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ABSTRACT

In this paper the thermal energy analysis of a reciprocating hermetic compressor is performed using a computational program. The simulation model employed in the program is based on energy balances. For the refrigerant gas inside the compressor cylinder use was made of the first law of thermodynamics including time variations of the mass and energy fluxes. The required temperatures at the suction chamber, cylinder walls, discharge chamber, discharge muffler, compressor shell, and ambient inside the compressor shell are obtained from steady state energy balances at various locations within the compressor. Effective overall heat transfer coefficients were determined experimentally, except for the heat transfer between the refrigerant and the cylinder walls which was obtained from existing correlations. A companion simulation program which represents the compressor working features was used to calculate the mass fluxes at the suction, discharge, and the leakage flux. Simulation results are presented for a small compressor and compared with experimental results. Good agreement prevails indicating that the major effects affecting the thermal performance of the compressor have been considered by the proposed model.

INTRODUCTION

There are several reports in the literature concerning numerical models to predict the performance of hermetic refrigeration compressors. Those models require thermodynamic relationships to describe the behavior of the gas inside the cylinder. These relationships can be obtained either through a polytropic transformation or through an energy balance. A further aspect to be considered is the heat transfer to and from the refrigerant as it passes through the compressor. The models described in the literature differ mainly on how the two aforementioned issues are addressed. On the next paragraphs some of the major works done on compressor modelling will be reviewed.

Qvale *et al.* [1] indicated some areas where research should be done to improve the current knowledge on compressor modelling. Heat transfer on the suction and discharge lines, on the valves, and between the gas and the cylinder walls are the most important points mentioned by the authors. According to [1] the numerical models dealing with the cylinder of hermetic compressors have frequently employed the assumption of perfect gas in a polytropic process. The exponent of the polytropic equation is usually adjusted to fit experimental results. In this regard the polytropic index incorporates the combined effect of the heat transfer between gas and cylinder, friction, and deviations from the perfect gas behavior. Therefore, the influence of these effects separately cannot be detected.

Karll [2] investigated through the first law of thermodynamics the process described by a real gas inside the cylinder. He considered the cylinder as a close system undergoing exchange of heat and work with the surroundings. Prakash and Singh [3] also employed the first law of thermodynamics but assumed perfect gas behavior. The heat transfer between the cylinder walls and the gas was predicted using the correlation given by Adair *et al.* [4]. Röttger and Kruse [5] verified through their model that for the compressor performance it is important to use the equation of state for real gas but for the valve performance suffices to use the perfect gas assumption. Ng *et al.* [6] also employed the first law of thermodynamics to model the compressor. Additionally they assumed the cylinder to be adiabatic. According to [6] the real gas model is justified only if the gas superheating at the suction is small and if the pressure ratio at the compressor is large. A simplified model based on a polytropic process was proposed by Suefuji and Nakayama [7]. The disadvantage of their model is that the suction and discharge temperatures must be known a priori. Brok *et al.* [8] have shown that in reciprocating compressors there are two major heat transfer contributions that should be carefully examined: the part of the compression power that is transferred to the external environment and that transferred to the suction gas. In [8] the heat transfer between refrigerant and cylinder walls was modelled via Adair *et al.* [4] correlation, modified by a factor to reduce the mixing inside the cylinder. This factor was adjusted to fit the experimental data. For the suction and discharge values use was made of the Dittus-Boelter correlation for turbulent flow [9]. In the suction chamber a correlation for annular concentric ducts was adopted, and for the discharge chamber a constant value of the heat transfer coefficient was employed. According to [8] the influence of the heat transfer inside the cylinder on the compressor performance is less than what is suggested in the literature. For low speed situations it was found that during compression at most 4% is gained by cooling and 3% is lost by superheating. Parise and Cartwright [10] also performed a compressor simulation including an energy balance in the cylinder. Heat transfer between refrigerant and cylinder walls was modelled using Adair *et al.* [4] correlation.

