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Advanced Numerical Simulation Model of Hermetic Reciprocating Compressors

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ADVANCED NUMERICAL SIMULATION MODEL OF HERMETIC RECIPROCATING COMPRESSORS. PARAMETRIC STUDY AND DETAILED EXPERIMENTAL VALIDATION.

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ABSTRACT

A complete and advanced numerical simulation model of the thermal and fluid-dynamic performance of hermetic reciprocating compressors has been developed and is used in the optimization Electrolux compressor design. During the last few years several global experimental validations have been carried out. A detailed instrumentation of a commercial reciprocating compressor for household refrigerators has been developed to determine the average temperature distribution on the surface and the gas at different strategic points in all the compressor zones. This work is focussed on presenting a parametric study of a concrete commercial hermetic reciprocating compressor by means of the numerical simulation model, together with a detailed experimental comparison. The objective is to demonstrate how both global and detailed experimental comparison confirms the possibilities offered by these models.

1 INTRODUCTION

The advanced numerical simulation model of hermetic reciprocating compressors has been carried out to obtain two diferent value groups. The first group are the different global working parameters (volumetric efficiency, power consumption, COP, etc.); the second group is detailed information of the whole compressor instaneous behaviour (pressure, velocity and temperature maps, frequency, angular acceleration, heat flow rates, motor torques, etc.). The model needs some geometric specific information, together with some additional information, specially convective heat transfer coefficients, friction factors and contraction coefficients.

An experimental unit designed and built to analyse single stage vapour compression refrigerant equipment has been numerically compared for each element and for the whole refrigerating system, using R134a with different compressor models. A commercial hermetic reciprocating compressor has been specially built and instrumented with 24 thermocouples to obtain a detailed experimental wall and fluid temperature map. This unit also provides information about compressor inlet and outlet temperature, pressure, together with the mass flow rate.

The objective of this work is to experimentally obtain both the fluid temperature map and the wall temperature distribution map in the whole compressor domain of a specific commercial hermetic reciprocating compressor model. The experimental results obtained are going to be used to validate and contrast the numerical results obtained by means of the simulation program.

2 NUMERICAL SIMULATION

The advanced numerical simulation model solves the thermal and fluid-dynamic behaviour of hermetic reciprocating compressor in the whole compressor domain. The domain is divided into fluid and solid control volumes. The fluid resolution is based on the full integration of the one-dimensional and transient governing equations of the flow (continuity, momentum and energy) in all the fluid compressor zones. The solid thermal behaviour is based on heat global balances at each solid component. The oil heat transfer is also taken into account. The model also carries out at each time-step the force balances in the connecting rod and crankshaft mechanism. The simulation incorporates a multidimensional model for the valve dynamic based on frequency modal analysis, which is able to use spring valves. Finally, the program evaluates the thermodynamic and transport properties for different refrigerant fluids and mixtures using local conditions.

2.1 Mathematical formulation

The governing equations of the flow (continuity, momentum and energy) for a finite control volume formulation together with state equation assuming one-dimensional and transient model are written below and their detailed discretization is presented in references [1], [2] and [3].

$$\frac{\partial m}{\partial t} + \sum \dot{m}_o - \sum \dot{m}_i = 0 \tag{1}$$

$$\frac{\partial m\bar{v}}{\partial t} + \sum \dot{m}_o v_o - \sum \dot{m}_i v_i = (p_i - p_o) A - f \frac{|\dot{m}|\,\bar{v}}{2A} P \Delta z \tag{2}$$

$$\frac{\partial m\bar{h}}{\partial t} - V\frac{\partial \bar{p}}{\partial t} + \sum \dot{m}_o h_o - \sum \dot{m}_i h_i = \sum \bar{\dot{q}}_w A_w \tag{3}$$

$$f(\rho, p, T) = 0 \tag{4}$$

A subroutine in the global program is implemented to improve the valve dynamics evaluation at each time step. Multidimensional valve dynamics model is based on the idea that the valve motion is made up of a superposition of infinite free vibrations [4]. If a compound valve system has to be implemented (i.e. a spring valve is added), this auxiliary valve is considered as a spring-loaded ring valve and one degree of freedom in the valve motion equation is evaluated.

