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THERMAL ANALYSIS OF A HERMETIC RECIPROCATING COMPRESSOR

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ABSTRACT

The development of more efficient hermetic compressors for refrigeration involves a sound analysis of the heat exchanges inside the machine to evaluate the interaction among compressor structure, flow characteristics, refrigerant properties and operative conditions.

This work presents a computational procedure for the steady-state thermal analysis of a hermetic reciprocating compressor. The machine is subdivided into six parts (shell, compressor body, suction muffler, suction chamber, discharge chamber, discharge line). The energy balance based on the first law is established for each component and for the overall system to obtain the temperature distribution inside the machine and the heat flow rates exchanged.

The results of the simulation are compared against the experimental measurements carried out on commercial units operating with R600a and R134a.

INTRODUCTION

Several computational models for the thermal analysis of refrigeration compressors are to be found in the open literature: they range from simple simulations on the effects of heat transfer on compressor performances to more complex procedures concerning the overall system. Among the first type of analyses, the one by Brock et al. 1980 [1] is particularly interesting, considering the heat exchanges from the compression process to the external medium and to the suction gas (internal heat transmission). This analysis is carried out by applying both simple overall thermodynamic relations (isentropic and polytropic equations) and also by a simulation model complemented with heat transfer equation for the cylinder, the suction and discharge lines. This model is limited only to open-type compressors and, therefore, it does not include the heat exchanges relative to electric motor, compressor shell and lubricant oil, which are very important in hermetic units. The most recent and interesting numerical codes for the overall compressor are those by Meyer et al. 1990 [2] and by Todescat et al. 1992 [3] and 1994 [4]. The model developed by Meyer et al. for a small reciprocating hermetic compressor is based on a steady-state energy balance for the different components and for the overall system, where the heat transfer coefficients are derived from available correlations or from experimental measurements. The mass flow rate and the compression process is analysed by assuming experimental values for the volumetric efficiency and the compression efficiency respectively. This program gives the temperatures and the heat flow rates relative to each component and the refrigerant gas outlet conditions, but it requires the value of the electric power input, which is usually something one would like to determine. The model by Todescat et al. is also based on a steady-state energy balance of the different compressor components and it also includes a sound analysis of the heat and work transfer during the compression cycle. A companion simulation program is used to evaluate mass flow rates, enthalpies and pressures inside the machine. This approach allows a computation of the temperature distribution inside the compressor, the refrigerant outlet conditions and also the power input. The results obtained agree well with the experimental measurements on a small hermetic compressor.

Present work reports a new model for the thermal analysis of a hermetic reciprocating compressor which is compared against the experimental data measured both on a R600a and a R134a commercial units in a wide range of operative conditions.

THEORETICAL ANALYSIS AND SIMULATION MODEL

Governing Equations

An exhaustive analysis of a compressor requires a dynamic simulation of the refrigerant flow, particularly during the compression cycle, to evaluate the instantaneous heat and work transfer and pressure and mass flow fluctuations. This approach is very complicated and gives results strictly correlated to the specific geometry of the machine simulated. The intent of this work is, rather, to develop a simple model easily adaptable to different hermetic units. The basic assumption of the model is to consider the refrigerant flow through the compressor as a one-dimensional steady-state current. In this way, for each component and for the overall system, it is possible to establish a steady-state thermal balance to compute temperatures and the heat and work flow rates. The main irreversibilities inside the compressor (electric energy conversion losses, friction losses, internal heat exchanges) are taken directly into consideration, with the exception of pressure losses and flow rate leakages.

