A thermal model for analysis of hermetic reciprocating compressors under the on-off cycling operating condition

S K Lohn, M C Diniz and C J Deschamps
POLO Research Laboratories
Federal University of Santa Catarina, 88040900, Florianopolis, SC, Brazil

E-mail: deschamps@polo.ufsc.br

Abstract. The on-off cycling operating condition of compressors is very common in low capacity refrigeration systems, being characterized by alternate periods in which the compressor is either operating (on) or idle (off). Thermal interactions between the compressor components affect its performance during the operating period and establish the initial condition for the compressor start up from idle condition. This paper presents a numerical model to predict the temperature field of hermetic reciprocating compressors under on-off cycling conditions. The model adopts a lumped formulation for control volumes formed in the fluid solution domain and the finite volume method to solve heat conduction in the solid components. Some required heat transfer coefficients were experimentally adjusted. Predictions for temperature were compared to measurements and good agreement was observed, especially for the thermal transient during the period in which the compressor is off.

1. Introduction
Reciprocating compressors are widely applied in low capacity refrigeration applications, commonly associated with an on-off cycling operating condition characterized by alternate periods in which the compressor is either operating (on) or idle (off).

Due to economic and environmental issues, the refrigeration industry is oriented towards performance improvement, thus requiring the analysis of aspects related to the transient operation of the system. The current literature related to transient system conditions is focused on the heat exchangers and expansion device [1]. Therefore, it is necessary to improve the knowledge on the compressor performance during the on-off cycling condition in order to increase the system efficiency. Thermal interactions between internal components of hermetic reciprocating compressors strongly influence their thermodynamic efficiency, as well their reliability. On the other hand, thermal interactions during the off period define the initial condition for the compressor start-up.

Heat transfer inside hermetic compressors has been studied both experimentally [2] and numerically [3]. Despite the good insight provided by experimentation, the use of numerical approaches presents significant benefits, such as versatility and fast results. Heat transfer in hermetic compressors is modelled employing different degrees of complexity, and a wide variety of steady state models is available in the literature [4]. Simple semi-empirical models, such as the one developed by Todescat et al. [5], provide very fast predictions of the temperature field. Other models, such as the totally differential model of Raja et al. [6] are characterized by high computational cost, although

1 To whom any correspondence should be addressed.
allowing deeper understanding of the phenomena. As a compromise, Ribas Jr. [7] developed a hybrid model that combines a differential approach for the solid domains and a lumped formulation for the gas chambers.

This paper presents a numerical model to predict the temperature field of a hermetic reciprocating compressor under on-off cycling conditions, based on the hybrid approach [7]. Predictions for temperature in different regions inside the compressor under the transient condition were validated through comparisons with measurements.

2. Compression cycle model

The thermal model requires input data related to the compression cycle. The modelling of the compression cycle employed in this work is based on four groups of equations [5], following a transient lumped formulation. The time scale of the compression cycle (milliseconds) is considerably smaller than that of the transient temperature field of solid components (minutes). Hence, the results obtained through the compression cycle model are integrated over a representative cycle to be used as input data in the thermal model.

The first group of equations calculates the instantaneous volume of gas inside the cylinder with respect to the compressor geometrical parameters. The second group of equations are the conservation equations (mass and energy) applied to the gas inside the cylinder. With the instantaneous cylinder volume and the mass of gas inside the cylinder, obtained through equation (1), it is possible to calculate the gas density.

$$\frac{\partial m_G}{\partial t} = \dot{m}_{in} - \dot{m}_{out, b} + \dot{m}_{dis} + \dot{m}_{dis, b} - \dot{m}_{leak},$$

(1)

The manipulation of the lumped energy balance originates the following equation to predict the gas temperature during the compression cycle:

$$\frac{dT_G}{dt} = \frac{1}{m_G c_v, G} \left[ H_G A_w T_w - h_G \frac{\partial m_G}{\partial t} + \sum_{in} \dot{m}_h - \sum_{out} \dot{m}_h \right]$$

$$- \frac{T_G}{m_G c_v, G} \left[ H_G A_w + \frac{\partial p_G}{\partial T_G} \frac{d\gamma_G}{dt} - \frac{\partial p_G}{\partial T_G} v_G \frac{dm_G}{dt} \right],$$

(2)

where $H_G$ is the convective heat transfer coefficient estimated with the correlation of Annand [8]. Two temperatures are very important in equation (2): the refrigerant temperature in the suction chamber ($T_{in}$) and the temperature of the cylinder wall ($T_w$). These temperatures can either be specified from experimental data or calculated with a thermal model, as in this work. With the temperature estimated through equation (2) and the density obtained via equation (1), the instantaneous pressure is calculated using an equation of state. The equation of state and all other thermodynamic properties are obtained using the Refprop library [9].