All compressor solid zones (muffler, cylinder head, crankcase, discharge tube and shell, adding to oil zone) are considered macro-volumes and are solved on the basis of heat global balances. The governing equation is the energy equation generalised to any macro-volume considering convection between solid k and fluid i and both conduction and radiation between solid k and solid j neighbours:

$$\frac{\rho_k^o c_{pk}\left(T_k - T_k^o\right) V_k}{\Delta t} = \sum_{cond} \frac{T_k - T_j}{R_{kj}} A_{kj} + \sum_{conv} Q_{ki} + \sum_{rad} h_{radkj}(T_k - T_j) A_{kj} \tag{5}$$

To evaluate the mechanical efficiency it is necessary to know the motor torque applied to the crankshaft and transmitted to the piston at each moment. Evaluating the pressure in the compression chamber at each time-step and the motor torque vs. crankshaft speed, it is possible to evaluate the crank angle degree, instantaneous frequency and motor torque by means of force balance equations, in the piston, in the connecting rod and in the eccentric and crankshaft. After evaluating the motor torque, experimental information about motor electrical efficiency vs. frequency gives electrical efficiency and losses.

In the momentum equation the friction factor, the contraction coefficients and the coefficients to evaluate the effective flow area are the same as used in reference [2]. The empirical inputs in the crankshaft vs. connecting rod force balances are the friction coefficients between cylinder piston, crankshaft piston pin, crankshaft connecting rod and bearings. The thermodynamic and transport properties for different refrigerant fluids and mixtures are evaluated by means of the REFPROP properties program [5].

2.2 Numerical procedure

The numerical procedure to solve the complete compressor (i.e. all the mathematical formulation at the same time) is similar to the idea shown in refernces [1] and [2]. Improvements are found in valve dynamic subroutine together with macro-volumes heat balances and the mechanical and electrical efficiency evaluations. All these formulation is shown in detail in reference [3].

The fluid flow domain is solved in the same manner as with reference [2]. All this domain where the gas is flowing is divided into control volumes. For each grid node the different scalar variables (temperature, pressure and density) are calculated. The temperature and pressure are evaluated at the centre of each node, while the velocity is determined using a staggered grid. A SIMPLE-like algorithm extended to compressible flows has been used [6]. The governing equations (1), (2), and (3) are discretised by means of an implicit control volume formulation, and the convective terms are numerically approximated using the first order upwind numerical scheme. The complete set of discretised momentum, energy and pressure correction equations is solved by the direct method TDMA (Tri-Diagonal Matrix Algorithm).

In the momentum subroutine, the valve dynamic model is solved as an iterative loop, which evaluates the effective force area, and the valve displacement, supposing a gradient pressure in the valve. Valve geometry, the first natural frequencies and the valve motion mode shapes, which are solved by a finite element commercial program is input information needed for the simulation.

For the resolution of the governing equations of the flow it is necessary to know the compression chamber volume at each time-step. It depends on the motor torque force balances. The motor torque equations system is linearly independent, thus is solved directly by means of inverse matrix system LU resolution. Crank angle acceleration is obtained and instantaneous crank angle position is solved using the Heund method.

When the governing equations are solved, it is possible to know all the convective heat knowing the solid wall temperature. It is possible to recalculate the macro-volumes temperatures at each time step or at the end of each iterative cycle, solving the energy macro-volume equations system. The equation system is carried out to be linearly independent, thus the equations group is solved directly by means of an inverse matrix system LU resolution, and the results are the new macro-volumes temperatures.

The program solves n time steps per cycle obtaining a mean frequency, the following time step per cycle is readjusted based on the last frequency obtained. The program is considered converged when differences between the last and following cycles in all fluid temperatures, pressures, velocities maps and solid temperatures reach the required accuarcy.