The compressor is subdivided into six parts: shell, body (cylinder, head and electric motor), suction muffler, suction chamber, discharge chamber, discharge line (see figure 1). The following governing equations are considered for each component of the compressor:

- Suction muffler. The suction muffler receives convective heat flow rate (Q_{mc}) from the recirculated gas in the shell and radiative heat flow rate (Q_{mr}) from the compressor body and rejects them to the suction refrigerant flow. Its energy balance, therefore, gives:

$$m_r (h_2 - h_1) = Q_{mc} + Q_{mr} \tag{1}$$

where m_r is the refrigerant mass flow rate, h is the specific refrigerant enthalpy, subscripts 1 and 2 are relative to muffler inlet and outlet respectively. The refrigerant gas flow rate at inlet of the muffler is a mixture of a fraction "x" of fresh refrigerant entering the hermetic unit and a fraction (1 - x) of gas recirculated in the shell. The value of the parameter "x" is derived from experimental observations. The convective heat transfer coefficient between recirculated gas and suction muffler is calculated by the Petukhov equation [5].

- Suction chamber. In the suction chamber the refrigerant flow receives convective heat flow rate (Q_{sc}) from the cylinder head:

$$m_r (h_3 - h_2) = Q_{sc} \tag{2}$$

where subscript 3 is for cylinder inlet. The heat transfer coefficient in the suction chamber is computed by a Dittus-Boelter [6] type equation corrected to account for the effect of pulsating flow upstream of the inlet valve.

- Compression cycle. In a small compressor during the compression cycle there is a relevant heat exchange between gas and cylinder; this process cannot, therefore, be considered adiabatic and can be simulated by a polytropic equation with an exponent n between k (adiabatic operation) and 1 (isotherm operation). The following equations are derived from this assumption to compute the operative conditions at outlet of the cylinder 4, the total electric power input W_{in} and the heat flow rate given to the cylinder Q_{cil}:

$$T_4 = T_3 (P_d/P_s)^{(n-1)/n} \tag{3}$$

$$W_{in} = m_r [n/(n-1)] R T_3 [(P_d/P_s)^{(n-1)/n} - 1] / (\eta_e \eta_m) \tag{4}$$

$$Q_{cil} = m_r \{[n/(n-1)] - [k/(k-1)]\} R T_3 [(P_d/P_s)^{(n-1)/n} - 1] \tag{5}$$

where T's are absolute temperatures, P_d and P_s are discharge and suction pressures, R is the gas constant and k the average isentropic index. The electrical and mechanical efficiencies η_e and η_m, which account for electric energy conversion losses and friction losses, and the value of the polytropic exponent n are derived from experimental measurements.

- Discharge chamber. The compressed gas flowing at high temperature through the discharge chamber heats the compressor body (Q_{dc}). The energy balance is:

$$m_r (h_4 - h_5) = Q_{dc} \tag{6}$$

where subscript 5 relates to the discharge chamber outlet. The heat transfer coefficient in the discharge chamber is computed by a Dittus-Boelter [6] type equation corrected to account for the effect of the pulsating flow downstream of the discharge valve.

- Discharge line. The compressed gas flowing towards the outlet of the hermetic compressor exchanges heat flow rate (Q_{dl}) with the recirculated gas inside the shell through the wall of the discharge tube. The energy balance for this heat transfer is:

$$m_r (h_5 - h_6) = Q_{dl} \tag{7}$$

where subscript 6 relates to the hermetic compressor outlet section. The heat transfer coefficients inside and outside the discharge tube are calculated by a Dittus-Bolter [6] type equation and by the Zhukauskas [7] equation respectively.

- Compressor shell. The shell receives convective heat flow rates from the recirculated gas (Q_{sg}) and from the lubricant oil (Q_{os}), and radiative heat flow rate (Q_{sr}) from the compressor body. It rejects this heat power to the surroundings both by natural convection (Q_{ac}) and radiation (Q_{ar}). This rejected heat flow rate has to be equal to the difference between the electric power input and the rate of enthalpy increase of the refrigerant across the hermetic unit. The relative energy balance is

$$Q_{sg} + Q_{os} + Q_{sr} = Q_{ac} + Q_{ar} = W_{in} - m_r (h_6 - h_0) \tag{8}$$

where 0 is the subscript referring to the hermetic unit inlet. The convective heat transfer coefficient between the shell and the environment air is calculated according to the traditional correlations for natural convection [8], while the convective heat transfer coefficient between the shell and the recirculated gas is computed by the Petukhov equation [5].