The third group of equations represents the valve dynamics. A one-degree of freedom mass-spring model is used, with effective force areas being used to characterize the force acting on the valve [10]. Finally, the fourth group of equations predicts the mass flow rates through the valves and the gas leakage through the piston-cylinder gap. Theoretical estimates of mass flow rate in valves are obtained from the isentropic flow through a convergent nozzle and then corrected by using the concept of effective flow area to account for viscous effects. Leakage through the piston-cylinder gap was modelled as a fully developed laminar Couette-Poiseuille flow [11].

Predictions from the compression cycle model that are used as input data in the thermal model are given in Table 1. The amount of heat generated due to friction in the piston-cylinder gap, $Q_{fric}$, is calculated with the Couette-Poiseuille velocity profile, whereas the electrical efficiency, $\eta_{elec}$, is obtained experimentally.
Table 1. Input data from compression cycle model to thermal model.

| Variable | Description | Calculation |
|----------|-------------|-------------|
| \( \dot{m} \) | Mass flow rate | \( f \int \dot{m}_{air} dt \) |
| \( \dot{Q}_W \) | Heat transfer rate inside cylinder | \( \dot{Q}_{pc} + A_w f \int H_G (T_W - T_G) dt \) |
| \( W_{ind} \) | Indicated power | \( f \int p_G dV_G \) |
| \( \dot{Q}_{bea} \) | Mechanical losses | Obtained from experimental data |
| \( \dot{Q}_{mot} \) | Electrical losses | \( (W_{ind} + \dot{Q}_{bea} + \dot{Q}_{pc})(1/\eta_{ele} - 1) \) |

3. Thermal model

Figure 1 presents a schematic view of the compressor simulated in this work, identifying the solution domains of each model employed. The hybrid thermal model solves the three-dimensional heat conduction in solid components with a differential formulation and the control volumes with gas through a lumped modelling approach.

3.1. Solid components

The differential model is solved, with suitable boundary conditions, through a commercial code [12] that uses the finite volume method. On the external surface of the compressor shell, convective heat transfer is established with the ambient. For the internal solid components in contact with refrigerant fluid (motor, cylinder, crankcase, etc.) a convective boundary condition was also specified:

\[
H_{ch} (T_{sol} - T_{ch}) = -k \frac{dT}{dn}, \tag{3}
\]

where \( T_{sol} \) is the temperature of a generic solid surface, \( T_{ch} \) is a temperature of a generic gas control volume and \( H_{ch} \) is the convective heat transfer coefficient. There are also regions inside the
compressor where internal heat generation must be specified. For instance, heat generation due to electrical motor inefficiencies is specified in the finite volumes that discretize the electrical motor, i.e.:

\[
\dot{Q}^{\text{mot}} = \dot{Q}_{\text{mot}}^{\text{vol}}.
\]  

(4)

Heat generation due to mechanical losses is characterized as a source term in a virtually defined thin wall between the shaft and the bearing [12]. Therefore, similarly to the heat generation in the motor:

\[
\dot{Q}^{\text{bea}} = \dot{Q}_{\text{bea}}^{\text{vol}}.
\]  

(5)

Finally, the total heat transfer at the cylinder wall is also used as boundary condition in the differential model:

\[
\frac{\dot{Q}_W}{A_W} = -k \frac{dT}{dn}.
\]  

(6)

3.2. Gas control volumes

The solution domain of gas inside the compressor is divided into several control volumes (internal environment, suction muffler, discharge chamber, discharge muffler and discharge line) and the temperature in each one of them is calculated using lumped energy balances. The transient term on the energy balances was only employed in internal environment control volume and the time step depends on the condition (on/off). Therefore, the implementation of the models for the on and off periods of the compressor will be addressed separately.

3.2.1. Off period. During the period in which the compressor is off, the compression cycle is not simulated. Furthermore, it is not necessary to simulate all the gas control volumes of the compressor immediately after the shutdown, as the higher heat capacity of the solid components forces thermal equilibrium between the solid walls and the adjacent gas in almost all the gas control volumes. The exception is the internal environment (ie) and, for this reason, only this control volume was simulated.