3 EXPERIMENTAL SET-UP

A commercial hermetic reciprocating compressor has been instrumented with 24 thermocouples to obtain an experimental temperature compressor map. Results are numerically compared and a detailed experimental validation has been carried out. Figure 1 shows the compressor with lid off and instrumented with all the internal thermocouples and vacuum pressure feedthroughs. Figure 2 shows thermocouples through different solid parts and between the shell and the crankcase. Experimental set-up is also explained in detail in reference [7].

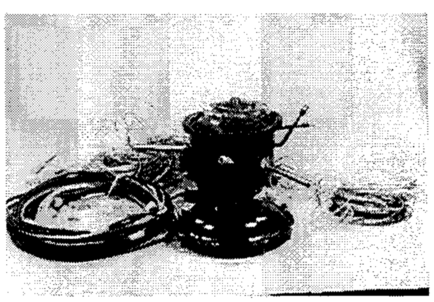

Figure 1: General compressor view.

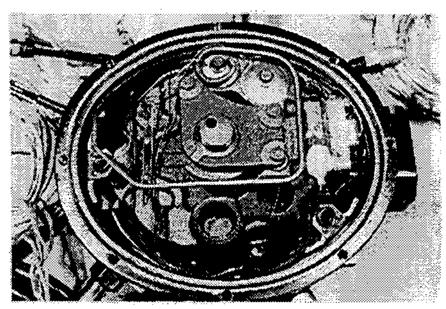

Figure 2: Thermocouples inside compressor.

3.1 Hermetic reciprocating compressor instrumentation

Twenty internal thermocouples (K type) have been built in four thermocouple groups with five thermocouples in each group. Thermocouple groups pass through the shell with vacuum pressure feedthroughs. Thermocouples inside the compressor shell are encapsulated with steel, and electrically insulated with MgO between two wires and the steel thin stick. Outside the compressor shell, thermocouples are covered with a steel grid. A third pin connects the steel capsulate inside the shell with a steel grid outside the shell and it is connected to the ground. Finally, 4 thermocouples measure external compressor shell wall temperature.

3.2 Thermocouple groups and positions

Several holes (0.5 mm diameter) have been necessary in different internal compressor pieces: i) plastic muffler to measure fluid at suction orifice; ii) cylinder head to measure fluid at discharge orifice, fluid at cylinder head and internal cylinder head wall; iii) crankcase to measure wall temperature at different compression chamber distances and iv) discharge chamber walls to measure fluid at discharge chambers.

Figure 3 shows thermocouples in cylinder head and valve plate through their holes in detail. Figure 4 shows several thermocouples at their positions in detail: fluid temperature between shell and crankcase, temperature in cylinder head and wall crankcase temperature. Thermocouples through their holes have been covered with an electrical and thermal insulator sealed paste, to maintain the piece hermetically sealed, thus avoiding contact between the piece and the thermocouple. Wall temperature thermocouples have been pasted to the wall with a high temperature and thermally conductive paste to obtain a more accurate wall temperature.

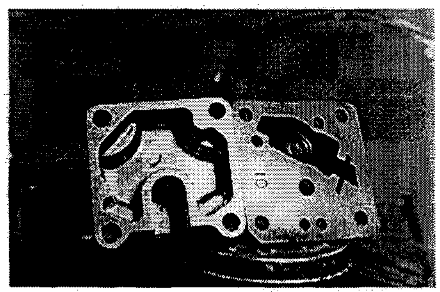

Figure 3: Cylinder head and valve plate details.

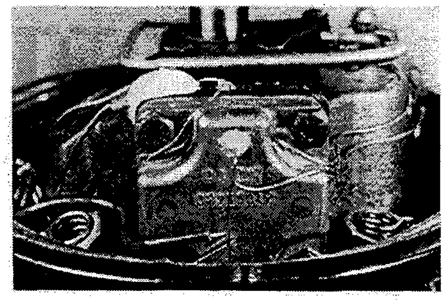

Figure 4: Cylinder head and crankcase details.

