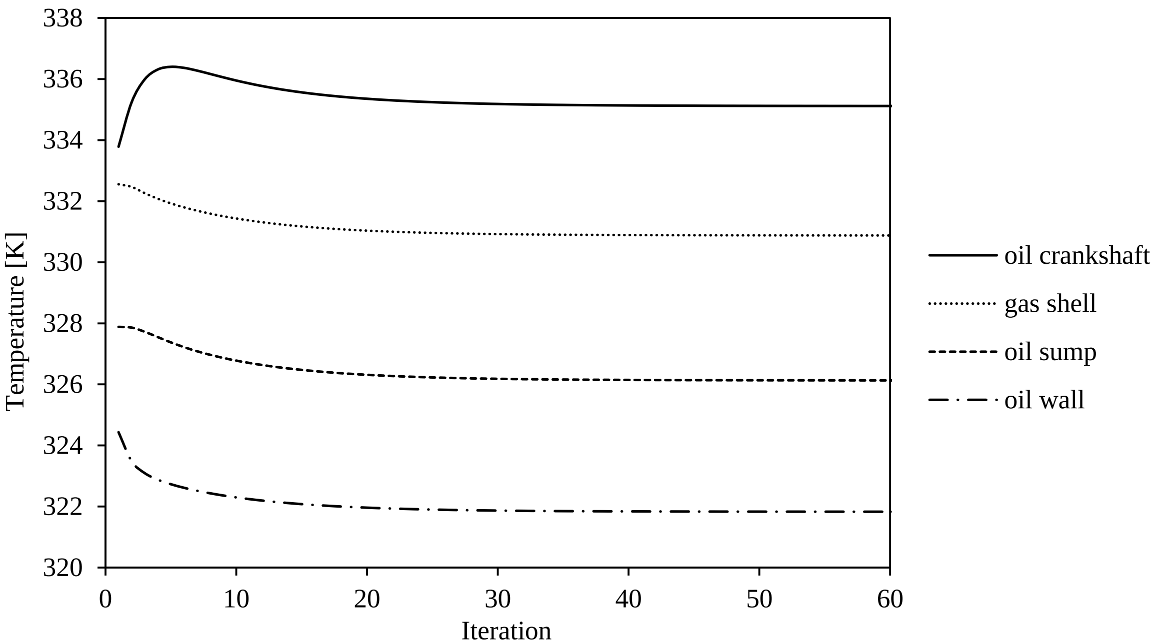

Figure 5: Temperature development during the inner iteration loop [K]

The comparison between the simulated and measured temperatures at the specific measuring points is illustrated in Table 3. The results of the thermal model are in agreement with calorimeter test data. Maximum absolute temperature difference between simulation and measurement is less than 4.2 K. Especially the temperatures of the solid parts inside the compressor and the lumped volumes are met very well. Although the thermal modeling of the hermetic compressor in the present study matches the experiment well, the uncertainties in the determination of the heat transfer coefficients should be discussed briefly. In general heat transfer can be modelled in two different ways: On the one hand by detailed CFD simulation of the complex gas flow inside the shell or by means of empirical correlations. With the CFD approach not only the model accuracy should increase but also the computation effort. If choosing empirical correlations a bad agreement between simulation and experimental data could turn out, that requires a slight calibration of the empirical correlations by experimental data. The temperature distribution in the solid parts of the compressor can be seen in Figure 6.

Table 3: Comparison between measured and simulated temperatures [K]

No.	Measuring point description	Measurement	Simulation	ΔT
T1	Stator lamination (surface)	332.9	333.4	0.5
T2	Stator windings (surface)	333.7	333.4	0.3
T3	Cylinder (surface)	337.7	338.3	0.6
T4	Shell (gas)	331.6	330.9	0.7
T5	Discharge muffler (surface)	350.9	350.9	0.0
T6	Suction muffler (surface)	328.2	326.8	1.4
T7	Valve plate (surface)	345.0	342.6	2.4
T8	Shell top outside (surface)	318.9	321.2	2.3
T9	Shell bottom outside (surface)	321.8	322.0	0.2
T10	Oil sump	327.8	326.1	1.7
T11	Shell inside (surface)	324.7	320.5	4.2

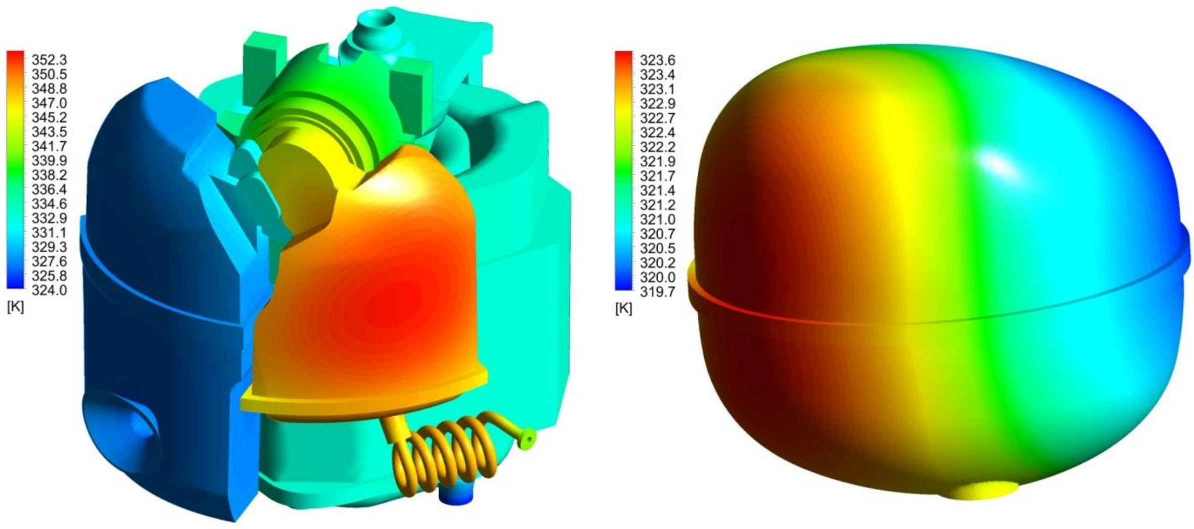

Figure 6: Temperature field of the solid parts

5. CONCLUSION

In this study a thermal model for the investigation of a hermetic reciprocating compressor is shown using mainly numerical methods. The method combines CFD simulation of the gas flow, numerical calculation of the heat conduction in the solid parts and simple lumped formulations of the refrigerant and oil in the compressor shell. Although some heat transfer coefficients are determined by simple correlations from literature, the results of the thermal modelling are in good agreement with experimental data gained by calorimeter tests. The present method is an advisable tool in the compressor development process along with detailed CFD simulation of the gas flow to validate thermodynamic compressor behaviour. Unavoidable uncertainties in the determination of the heat transfer coefficients between the solid parts and the fluids (refrigerant and oil) have to be considered in the evaluation of the results. To get a thermal model for the usage in thermodynamic parameter studies it is useful to calibrate the heat transfer coefficients with experimental data.

NOMENCLATURE

c	specific heat capacity	(J/kg K)	OS	oil in the sump	
$\dot{m}$	mass flow rate	(kg/s)	OW	oil on the walls	
m	mass	(kg)	oil	compressor oil	
n	iteration step		SG	shell gas	
$\dot{Q}$	heat flux	(W)	SP	solid parts	
T	temperature	(K)	v	isochoric	
t	time	(s)			
Subscripts			**Greek symbols**		
CV	control volumes		Δ	difference	(-)
OC	oil in the crankshaft		ε	convergence criteria	(-)

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