- Lubricant oil. The lubricant oil receives convective heat flow rate from the compressor body (Q_{oc}) and rejects it to the shell (Q_{os}). Its governing equation is:

$$Q_{oc} = Q_{os} \tag{9}$$

The heat transfer coefficients between oil and shell and oil and electric motor are determined by the classical natural convection heat transfer correlations [8] while the heat flow rate between oil and cylinder and cylinder head is derived from the lubricant flow rate and lubricant enthalpy increase.

- Recirculated gas. The recirculated gas inside the shell receives heat power from the compressor body (Q_{gc}) and the discharge line (Q_{dl}) and is cooled by the injection of fresh refrigerant gas and by heat transferred to the shell (Q_{sc}) and to the suction muffler (Q_{mc}). The energy balance equation is:

$$Q_{gc} + Q_{dl} = Q_{mc} + Q_{sc} + (1 - x)\, m_r (h_g - h_0) \tag{10}$$

where subscript g relates to the recirculated gas.

- Compressor body. The compressor body (cylinder, cylinder head and electric motor) is heated by the gas during the compression cycle (Q_{cil}) and in the discharge chamber (Q_{dc}) and by the electric conversion losses ($(1-\eta_e)W_{in}$) and the friction losses ($(1-\eta_m)\eta_e W_{in}$). It rejects this heat flow rate by radiative heat exchanges with shell (Q_{sr}) and muffler (Q_{mr}) and by convective heat exchanges with the refrigerant flow in the suction chamber (Q_{sc}), the recirculated gas (Q_{gc}) and lubricant oil (Q_{oc}). The relative energy balance equation is:

$$Q_{cil} + Q_{dc} + (1-\eta_e)\, W_{in} + (1-\eta_m)\, \eta_e\, W_{in} = Q_{sr} + Q_{mr} + Q_{sc} + Q_{gc} + Q_{oc} \tag{11}$$

The radiative heat flow rates in all the above components are calculated by the suitable equations for radiation [9].

The refrigerant mass flow rate through the compressor is calculated by the following relation:

$$m_r = \eta_v \, V \, \omega / v_3 \tag{12}$$

where V is the swept volume of the compressor, ω the electric motor rotation speed, v_3 the refrigerant specific volume at inlet of the cylinder and η_v the volumetric efficiency of the compressor.

Computer Program

The simulation computer code has an iterative structure. Input values are set for the boundary conditions (surroundings temperature, suction refrigerant temperature and pressure, discharge pressure) and the characteristic parameters of the hermetic unit (geometrical data, refrigerant, electric motor speed, electrical efficiency, mechanical efficiency, volumetric efficiency, polytropic index "n" of compression and fraction "x" of fresh gas directly entering the suction muffler). Guessed values are assumed for the refrigerant temperature at inlet of the cylinder, for the recirculated gas temperature and for the temperature of the cylinder. Then the refrigerant mass flow rate is calculated by eq. (12) and the thermal balance is established for each compressor component, according to eqs. (1 → 11). A specific subroutine has been developed for each component, while a linked computer program provides the properties of the operative fluid. Like so, the characteristic temperatures inside the compressor and along the refrigerant flow are computed together with the heat flow rates and power input. The guessed temperatures are compared against the calculated ones and further iterations are carried out till convergence is reached. The final output results include refrigerant mass flow rate, refrigerant outlet temperature, temperatures inside the hermetic unit, heat flow rates and electric power input.