3.3 Thermocouple calibration

Thermocouples have been calibrated by means of a thermostatic refrigerating unit working at different temperatures and using mineral oil. A precision Platinum Resistance Thermometer is used as the reference value. Both thermometers have been put together. PTR is read with an individual data acquisition model. The PTR system accuracy is $\pm$ 0.025°C. The Data Acquisition and Control System DACS used guaranties an accuracy of $\pm$ 0.35°C while internal isothermal reference temperature blocks in DACS are between 23°C $\pm$ 0.5°C. Thermocouples K type have accuracy before calibration of $\pm$ 1.1°C, after calibration this error is state to uncertainty.

3.4 Experimental unit description

An experimental unit has been built to study single stage vapour compression refrigerating systems. This unit has been designed to validate mathematical models of the thermal and fluid dynamic behaviour of single stage vapour compression refrigerating units in general, and specially for condensers, evaporators and hermetic reciprocating compressors.

Elements that make up the experimental unit are basically: hermetic reciprocating compressor, double-pipe condenser and evaporator, capillary tube and different tube connections. Auxiliary fluid for condenser and evaporator is water. The fluid flow temperatures inside tubes and annuli are measured with calibrated platinum resistance thermometer sensors Pt-100. These sensors are located at the inlet and outlet sections of each element of the main circuit and secondary circuits. Condenser and evaporator pressures are measured by transducers, the accuracy is within $\pm 0.1\%$. Mass flow rate inside main circuit is measured with a Coriolis type mass flow-meter, the accuracy is within $\pm 0.2\%$. Volumetric flow in secondary circuits is measured with magnetic flow-meters, the accuracy is from $\pm 0.1\%$ to $\pm 0.5\%$. Temperature in auxiliary circuits is controlled by thermostatic units and two modulating solenoid valves control mass flow rate in these circuits. The Data Acquisition and Control Unit HP E1300A, and a personal computer, process all compressor and experimental unit information. References [7], [8] and [9] explain the experimental unit described in detail.

4 RESULTS

In this work the complete numerical compressor simulation model has been presented, while the hermetic reciprocating compressor instrumentation and the experimental set-up prepared to work with commercial compressors have also been studied. The mathematical model developed needs different additional information, a part of this information are the convective heat transfer coefficients between the refrigerant gas and the solid walls. A detailed experimental test of the average temperatures of both fluid and surfaces in all the compressor zones has been carried out to obtain an accurate information of these global heat fluxes. The objective of this work is to demonstrate the possibilities of the numerical simulation model and how a detailed experimental temperature map comparison is able to help for a better evaluation of the heat transfer coefficients.

4.1 Global validation

Several parametric studies and global parameter comparisons between numerical simulation results and experimental data have been presented in references [3], [7], [8] and [9]. These results show the possibilities of the numerical simulation presented here. Differences are typically lower than 10% in mass flow-rate, volumetric efficiency and COP. An hermetic reciprocating compressor has been tested in an Electrolux calorimeter before being instrumented. All input data information is: inlet temperature, inlet and outlet pressure, compressor geometry, valve dynamic modal analysis and eletrical motor curve. No other data is necessary, the rest of the information is output data. However, direct gas mass fraction from inlet compressor to suction muffler is experimentaly approximated to 0.3%. This value is closed to data considered in references [10] and [11]. Global results are compared in Table 1 considering ASHRAE conditions: 32 °C ambient and inlet temperature and 55°C condenser temperature. Four different cases are evaluated considering fluid refrigerant R-134a. Results show a good agreement.

Table 1: Comparative global results: calorimeter test vs. numerical simulation.