Immediately after the compressor is turned off, pressure in all control volumes defined for the gas domain is considered equal to the equalization pressure. The energy balance for the compressor internal environment is given by:

\[
H_{ch} (T_{\text{sol}} - T_{\text{ie}}) \Delta t = m c_p(T_{ie}^{t+\Delta t} - T_{ie}^{t}).
\]  

(7)

During the off period, heat transfer is mainly caused by natural convection, with the convective heat transfer coefficients being estimated using a modified version of the Catton equation for a rectangular cavity [13]:

\[
H_{ch} = C \left( \frac{Pr}{0.2 + Pr \alpha \nu} \right)^{0.28} k \left[ T_{\text{sol}} - T_{ie} \right]^{0.28},
\]  

(8)

where \( C \) is an experimentally calibrated coefficient required to adjust the Catton equation to the complex geometries of the compressor components. Four heat transfer coefficients are necessary to
represent the convective heat transfer in the compressor internal environment, and a different constant $C$ is required for each one. A new temperature for the internal environment chamber is calculated using equation (7).

### 3.2.2. On period

The simulation of the compression cycle is carried out when the compressor is switched on. The transient term in the energy balances on the control volumes is only important in the equation for the internal environment, as the thermal capacities of the remaining volumes are negligible. This energy balance remains in the same form shown in equation (7). The heat transfer coefficients for the internal environment are calculated using a modified version of the correlation for turbulent heat transfer over flat plates:

$$H_{ch} = C \left( \frac{\dot{m}_{oil}^{0.8}}{Pr^{0.33} \mu^{0.8}} \right),$$  

where the mass flow rate, $\dot{m}_{oil}$, is calculated using the procedure presented by Drost [14]. The other control volumes for the gas (suction and discharge paths) are simulated using steady state energy balances given by

$$H_{ch}(T_{sol} - T_{ch}) = \dot{m}(h_{out} - h_{in}),$$

where the mass flow rate, $\dot{m}$, is calculated in the compression cycle model. In this case, the convective heat transfer coefficients were estimated using modified equations for turbulent flow inside ducts, given by:

$$H_{ch} = C \left( \frac{\dot{m}^{0.8}}{Pr^{0.33} \mu^{0.8}} \right),$$

where $C$ is a constant that is experimentally adjusted, since it is affected by the compressor geometry. In total, there are eight heat transfer coefficients to be determined for the on period.

### 3.3. Coupled solution

The thermal model is simulated in a coupled manner with the compression cycle model explained in Section 2. The coupled solution is represented in the flowchart of Figure 2, where the inputs and results of each sub model can be observed. The inputs in dark grey are given by experimental data, while the ones in light grey are updated by the different models. Every time the compression cycle model is called, five cycles are simulated in order to achieve a fully established cyclic condition for the input conditions. After receiving data from the compression cycle model, the thermal model runs a certain number of time steps while maintaining constant the input data received from the compression cycle model. During these iterations, the differential model and the lumped models for the control volumes exchange information through heat transfer between components. After the specified number of iterations, the thermal model provides the temperatures to be used as input in the simulation of the compression cycle. The procedure is repeated along the time defined for the on period.

### 4. Results

As mentioned in the previous section, it is necessary to adjust the constants $C$ for the heat transfer coefficients of the thermal model based experimental data. To this extent, a R134a compressor with a displacement volume of 7.55 cm$^3$ was tested in a hot cycle test bench, being submitted to the operating condition characterized by $T_{evap} = -23.3^\circ C$ ($p_{evap} = 0.115$ MPa) and $T_{cond} = 40.5^\circ C$ ($p_{cond} = 1.03$ MPa). Both the gas inlet temperature and the external environment temperature were set to $32^\circ C$. The compressor was instrumented with thermocouples in locations in the gas and solid: (a) gas: suction line, suction muffler inlet, suction muffler outlet (suction chamber), discharge chamber, discharge muffler, discharge line, internal environment; (b) solid: cylinder, shell and motor.
The temperature distribution obtained after the compressor reached thermal equilibrium was used to adjust the constants C in the heat transfer correlations for the on period. For the suction region, the heat transfer coefficients were within 30 and 60 W/m²K during the on period. In the discharge region, the values ranged from 100 to 250 W/m²K. For the off period, an experimental temperature distribution obtained 25 minutes after the compressor was turned off was adopted to adjust the constants C. For the off period the heat transfer coefficients are lower since heat transfer occurs by natural convection.