Lee *et al.* [11] proposed a simulation model emphasizing the need for a correct evaluation of the thermodynamic properties inside the cylinder. They employed both the first law and a polytropic process. According to [11] the pressure inside the cylinder is not strongly affected by different values of the polytropic exponent but the temperature showed a great dependance on the value adopted. The mass flow rate through the compressor has shown to be weakly dependant on the thermodynamic model employed. Liu and Zhow [12] obtained experimentally the temperature distribution on the cylinder lateral walls of a reciprocating compressor (averaged during a cycle) for different pressure ratios, suction temperatures and rotational speeds. They have observed that the temperature variation along the axial direction is due to heat transfer to the suction gas. The temperature variation along the circumferential direction is due to the friction between the piston and the cylinder wall and also due to the discharge location. Schary *et al.* [13] performed a compressor simulation incorporating in their model gas pulsation and a polytropic process. The numerical results were checked against experimental results an a surprisingly good agreement was achieved.

Some few works reported more elaborate models where the heat transfer inside the cylinder is evaluated through the differential form of the energy equation. Chong and Watson [14] solved the continuity, momentum and energy equations in differential form using a finite difference scheme. Their analysis assumed laminar flow and ignored the gas suction and discharge through the valves. They observed that as the rotational speed increases, the thermal boundary layer becomes so thin that heat transfer can be predicted through a conductive one-dimensional model which requires only the boundary layer thickness. More recently, Recktenwald [15] (also Recktenwald *et al.* [16]) employed two models to investigate the instantaneous heat transfer between the cylinder walls and the gas in a reciprocating compressor. One model used mass and energy balances to predict the instantaneous bulk thermodynamic properties of the gas in the cylinder. Heat transfer between the cylinder walls and the gas was calculated with Adair*et al.* [4] correlation. The other model solved the unsteady continuity, momentum, and energy equations for the gas in the cylinder using a finite volume technique. No heat transfer correlation was needed in this model. The instantaneous heat transfer predicted by the simple model using Adair *et al.* [4] correlation was an order of magnitude less than that predicted by the finite volume model indicating that instantaneous heat transfer in compressor cylinders is much greater than many researchers have assumed. The average heat transfer coefficient during a cycle calculated in [15] is at least twice bigger than the value that is predicted from Adair *et al.* [4].

The main goal of the present work is to present a model based on the first law of thermodynamics for simulating the performance of a hermetic refrigeration compressor, and in this regard shed more light on this subject. In what follows the energy balance for the refrigerant inside the cylinder is computed for each instant during the operating cycle and take into account time variations of mass and energy fluxes. For this balance the temperatures at the discharge, the discharge chamber, the suction line, the gas inside the compressor shell, the compressor shell and the cylinder are determined from steady state energy balances. Those balances require heat transfer coefficients which are obtained either from existing correlations or from experimental measurements. Similar procedure has recently been adopted by Meyer and Doyle [17-18] to develop an analytical model for the heat transfer to the suction gas in a low-side hermetic refrigeration compressor.

ENERGY BALANCES

The energy balances to be explored in this section will be developed with the aid of Fig. 1. This figure presents a schematic view of a hermetic refrigeration compressor illustrating the motor, the cylinder and piston, the discharge and suction valves, the discharge and suction chambers, as well as the compressor shell. Also shown in Fig. 1 by dashed lines are the selected control volumes chosen for the energy balances.

Energy Balance for the Gas inside the Cylinder

For the gas inside the cylinder an energy balance is computed at each instant during the operating cycle. According to the first law of thermodynamics this balance can be written as

$$Q_c + W_i = \frac{\partial}{\partial t}\int_{c.v} \rho e d\forall + \int_{c.s.} \rho e \vec{V}.d\vec{A} \tag{1}$$

where e is the specific energy of the gas, $\forall$ is the cylinder volume and $\vec{V}$ is velocity; Q_c is the rate of heat transferred between the gas and the cylinder walls (including piston and valve plate), W_i is the power input. The two terms on the right-hand side of eq. (1) are, respectively, the variation with time of the energy inside the cylinder and the flux of energy associated with the mass fluxes through the suction and discharge valves and with the leakage mass flow through the clearance between piston and cylinder wall. From thermodynamics considerations eq. (1) can be written as (see Todescat *et al.* [19]),

$$\frac{dT}{dt} = \frac{1}{mc_v}\left\{Q_c + \dot{m}_s(h_{sc} - h) - \frac{T}{\rho}\frac{\partial p}{\partial T}\bigg|_\rho \left[\rho\frac{d\forall}{dt} - (\dot{m}_s - \dot{m}_d - \dot{m}_v)\right]\right\} \tag{2}$$

At each instant during the compressor cycle the temperature of the gas inside the cylinder is determined from eq. (2). The heat transfer rate, Q_c, is obtained using an effective heat transfer coefficient as,

$$Q_c = h_c\, A_c\, (T_c - T) \tag{3}$$

where A_c is the instantaneous cylinder surface area and T_c is an effective surface temperature. In the present analysis h_c is evaluated according to three different expressions. That of Adair *et al.* [4], that of Annand [4], and a third expression obtained simply multiplying Annand's formula by a factor of three. This last correlation was used to explore the influence of higher heat transfer coefficients on the compressor performance according to what is recommended by Recktenwald [15].