RESULTS AND COMPARISON WITH EXPERIMENTATION

The experimental measurements have been carried out on a R600a reciprocating hermetic compressor with 8 cm^3 swept volume and on a R134a reciprocating hermetic compressor with 6 cm^3 swept volume. Each compressor has been inserted in a calorimetry rig for the measurement of the refrigerating capacity in accordance with ASHRAE Standard (temperature at outlet of the condenser and outlet of the evaporator 32°C, condensation temperature 55°C). During each test the following parameters are measured: electric power input, refrigerating capacity, evaporation and condensation pressures, gas temperature at inlet and at outlet of the compressor, refrigerant temperature at outlet of the condenser and at outlet of the evaporator. The compressor is also equipped with several copper-constantan thermocouples to measure the temperatures of its components and of the refrigerant in different positions. During each run the following data is collected: the temperature of the shell in different positions; the temperature of lubricant oil; the temperature of the recirculated gas; the temperature of the electric motor, of the cylinder and the head of the cylinder, the temperature of the refrigerant gas at inlet of the suction muffler, at inlet and at outlet of the cylinder.

Table 1 and 2 give the comparison between the experimental data measured and the values calculated by the simulation program under the same operative conditions both for R600a and R134a hermetic compressors. Three different operative conditions are considered in accordance with ASHRAE standard: evaporation temperatures -35°C, -23.3°C and -10 °C, respectively. The experimental data reported is the average values of the data collected in several repeated runs. As one can see there is a fair agreement between the calculated values and the experimental ones except for R134a unit at -35°C evaporation temperature. This operative condition is characterized by a high pressure ratio and a low refrigerant flow rate; there is a strong degradation of the electric, mechanical and volumetric performances of the machine, while some effects, not included in the present analysis (such as gas leakages), become relevant.

The results of the simulation include the information necessary to evaluate the potential performance improvement of the compressor, such as the heat flow rates regarding the different components. This data is not easy to determine by direct measurements, and therefore the code, as a means to generalize and extend the information of experimental tests, is a useful design tool.

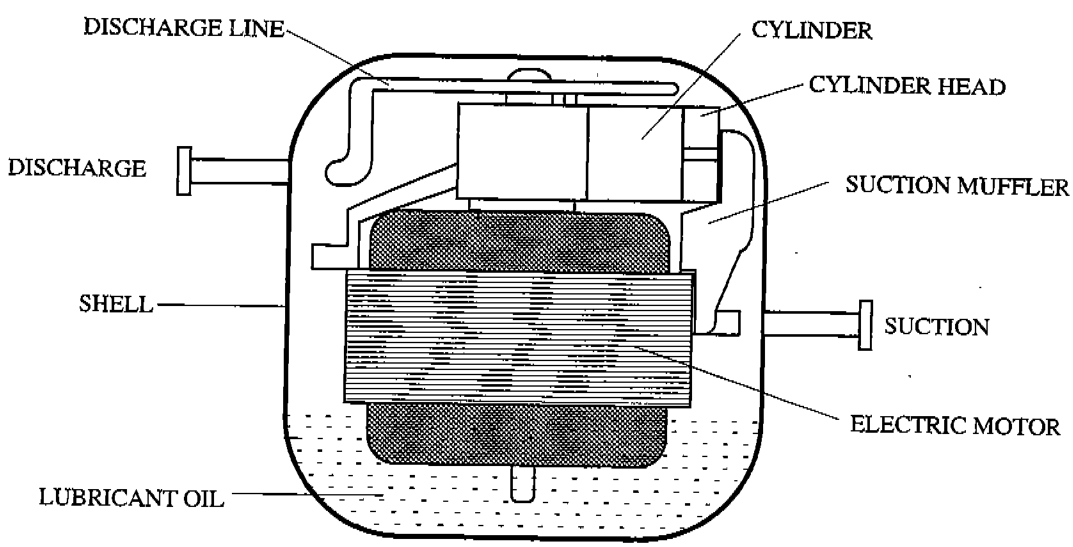

Figure 1. Schematic view of the compressor

CONCLUSIONS

A computer program for the thermal analysis of a small hermetic reciprocating compressor is presented. The results of the simulations are in fair agreement with the experimental data measured both on a R600a and a R-134a hermetic units. The computer code is able to determine the temperature levels inside the machine, the power input and the heat flow rates exchanged. It can also be easily adapted to different operative fluids and different compressor geometries, and therefore it can be a useful tool for design and development purposes. Future development of this program will involve the coupling with an unsteady-state analysis of the compression cycle, so as to eliminate the recourse to the compression quasi-static polytropic equation.