The mesh used to discretize the solid components was generated for the numerical model after introducing simplifications in the original geometry. In regions of the fluid flow where heat flux and exchange of momentum are intense, special care was taken to ensure that the gradients were correctly represented. The mesh generated had a total of 1.3 million volumes to guarantee the truncation errors were negligible. Due to the very different heat capacities of the gas and solid parts, the time step in the solution of the problem is limited by the thermal transient of the gas inside the chambers. The values used for the time step were $\Delta t = 7s$ for the off period and $\Delta t = 2s$ for the on period.

![Flowchart of coupled simulation.](image-url)
For the off period, the simulation model was validated by comparing numerical and experimental results for temperature distribution of the compressor, during the 210 minutes after the compressor was shut down from a thermally stabilized condition. Figures 3 and 4 show very good agreement between numerical results and experimental data. During the off period, there is no gas compression, heat generation in the motor and bearings. Hence, the compressor average temperature is reduced due to the heat transfer to the external environment. As can be observed, the decay of temperature is similar for the solid and gas regions. This indicates that the temperature distribution during the off period is governed mainly by the heat capacity of the compressor solid parts.

**Figure 3.** Numerical vs. experimental results of $T_w$ and $T_{mot}$ for the off period.

**Figure 4.** Numerical vs. experimental results of $T_{sc}$ and $T_{ie}$ for the off period.

Figures 5 and 6 show the comparison between numerical and experimental results for a period after the compressor is switched on. The agreement is still good, despite the presence of some deviations in the first 90 minutes of operation. In fact, better agreement is observed as the thermal equilibrium condition is reached ($t > 180$ min), because the constants $C$ for the heat transfer coefficients were adjusted using the temperature distribution for this condition.

**Figure 5.** Numerical vs. experimental results for $T_{w}$ and $T_{mot}$ for the on period.

**Figure 6.** Numerical vs. experimental results for $T_{ie}$ and $T_{sc}$ for the on period.
After being validated, the model can be used to evaluate the compressor temperature distribution for cycling operating conditions. One test was defined by maintaining the compressor on for 30 minutes and then off during the next 30 minutes. Results for temperature variation at the cylinder wall and electrical motor are shown in Figures 7 to 10. Comparing Figures 7 and 8, it can be seen that the variation of the cylinder temperature is greater than that of the motor during the on-off cycling. Figure 9 shows an interesting behaviour for the suction chamber temperature. While during the on condition $T_{sc}$ is lower than the $T_{ie}$ (Figure 10), immediately after the compressor is turned off the temperature of the suction chamber rises due to heat transfer with discharge chamber and cylinder.

Figure 11 illustrates the temperature field in the solid components during one cycle of the on-off transient operating condition. Figure 11 (a) shows the temperature of the solid parts immediately after the compressor is turned on, i.e., after 30 minutes of off period. Figures 11 (a) to 11 (d) show the increase in compressor temperature that occurs during the on period. This aspect is more pronounced for solid components near the discharge system. The highest temperatures can be seen in Figure 11 (d), which is the moment when the compressor is switched off. The cooling down process after the compressor is switched off is illustrated by Figures 11 (e) and (f).

![Figure 7. Cycling results for $T_w$.](image1)

![Figure 8. Cycling results for $T_{mot}$.](image2)

![Figure 9. Cycling results for $T_{sc}$.](image3)

![Figure 10. Cycling results for $T_{ie}$.](image4)
5. Conclusions
This paper presented a thermal model developed to predict the temperature distribution of a small reciprocating compressor under on-off cycling condition. The simulation approach is based on a coupled solution procedure of three models: (i) a transient lumped model for the compression cycle, (ii) lumped energy balances for the remaining gas inside the compressor and (iii) finite volume model to solve heat conduction in the solid components. Some correlations of heat transfer coefficients were adjusted by using experimental data of temperature distribution. The model was validated through comparisons with temperature measurements in a period after the compressor was shut down (off period) and in a period after the compressor was switched on (on period). Good agreement was verified in both conditions, especially in the period when the compressor was off. The model was also applied in cycling operating conditions; again showing good agreement with measurements.