All the thermodynamics properties in eq. (2), including c_v, are evaluated according to Martin-Hou equation of state for real gas [20]. The mass fluxes at the suction, discharge, and the leakage flux, $\dot{m}_s$, $\dot{m}_d$ and $\dot{m}_v$, are obtained from a simulation program which represents the working features of the compressor. This program is similar to those developed by Ussyk [21] and Ferreira [22]. The specific enthalpy of the suction gas, h_{sc}, is a function of the specified suction pressure and the unknown temperature, T_{sc}. The temperature T_{sc} as well as the other temperatures inside the compressor are obtained through specific energy balances as explored next.

Energy Balance at the Muffler and Suction Chamber

According to Fig. 1, an energy balance at the muffler and suction chamber requires that

$$Q_s = \dot{m}_s\, h_{sc} - \dot{m}\, h_s - \dot{m}_v\, h_{sc} - \dot{m}_{bs}\, h_c \tag{4}$$

where Q_s is the heat transfer between the gas inside the chamber and that inside the compressor shell. In terms of an overall energy balance Q_s can be written as,

$$Q_s = \overline{UA}_s\, (T_{sc} - T_{ms}) \tag{5}$$

in which $T_{ms} = (T_s + T_{sc})/2$; T_s is the temperature of the intake gas and T_{sc} is the temperature of the gas exiting the chamber and entering the cylinder. The mass fluxes with their respective enthalpies appearing in eq. (4) are indicated in Fig. 1. As the leakage mass flux $\dot{m}_v$, the backflow at the suction valve, $\dot{m}_{bs}$, and the mass flow rate produced by the compressor, $\dot{m}$, are obtained by the simulation program that represents the working features of the compressor as mentioned previously.

Energy Balance at the Cylinder

An overall energy balance at the cylinder along the entire operating cycle requires that,

$$Q_c = \dot{m}_s\, h_{sc} - \dot{m}_{bs}\, h_c + \dot{m}_{bd}\, h_{dc} - \dot{m}_d\, h_c + W_i + W_p - \dot{m}_v\, h_c \tag{6}$$

where Q_c is given by eq. (3).

It should be noted that eq. (1) was written for the gas inside the cylinder whereas eq. (6) represents an energy balance for a control volume that envelops the cylinder itself, including the piston. In this regard, the frictional power, W_p, was taken account in eq. (6). To calculate the mechanical losses due to friction, W_p, variation of the oil viscosity with temperature was considered.

Energy Balance at the Discharge Chamber

The energy balance performed at the suction line according to equation (4) included both the muffler and the suction chamber. For the discharge line the muffler and the discharge chamber will be considered separately. According to Fig. 1 the energy balance at the discharge chamber can be written as,

$$Q_{dc} = \dot{m}_d\, h_c - \dot{m}\, h_{dc} - \dot{m}_{bd}\, h_{dc} \tag{7}$$

where the heat transfer between the gas inside the discharge chamber and that inside the compressor shell, Q_{dc}, is expressed as,

$$Q_{dc} = \overline{UA}_{dc}\,(T_{dc} - T_{ie}) \tag{8}$$

The backflow at the discharge valve, $\dot{m}_{bd}$, is obtained via the compressor simulation program as discussed earlier.

Energy Balance at the Discharge Muffler

According to Fig. 1, an energy balance for the discharge muffler can be written as,

$$Q_d = \dot{m}(h_{dc} - h_d) \tag{9}$$

where,

$$Q_d = \overline{UA}_d\,(T_{m,d} - T_{ie}) \tag{10}$$

in which $T_{m,d} = (T_d + T_{dc})/2$; T_d is the temperature of the gas leaving the compressor and T_{dc} is the gas temperature at inlet of the discharge muffler.

