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A STUDY ON THE POLYTROPIC EXPONENT OF RECIPROCATING HERMETIC COMpressors

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ABSTRACT

The present paper is concerned with the study of the polytropic exponent of reciprocating hermetic compressors. The polytropic exponent finds application in the development of simpler simulation models for hermetic compressors. In such cases it is usually taken as a fixed input value. A previous study has shown that, for a given open reciprocating compressor, it remains fairly constant, even when the compressor is run with different refrigerants. In the present paper experimental data from hermetic reciprocating compressors have been analyzed to investigate the behavior of the polytropic exponent under different operating conditions. A mathematical model, based on mass, energy and momentum balance equations, and using data from standard calorimetric tests as input, was developed to determine the polytropic exponent. Results have shown that, for hermetic compressors, the assumption of a constant polytropic exponent is still valid.

INTRODUCTION

Simulation models have become an important tool in the analysis of heat pump, air conditioning and refrigeration systems. For the simulation of the compressor, a significant number of these models adopt the polytropic compression assumption. It only requires, for the characterization of the compression process, the knowledge of the polytropic exponent, which is regarded as a fixed empirical value. This methodology has been used, for example, in the simulation of systems employing open (Herbas et al., 1993; Monteiro et al., 1995) as well as hermetic reciprocating compressors (Suefuji and Nakayama, 1980; Domanski and Didion, 1983; Domanski, 1986; Marques and Melo, 1990; Yuan and O'Neal, 1994).

The objective of the present work is to determine how reasonable is the assumption of a constant polytropic exponent. A previous paper (Pereira et al., 1995) has shown that, for a given open reciprocating compressor, the polytropic exponent of compression remains fairly independent of the operating conditions. Experimental data from five different compressors, running on air, R12 and mixtures of R32/R152a and R32/R134a, were employed in the analysis. For each run, the exponent was calculated from the pressure and temperature measurements, taken at the suction and discharge of the compressor. Results for the zeotropic mixtures have also demonstrated that, if different working fluids, with otherwise similar properties, are to be tested in the same compressor, the polytropic exponent does not vary significantly. For a typical compressor it was shown that the uncertainty in the determination of the polytropic exponent, taken as an average from the error band of the data points, would result in percentage deviations of ± 0.5%, ± 2% and ± 2%, for the refrigerant mass flow rate, the discharge absolute temperature and the compression work, respectively.

The present paper extends this analysis to hermetic reciprocating compressors. From the modelling point of view, the hermetic compressor is a far more complex system than the open compressor, as it also involves the energy interactions of the with the electric motor, mufflers, discharge line and the
shell environment. The polytropic exponent is calculated, of course, from the thermodynamic states of the gas at the suction and discharge plena. Its evaluation from external states (shell inlet and outlet) is conceptually erroneous and, not surprisingly, leads to inconclusive results (Akella et al., 1986). However, the values of pressure and temperature at these points are not customarily available, only if fully instrumented compressors (usually called “thermocoupled” units) are employed. In the present work data were obtained from standard calorimetric tests, which means that only external pressures and temperatures, taken from outside the compressor shell, were available. Therefore, a method to determine the polytropic exponent had to be devised. In the following sections a brief description on the experimental data and the model is presented. Results are compared with those of a recent study on hermetic compressors (Popovic and Shapiro, 1995).

EXPERIMENTAL DATA

As explained above, no direct measurements of the pressure and temperature at the suction and discharge plena were taken. Instead, tests were performed with a typical calorimeter, with the following quantities being measured: electric power input, inlet and outlet pressures, inlet and outlet temperatures, electric motor speed, external temperatures at the bottom and top of the shell, refrigerant mass flow rate, cooling air volumetric flow rate and ambient conditions (pressure, temperature and relative humidity). Efficiency and load curves of the electric motor were also available, as well as the compressor geometry: shell (height, top and bottom diameters), clearance ratio, bore and stroke.

MATHEMATICAL MODEL

The model was based on that developed for the HPSIM heat pump model, by Domanski and Didion (1983). Figure 1 shows the schematics of the compressor, showing the control volumes and energy flows considered. The same numbering sequence (3 for inlet and 8 for outlet) of Domansky and Didion (1983) was used. The hermetic compressor, consisting of an electric motor and a cylinder-piston assembly, all contained in a shell, was divided into a number of control volumes. These volumes comprised: shell material (two control volumes: one consisting of the top and side surfaces of the shell and the other, in contact with the oil, covering the bottom), suction side (internal shell environment), oil volume, cylinder, suction and discharge conduits (each control volume encompassing the respective mufflers and plena) and the discharge line.

Main Assumptions

Simplifying assumptions were needed to formulate the compressor model. They are as follows:

1. All processes take place in steady-state regime. The effects of any dynamic phenomenon, such as gas pulsation, are only taken into account in the global performance of the compressor;
2. Shell material (top and bottom) and environment (suction side) control volumes as well as the oil have uniformly distributed properties;
3. Thermophysical properties of the gas are uniformly distributed in all the sections that separate the control volumes;
4. The power losses due to the electric motor and mechanical inefficiencies are accounted for;
5. The following heat exchange rates are considered: between the refrigerant and can control volumes, \(Q_{C\text{Al}}\), refrigerant and lubricant oil, \(Q_{C\text{Oll}}\); refrigerant and cylinder external surface, \(Q_{XS}\); refrigerant and electric motor, \(Q_{M\text{Ox}}\); lubricant oil and can, \(Q_{C\text{Al}}\); refrigerant and suction conduits, \(Q_{45}\); refrigerant and discharge conduits, \(Q_{67}\); refrigerant flowing through the shell environment and discharge tube, \(Q_{78}\).
6. The compression and expansion processes in the cylinder are polytropic, with the same exponent.