	Experimental data						*Numerical results*					
T_{evap} (C)	$\dot{m}$ (kg/h)	η_v (%)	T_{out} (C)	ω_{real} (Hz)	$\dot{W}_E$ (W)	COP	$\dot{m}$ (kg/h)	η_v (%)	T_{out} (C)	ω_{real} (Hz)	$\dot{W}_E$ (W)	COP
-10.0	7.03	64.4	109.5	48.50	207	1.75	7.49	67.7	109.0	48.86	191.5	1.83
-23.3	3.58	57.7	104.5	49.00	143	1.30	3.91	62.0	107.1	49.27	135.5	1.41
-30.0	2.31	51.0	97.0	49.25	113	1.07	2.40	53.6	99.1	49.49	104.8	1.14
-35.0	1.54	43.6	89.0	49.50	93	0.87	1.78	48.6	93.0	49.59	89.8	0.99

4.2 Detailed comparative results

After compressor instrumentation, a detailed experimental results of the average temperatures of both fluid and surfaces in all compressor zones has been carried out. Numerical results are compared with experimental data in all these cases.

Table 2 show experimental compressor temperature map for fluid flow and solid walls at different compressor points in the experimental unit. Boundary experimental conditions of these cases are exactly the same as Table 1 boundary conditions.

The heat transfer coefficients in the hermetic reciprocating compressor numerical simulation model has been divided in several groups. First group are inlet heat transfer coefficients in tubes and chambers inside compressor muffler, crankcase and discharge tube. These coefficients are evaluated using usual empirical correlations referenced in [3]. The convective heat transfer coefficient in compression chamber used is referenced in [12]. The heat coefficients between the shell and environment are the same referenced in [10]. The oil energy balance and the mechanical and electrical heat transfer losses are considered as reference [11]. Second group corresponds to the external suction muffler, external cylinder head wall, external crankcase wall together with internal shell wall and external discharge tube walls. This last values group is based on the heat transfer coefficients referenced in [10] and [13] with values ranging between 50 W/m ^{2o}C and 200

W/m^{2o}C. Numerical results presented are evaluated considering this heat transfer coefficient constant and equal to 150 W/m^{2o}C. Detailed comparative values for numerical results and experimental data are shown in Figure 5.

Table 2: Experimental values at different compressor points for Tevap= -10.0C; -23.3C, -30.0C and -35.0C.

Fluid temperatures					***Internal solid temperatures***				
1 Inlet compressor	32.5	33.0	33.0	33.0	13 Muffler	77.1	82.3	83.0	84.9
2 Inlet suction muffler	69.3	71.8	69.8	66.1	14 Comp. chamb.	113.4	114.8	109.6	103.8
3 Suction orifice	81.8	86.5	88.3	87.7	15 Comp. chamb.	111.4	112.3	102.3	102.0
4 Discharge orifice	134.9	146.1	145.7	141.9	16 Ext. crankcase	88.8	88.2	84.9	84.9
5 Cylinder head	139.3	142.5	135.2	120.6	17 Ext. crankcase	85.8	87.2	87.6	84.9
6 Discharge chamber 1	126.0	122.9	112.2	103.0	18 Cylinder head	128.5	131.1	124.4	116.6
7 Discharge chamber 2	118.4	115.0	106.4	98.1	19 Int. shell	70.8	72.6	70.5	66.9
8 Outlet compressor	109.5	104.5	97.0	89.0	***External solid temperatures***				
9 Remnant gas top	94.3	97.6	92.8	86.7	20 Bottom shell	72.8	74.0	71.3	67.4
10 Remnant gas top	91.5	93.7	91.0	87.2	21 Middle shell	77.1	78.3	74.9	72.5
11 Remnant gas middle	81.7	83.4	82.0	80.8	22 Top shell	82.0	83.3	79.2	76.1
12 Remnant gas middle	82.5	85.9	86.2	83.0	23 Oil	87.9	88.5	86.9	83.0

Figure 5 shows detailed numerical results obtained in the same cases of Table 1 and compared with experimental data of Table 2. Results presented have been carried out without taking into account a parametric influence study of the heat transfer coefficients commented above. Possible correlations improvements or high-level numerical simulation model in specific compressor parts are going to be future action to obtain better agreement. Instead of that and together with global numerical comparative results presented above, numerical simulation model present reasonable agreement results. It is important to point out that no attempt has been made to tune the empirical coefficients to the experimental data.