Overall Energy Balance

The compressor exchanges heat with the surroundings according to its shell temperature. To obtain this temperature the following overall energy balance can be performed,

$$Q_e = \dot{m}(h_s - h_d) + W_e \tag{11}$$

where,

$$Q_e = \overline{UA}_e\,(T_b - T_{ee}) \tag{12}$$

The shell temperature is related to the gas temperature inside the compressor, T_{ie}, by simply equating Q_i and Q_e as shown in Fig. 1, that is,

$$Q_i = \overline{UA}_i\,(T_{ie} - T_b) = \overline{UA}_e\,(T_b - T_{ee}) = Q_e \tag{13}$$

Equations (4), (6), (7), (9), (11) and (13) are used to determined T_{sc}, T_c, T_{dc}, T_d, T_b and T_{ie}. Those equations are nonlinear due to the dependance of the specific enthalpy on the temperatures, and are solved simultaneously and iteratively. The required mass fluxes are obtained through the compressor simulation program. To evaluate the heat fluxes according to eqs. (5), (8), (10), (12) and (13), overall heat transfer coefficients are required. The precise determination

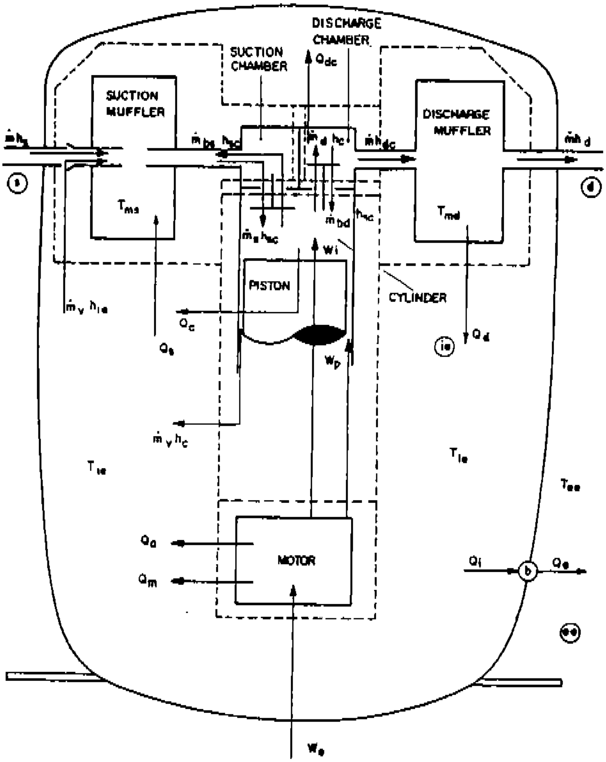

Fig. 1 -Schematic view of a hermetic refrigeration compressor ilustrating the control volumes chosen for the energy balances

of those heat transfer coefficients is a difficult task. This is so due to the complicated geometry of the parts involved as well as due to the complexity associated with the heat transfer mechanisms. Inside the suction and discharge lines, for instance, the flow is pulsated in addition to be turbulent and three-dimensional. Also, the prediction of the heat transfer between the gas inside the compressor and the compressor shell have to be made considering the presence of the lubricating oil. At present it seems to be no detailed report in the literature concerning the aforementioned difficulties. A practice that has currently been adopted is to rely on experiments in order to estimate the heat fluxes. This same procedure was used in the present work. Through temperature measurements, and knowing the compressor operating conditions, the overall heat transfer coefficients required in the present model were determined. The compressor investigated here is a small reciprocating hermetic compressor (about 1/4 horsepower) and typical values of the overall heat transfer coefficients are shown in Tab. 1.

Table 1 - Typical values of the overall heat transfer coefficients determined from experimental data [W/K]

$\overline{UA}_s$	$\overline{UA}_{dc}$	$\overline{UA}_d$	$\overline{UA}_i$	$\overline{UA}_e$
0.76	0.57	2.3	9.2	4.5

At this point the model used to simulate the thermal performance of the compressor has been completed. The temperature of the gas inside the cylinder is determined at each instant during the operating cycle using eq. (2) which was solved by a Runge-Kutta algorithm. The temperatures at the suction chamber, cylinder walls, discharge chamber, discharge muffler, compressor shell, and ambient inside compressor shell, are averaged over the cycle and are determined from eqs. (4), (6), (7), (9), (11) and (13) via Newton-Raphson. A companion program was used to evaluate the requiring friction losses and the mass fluxes. Except for the heat transfer coefficient between the gas and the cylinder walls which was obtained from correlations available in the literature, the heat transfer coefficients were determined using experimental data.