Table 1. Comparison between experimental and calculated parameters relative to a R600a hermetic compressor.

Input Data	T_{evap} = -35°C		T_{evap} = -23.3°C		T_{evap} = -10°C	
Surroundings temperature (°C)	32.0		32.0		32.0	
Compressor inlet temperature (°C)	38.0		36.1		33.6	
Compressor suction pressure (kPa)	36.5		63.1		108.7	
Compressor discharge pressure (kPa)	779		779		779	
Volumetric efficiency	0.48		0.66		0.82	
Polytropic index "n"	1.030		1.045		1.058	
Fresh gas fraction "x"	0.20		0.25		0.28	
Characteristic Parameter	**Exp.**	**Calc.**	**Exp.**	**Calc.**	**Exp.**	**Calc.**
Electric power input (W)	68.1	67.3	91.4	90.7	127.3	130.3
Refrigerant mass flow rate (kg/h)	0.484	0.482	1.169	1.156	2.525	2.530
Suction muffler inlet temperature (°C)	66.2	66.1	65.0	65.4	63.7	63.6
Cylinder inlet temperature (°C)	90.8	87.4	88.5	85.2	81.2	79.9
Cylinder outlet temperature (°C)	119.9	121.0	125.8	126.2	120.1	119.8
Compressor outlet temperature (°C)	75.3	79.9	86.4	86.0	91.9	88.1
Recirculated gas temperature (°C)	78.3	72.8	82.0	74.7	80.8	74.7
Shell average temperature (°C)	62.8	63.5	66.9	67.1	67.8	69.1
Electric motor average temperature (°C)	87.7	87.2	89.0	87.8	87.7	87.3

Table 2. Comparison between experimental and calculated parameters relative to a R134a hermetic compressor.

Input Data	T_{evap} = -35°C		T_{evap} = -23.3°C		T_{evap} = -10°C	
Surroundings temperature (°C)	31.9		32.2		32.4	
Compressor inlet temperature (°C)	38		35.8		34.2	
Compressor suction pressure (kPa)	66		114.8		200.5	
Compressor discharge pressure (kPa)	1483		1484		1486	
Volumetric efficiency	0.53		0.71		0.81	
Polytropic index "n"	1.045		1.059		1.065	
Fresh gas fraction "x"	0.325		0.375		0.375	
Characteristic Parameter	Exp.	Calc.	Exp.	Calc.	Exp.	Calc.
Electric power input (W)	91	91.4	134.7	137.2	196.4	196.5
Refrigerant mass flow rate (kg/h)	1.214	1.211	2.847	2.819	5.697	5.712
Suction muffler inlet temperature (°C)	69.6	69.2	71.6	71.0	72.0	71.2
Cylinder inlet temperature (°C)	97.3	98.3	98.9	100.2	94.9	97.0
Cylinder outlet temperature (°C)	150.6	151.5	156.7	157.4	145.5	145.2
Compressor outlet temperature (°C)	85.5	92.8	102.1	105.2	111.9	109.2
Recirculated gas temperature (°C)	86.2	83.7	93.3	91.1	92.7	92.5
Oil temperature (°C)	82.9	86.1	90.3	94.1	91.5	95.8
Shell average temperature (°C)	73.7	73.3	81.0	82.1	83.6	85.9
Cylinder average temperature (°C)	100.1	105.2	109.7	112.5	110.9	111.9
Cylinder head average temperature (°C)	113.7	124.1	129.0	131.8	129.3	131.1
Electric motor average temperature (°C)	95.9	95.7	103.5	102.8	105.4	102.2

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