Acknowledgments
The present study was developed as part of a technical-scientific cooperation program between the Federal University of Santa Catarina and EMBRACO. The authors also acknowledge the support provided by EMBRACO and CNPq (Brazilian Research Council) through Grant No. 573581/2008-8 (National Institute of Science and Technology in Refrigeration and Thermophysics) and CAPES (Coordination for the Improvement of High Level Personnel).
| General and Greek symbols |
|---------------------------|
| A  | Heat transfer area  | (m²) |
| C  | Constant           | (-)  |
| c  | Specific heat      | (J/kg.K) |
| f  | Frequency of operation | (Hz) |
| h  | Enthalpy           | (J/kg) |
| H  | Heat transfer coefficient | (W/m²K) |
| k  | Thermal conductivity | (W/m.K) |
| m  | Mass               | (kg) |
| ṁ | Mass flow rate     | (kg/s) |
| N  | Time step number   | (-)  |
| \( \alpha \) | Thermal diffusivity | (m²/s) |
| \( \beta \) | Coefficient of thermal expansion | (1/K) |
| \( \gamma \) | Specific volume    | (m³/kg) |
| \( \mu \) | Dynamic viscosity  | (Pa.s) |

| Subscripts |
|------------|
| amb | External ambient |
| b   | Backflow |
| bea | Bearings |
| ch  | General gas chamber |
| dis | Discharge |
| dc  | Discharge chamber |
| dm  | Discharge muffler |
| dl  | Discharge line |
| ele | Electrical |
| G   | Gas inside cylinder |
| ie  | Internal environment |
| in  | Inlet |
| leak | Leakage |
| mot | Motor |
| oil | Oil |
| out | Outlet |
| pc  | Piston cylinder gap |
| sol | General solid surface |
| sc  | Suction chamber |
| sm  | Suction muffler |
| suc | Suction |
| w   | Cylinder wall |

**References**

[1] Hermes C J L and Melo C 2008 A first-principles simulation model for the start-up and cycling transients of household refrigerators *Int. J. Refrig.* **31** 1341-57

[2] Dutra T and Deschamps C J 2013 Experimental characterization of heat transfer in the components of a small hermetic reciprocating compressor *Appl. Therm. Eng.* **58** 499-510

[3] Ooi K T 2003 Heat transfer study of a hermetic refrigeration compressor *Appl. Therm. Eng.* **23** 1931-1945

[4] Ribas Jr F A, Deschamps C J, Fagotti F, Morriesen A and Dutra T 2008 Thermal analysis in reciprocating compressors – a critical review *Proc. Int. Compressor Eng. Conf. (Purdue)* Paper 1907

[5] Todescat M L, Fagotti F, Prata A T and Ferreira R T S 1992 Thermal energy analysis in reciprocating hermetic compressor *Proc. Int. Compressor Eng. Conf. (Purdue)* 1419-28

[6] Raja B, Seckar S J, Lal D M and Kalanidhi A 2003 A numerical model for thermal mapping in hermetically sealed reciprocating compressor *Int. J. Refrig.* **26** 652-58

[7] Ribas Jr F A 2007 Thermal analysis of reciprocating compressors *Proc. Int. Conf. on Compressors and their Syst. (London)* 277-87

[8] Annand W J D 1963 Heat transfer in the cylinders of reciprocating internal combustion engines *Proc. I. Mech. Eng.* **117** 973-96

[9] NIST 2007 *Refprop – Reference fluid thermodynamic and transport properties* (NIST)

[10] Deschamps C J, Prata A T and Ferreira R T S 2000 Modelling of Turbulent Flow Through Radial Diffusers *J. of Brazilian Society of Mechanical Sciences* **n 22** pp 31-41

[11] Ferreira R T S and Lilie D E B 1984 Evaluation of the Leakage Through the Clearance Between Piston and Cylinder in Hermetic Compressor *Proc. Int. Compressor Eng. Conf. (Purdue)* pp 1–6

[12] ANSYS Inc 2011 *User’s guide v.12.0.0.0* (ANSYS)

[13] Catton I 1978 Natural convection in enclosures *Proc. Int. Heat Transfer Conf. (Toronto)* **6** 13-31

[14] Drost R T and Quessada J F 1992 Analytical and experimental investigation of a scroll compressor lubricating system *Proc. Int. Compressor Eng. Conf. (Purdue)* 551-60