RESULTS AND DISCUSSIONS

The model presented in the previous section was used to simulate the thermal performance of a small reciprocating hermetic compressor (about 1/4 HP) subjected to the following operating conditions,

evaporating temperature	=	$-23.3^0 C$
condensing temperature	=	$54.4^0 C$
superheating temperature	=	$32.2^0 C$
subcooling temperature	=	$32.2^0 C$
ambient temperature	=	$32.2^0 C$
compressor inlet temperature	=	$32.2^0 C$

Table 2 explores the influence of different correlations for the heat transfer coefficient between the gas and the cylinder walls on the compressor performance. Except for the temperature of the cylinder walls, T_c, use of different correlations for the heat transfer coefficient h_c has little effect on the quantities presented in Tab. 2. Even when h_c was multiplied by a factor of three the quantities presented in Tab. 2 seem to show little reaction to that. From Tab. 2 it is also seen that the model presented here is able to predict, with good accuracy, the experimental results. Annand's correlation multiplied by three seems to yield results that are slightly better than those obtained using the two other correlations. Because of that, in what follows, use will be made of this correlation. For other conditions (not shown in Tab. 2) Annand's correlation times three also presented a better agreement with the experimental results.

Table 2 - Influence of different correlations for the cylinder heat transfer coefficient on the compressor performance

Temperatures [°C]	h_c			Experiment
	Adair	Annand	Annand (x3)	
T_{sc}	55.3	55.2	56.0	56.2
T_c	85.5	93.3	98.4	101.0
T_{dc}	138.8	138.1	138.7	140.8
T_d	88.0	87.2	86.3	88.1
T_b	64.4	64.0	64.2	65.0
T_{ie}	80.1	79.5	79.7	80.9
Power Consumption [W]	191.1	187.8	185.6	192.5
Mass Flow Rate [kg/s]	1.949×10^{-3}	1.930×10^{-3}	1.856×10^{-3}	1.865×10^{-3}
EER [Btu/W.h]	5.02	5.06	4.92	4.84

The small influence of h_c on the thermal performance of the compressor is demonstrated in Fig. 2. This figure shows the various heat transfer contributions plotted as a function of the temperature of the compressor shell, T_b. Experimentally, T_b can be varied by controlling the airflow over the compressor shell.

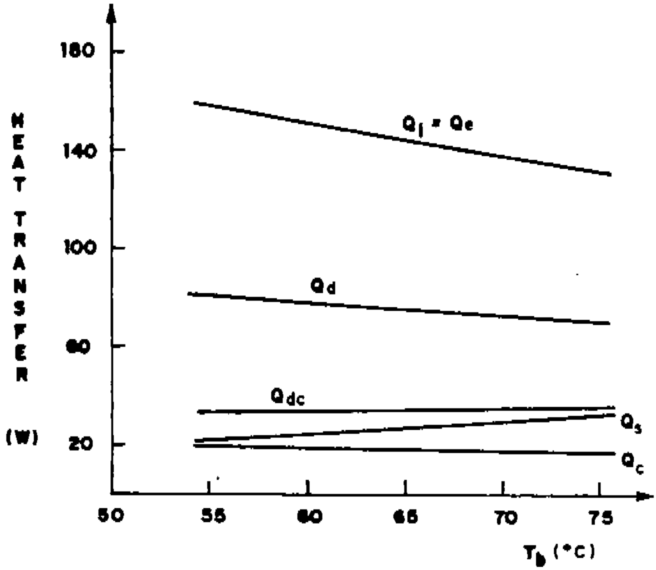

Fig. 2 - Heat transfer contributions as a function of the temperature of the compressor shell, T_b