Energy and Mass Flows

The compressor model is a result of the equations describing the energy and mass balances over each control volume, as well as equations for refrigerant pressure drop across valve and line passages and heat transfer equations describing the interaction between the control volumes. Below is a description of the energy and mass flows across the boundaries of the control volumes.

- **Overall control volume:** Refrigerant flows across the control surface through the inlet and outlet tubes. There is an inflow of electrical energy as well as heat losses from the compressor shell to the surrounding air.

- **Shell environment control volume (path 3-4):** Refrigerant enters the compressor can through section 3. One fraction of the mass flow goes directly to the suction muffler \( m_N \) and the remaining flow \( m_X \) is directed towards the electric motor, to cool motor windings, and to the other parts of the compressor (can, manifolds, discharge tube and the cylinder body), with which it exchanges heat. After that this flow \( m_X \) mixes with the other fraction \( m_Y \), restoring the total refrigerant mass flow rate, and enters the suction muffler. The power loss due to the electric motor inefficiency goes to the refrigerant as \( Q_{Mox} \). The power loss due to mechanical friction goes to the oil.

- **Suction conduits control volume (path 4-5):** The refrigerant undergoes pressure drop and heat transfer with the surfaces of the suction conduits (muffler, suction plenum and valve passages).

- **Cylinder control volume (process 5-6):** For the compression process, figure 2, the mass flow rate, compression work and heat transfer rate are evaluated from assumption 6.

- **Discharge conduits control volume (path 6-7):** Process equivalent to 4-5.

- **Discharge tube control volume (path 7-8):** Discharge refrigerant pressure drop and heat loss.

Mathematical model

The mass, energy and momentum balance equations were applied to each of the control volumes and, combined with the corresponding heat transfer and pressure drop equations, resulted in a nonlinear system of 29 algebraic equations. Following the same procedure established by Domanski and Didion (1983), the equations were rewritten so that geometric, heat transfer and pressure drop parameters (independent of the operating conditions and refrigerant properties) could be identified. This resulted in 36 unknown variables, divided in heat transfer rates, compressor parameters (the polytropic exponent included) and refrigerant thermodynamic states. This required seven additional equations that had to be obtained from additional hypotheses, as follows:

- Variations of the refrigerant specific volume between points 6 and 8 are negligible;
- Pressure drops \( \Delta P_{34} \) and \( \Delta P_{78} \) are taken from the literature (Paczuski, 1982);
- Temperature increment \( \Delta T_{45} \) is also taken from the literature (Meyer e Doyle, 1990; Todescat et al., 1992);
- An empirical factor, \( \xi \), is proposed to ensure that the temperature in section 7 is in the range defined by \( T_6 \) and \( T_8 \). Typical values for \( \xi \), an input value for the model, are evaluated from existing experimental data (Meyer e Doyle, 1990).

\[
\xi = \frac{T_6 - T_7}{T_6 - T_8}
\] (1)
• The temperature of the refrigerant flowing inside the compressor shell, $T_X$, should also be limited by $T_{CAS}$ and $T_s$. For this purpose an additional factor, $\gamma$, is proposed and assessed from the literature (Meyer e Doyle, 1990).

$$\gamma = \frac{T_X - T_{CAS}}{T_s - T_{CAS}}$$

(2)

• Pressure drop between sections $x$ and 4 is negligible. This means making $P_x = P_4$ and transferring the $x$-$4$ entrance effect to the pressure drop of path $\gamma$-$5$.

Using the above assumptions, a system of 36 equations was obtained and solved. Shortage of space precludes a full treatment to be presented. Further details can be found in Motta (1995).

RESULTS AND CONCLUDING REMARKS

Standard calorimetric tests were performed for an existing hermetic compressor working with R-22, under several operating conditions. Figures 3, 4 and 5 show the polytropic exponent values plotted against the compression ratio, the suction pressure and mass flow rate, respectively. The figures show the polytropic exponent lying between 1.1 and 1.21, with no observed trend for either an increase or decrease. These results conflict with recent studies from Popovic and Shapiro (1995), where the polytropic exponent was found to decrease with the compression ratio. Experimental data, contrary to the limited information of the present work, covered two compressors and several different refrigerants, including mixtures. On the other hand, the model from Popovic and Shapiro (1995), to determine the polytropic exponent from external inlet and outlet states, did not consider the heat transfer processes between the refrigerant in the compressor shell and the electric motor, oil, discharge line and suction conduits. Besides, the disposition of the suction muffler, which determines the $x$-$y$ flow distribution, has not been considered. All these effects, detached from the actual compression process, may be have contributed for the trend observed. Further research is needed. Model limitations could be overcome with direct measurements of the refrigerant states at the discharge and suction chambers.

REFERENCES


Figure 2 - Cylinder control volume.

Figure 3 - Polytropic exponent against compression ratio.

Figure 4 - Polytropic exponent against suction pressure.

Figure 5 - Polytropic exponent against mass-flow rate.

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