Temperature results form points 2 to 8 are fluid temperatures, which are numerically underestimated in suction line and reasonably good in discharge line. Temperature 1 is boundary condition. Temperature points from 9 to 12 are solid temperatures, which have good agreement compared with experimental data. Numerical solid temperatures are obtained by means of global heat balances over solid parts, while experimental temperatures are basically local wall surface temperatures. This is one important difference between numerical results and experimental data.

4.3 Detailed numerical analysis

After global validation results and detailed comparative cases, a detailed numerical analysis is shown to demonstrate the possibilities of this kind of models and numerical information obtained. Figure 6 shows for the cases presented below: pressure vs. volume diagram; pressure vs. crank angle degree and pressure in suction orifice, compression chamber and discharge orifice vs. crank angle degree. These results allows to obtain local and instantaneous numerical values.

It is interesting to take into account that motor torque, piston position and mechanical and electrical losses are evaluated at each crank angle degree. Thus, P-v diagram shows local values of compression chamber volume. Figure 7 shows instantaneous frequency, motor torque and resistance torque at each crank angle degree. This Figure allows to know differences depending on evaporation temperature; showing that curve evolution is very similar.

Finally last illustrative results are shown in Figure 8. Heat transfered in compression chamber is presented for the cases studied above together with the instantaneous compression power for the same cases.

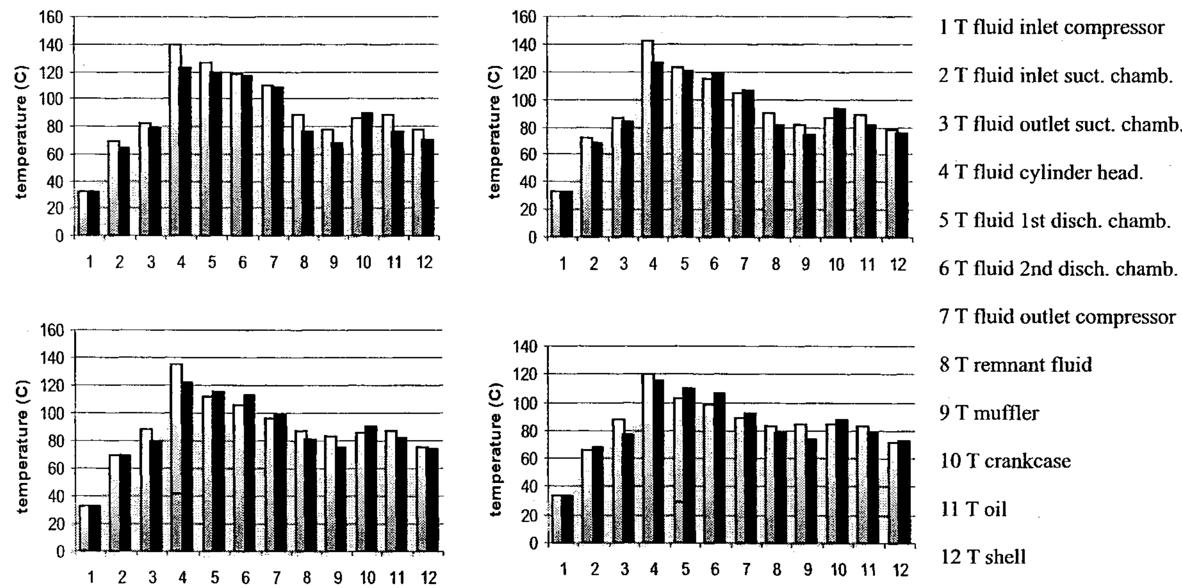

Figure 5: Experimental and numerical temperature values at different compressor points.