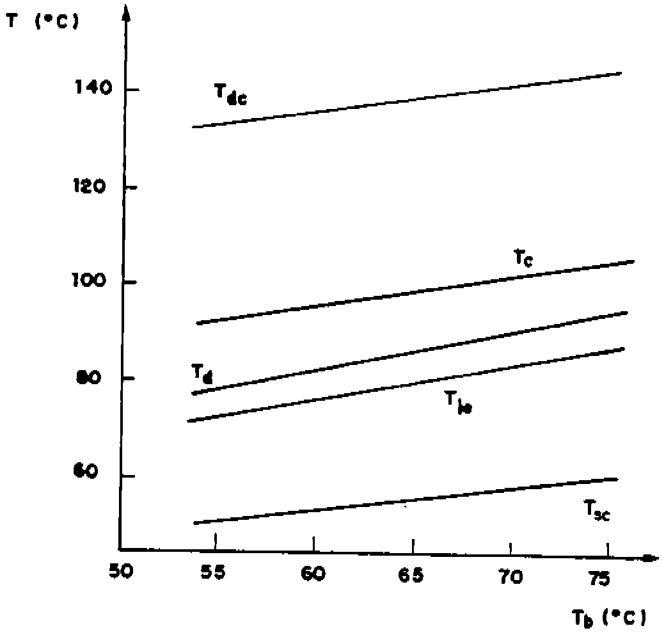

Fig. 3 - Calculated temperatures as a function of the temperature of the compressor shell, T_b

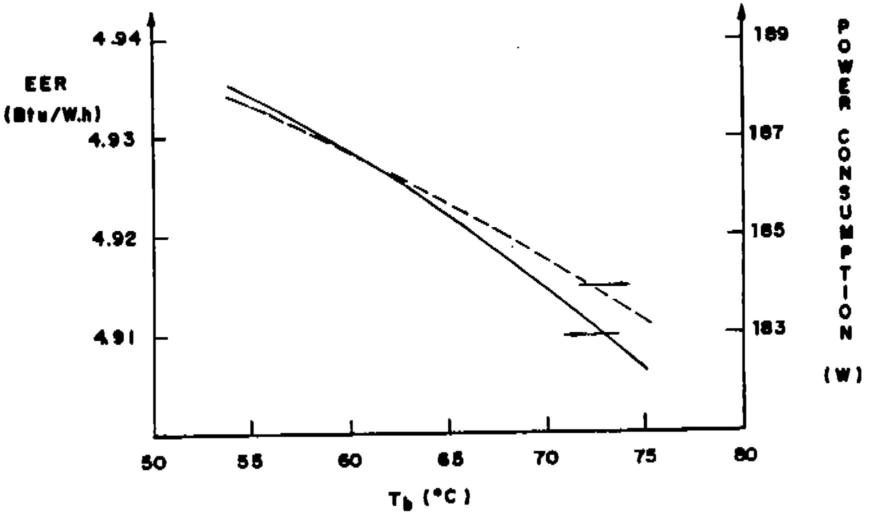

Fig. 4 - Energy efficiency ratio and power consumption as a function of the temperature of the compressor shell, T_b

From Fig. 2 it is seen that Q_c represents the least heat transfer contribution among those entering in the energy balances. Therefore, variations on Q_c are expected to have little effect on the thermal performance of the compressor. It is also seen from Fig. 2 the important role played by the heat transfer Q_e on the overall energy balance. On a relative basis, the heat transfer to the refrigerant at the muffler and suction chamber, Q , is the contribution that is more affected by changes on the temperature of the compressor shell. As T_b varies from 55 to $75^0 C$, Q_s increases by about 50%.

The influence of the temperature of the compressor shell on the other temperatures is shown in Fig. 3. As expected, all temperatures increase with increasing values of T_b. The increase of the refrigerant temperature at the suction line, $T_{s,c}$, represents a waste of the cooling capacity which reduces the cycle efficiency. As T_b varies from 50 to $75^0 C$, $T_{s,c}$ increases around $10^0 C$.

Results for the energy efficiency ratio, EER, and for the compressor power consumption are explored in Fig. 4. A comparison of the power consumption and the heat transfer to the surroundings, Q_e, indicates that between 70 and 80% of the power consumption is wasted as heat to the external environment. This result seems to reinforce Brok *et al.* [8] reccommendation that heat losses to surroundings should be carefully examined.

CONCLUSIONS

The present paper presents a model to perform thermal energy analysis of reciprocating hermetic compressors. The gas inside the cylinder was modelled through the first law of thermodynamics taking into account time variations of mass and energy fluxes. Steady state energy balances were used to obtain the temperatures of the compressor components and of the refrigerant at various locations along the flow path. Those balances require heat transfer coefficients which were determined either from existing correlations or from experimental measurements.

This model was used to simulate the thermal performance of a small compressor (about 1/4 horsepower). Comparisons between experimental results and those obtained with the simulation showed good agreement, indicating that the model can be successfully employed for design studies.

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