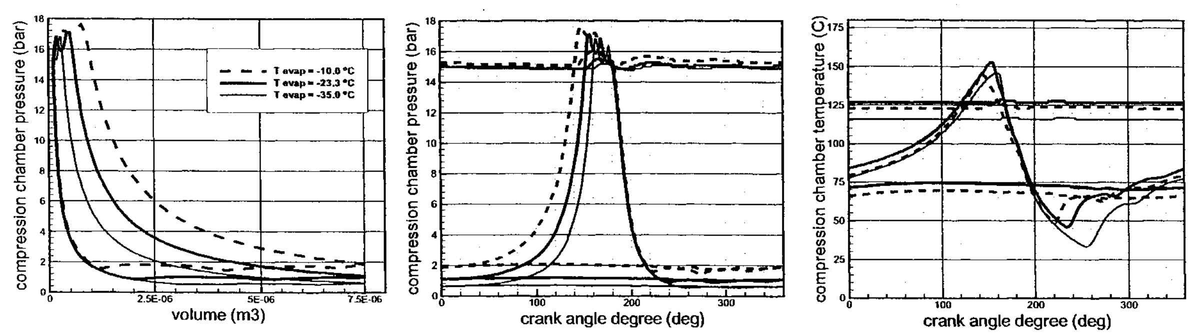

Figure 6: Numerical pressure values in compression chamber vs. volume and crank angle degree.

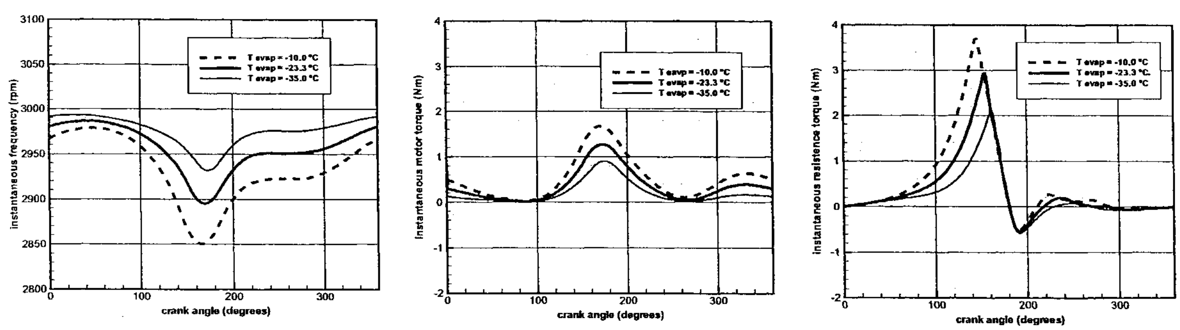

Figure 7: Instantaneous frequency; motor resitence and motor torque.

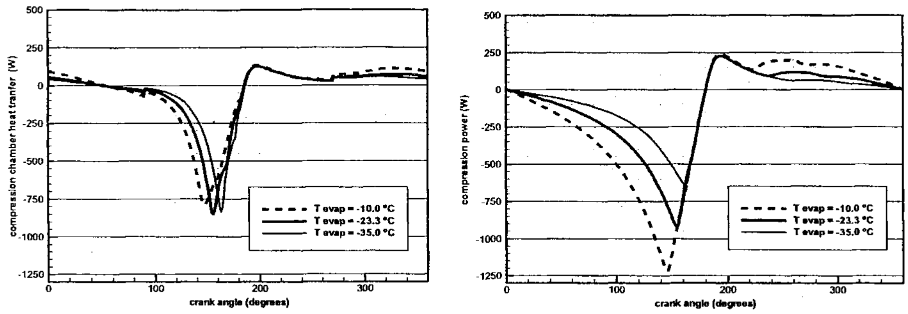

Figure 8: Instanteneous heat transfer and compression power.

5 CONCLUSIONS

A complete numerical simulation of the whole compressor domain has been presented. A detailed instrumentation of a commercial hermetic reciprocating compressor has been carried out. Detailed empirical temperature information has been shown. The parametric study shows the additional information need and its importance in the numerical simulation. Finally detailed numerical analysis shows the importance of these kind of numerical models to optimise the compressors design, and the importance to obtain general and accurate detailed numerical or experimental information to feed the program presented.

6 ACKNOWLEDGEMENTS

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