Simplified modelling of an open-type reciprocating compressor

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Abstract

The first part of this paper presents a detailed experimental analysis of an open-type reciprocating compressor equipped with internal sensors. The analysis reveals the main processes affecting the refrigerant mass flow rate as well as the compressor power and the discharge temperature.

Based on these experimental results, a simplified steady-state model of the reciprocating compressor is proposed. The refrigerant mass flow rate is supposed to be affected by the clearance volume re-expansion, by a pressure drop occurring through a supply flow restriction and by a temperature increase due to some heat transfer from a supposed-to-be isothermal wall. The friction power loss is supposed to be transferred to this fictitious wall, which is also exchanging heat with the discharged gas and with the ambient.

The model is able to determine the ambient losses and so, the exhaust temperature.

The compression itself is considered as adiabatic, reversible and therefore isentropic. The friction power loss is decomposed into a constant contribution and another one proportional to the isentropic power. © 2002 Éditions scientifiques et médicales Elsevier SAS. All rights reserved.

Keywords: Reciprocating compressor; Internal measurements; Internal heat transfers; Pressure drops; Manifold; Power losses; Model

1. Introduction

Refrigeration by vapour compression cycle is still the most widely used cycle in domestic as in industrial applications, like air conditioning, domestic refrigeration, food processing, and many other industrial process. The technology moving has been particularly active in the change of refrigerant and in the development of new types of compressors in the past decades. The importance of reciprocating compressors has stabilized at a lower level in North America, while in South America, Europe and Asia, it has maintained a strong position. The service and repair equipments remain less sophisticated for the reciprocating compressors than for other ones (rotary and centrifugal) [1].

The object of this paper is to show that a simple phenomena oriented model can be accurate enough to predict the performances of a refrigeration compressor.

Typical relative accuracies are about ±5%. The example considered is an open-type reciprocating compressor.

Different degrees of refinement can be found in the literature in the modelling of reciprocating compressor. Many models aim to study one particular effect, like valve flow, cylinder heat transfer, bearing losses, noise, etc. [2]. A few studies only aim to present the complete process. Detailed models, which take into account the detailed geometry and characteristics of the compressor (valves geometry and stiffness, etc.), require a large number of parameters which are known … only by the manufacturer [3–5]. This kind of models is useful for the compressor designer.

On the other hand simplified models are necessary to design and to control a refrigeration plant, to minimize energy consumption, to maximize production, to detect fault and to make diagnosis. Simple models can be polynomial models based on statistical correlations on system or on compressor performance variables (COP, cooling capacity, efficiencies) or based on physical assumptions with the parameters having more physical sense [6,7]. The polytropic approach is the most widely used for simple com-

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Nomenclature

- **AU**: heat transfer coefficient \( \text{W} \cdot \text{K}^{-1} \)
- **C**: coefficient
- **c**: specific heat \( \text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1} \)
- **h**: specific enthalpy \( \text{J} \cdot \text{kg}^{-1} \)
- **M**: mass flow rate \( \text{kg} \cdot \text{s}^{-1} \)
- **N**: revolution speed \( \text{Hz} \)
- **p**: pressure \( \text{bar} \)
- **Q**: heat flow rate \( \text{W} \)
- **s**: specific entropy \( \text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1} \)
- **t**: temperature \( ^\circ \text{C} \)
- **T**: Torque \( \text{N} \cdot \text{m} \)
- **v**: volume \( \text{m}^3 \cdot \text{kg}^{-1} \)
- **V**: volume \( \text{m}^3 \)
- **\dot{V}**: volume flow \( \text{m}^3 \cdot \text{s}^{-1} \)
- **W**: power \( \text{W} \)

Subscripts

- **amb**: ambient
- **c**: clearance
- **comp**: compressor
- **ex**: (machine) exhaust (as indicated in Fig. 9)
- **ex1**: after discharge cooling down (as indicated in Fig. 9)
- **ex2**: after isentropic compression (as indicated in Fig. 9)
- **f**: factor
- **in**: internal
- **loss**: losses
- **loss0**: constant losses
- **mes**: measured
- **mot**: motor
- **p**: at constant pressure, parasite
- **s**: isentropic, swept
- **sh**: shaft
- **sim**: simulated
- **su**: (compressor) supply (as indicated in Fig. 9)
- **su1**: after supply throttling (as indicated in Fig. 9)
- **su2**: after supply heating up (as indicated in Fig. 9)
- **V**: at constant volume
- **w**: wall
- *****: measured in the cylinder head

Greek symbols

- **α**: loss factor
- **δ**: dimensionless diameter
- **Δ**: difference
- **ε**: efficiency

2. Description of the test apparatus

Since this study focuses on the compressor only, the tests were carried out in a fully gas network configuration, i.e., without condensation. This configuration presents the following advantages with respect to a conventional refrigeration loop with phase change: lower thermal capacity of the system, easier control, smaller refrigerant charge, no risk of getting liquid refrigerant at the suction inlet of the compressor and lower energy consumption [14].

A control valve was placed at cooler supply. The main drawback of a low pressure cooler configuration is the limit on the minimum suction temperature imposed by the temperature of cold water. Note that in the case of a high pressure gas cooler configuration, the temperature limit is imposed by the risk of condensation.

The test bench (Fig. 1) is composed of:

- an electric asynchronous motor,
- a valve that controls the compressor supply-exhaust pressure difference,
- a concentric, water cooled by-pass heat exchanger,
- a high pressure vessel and a low pressure refrigerant vessel.
A 2-cylinders open type compressor is installed on this test bench; the compressor displacement is 680 cm$^3$. The refrigerant used is R12.

Pressure and temperature measurements are made at all characteristic points of the refrigerant circuit. All measurements are recorded using an acquisition card with a time period of 4 s. Hall effect sensors are used for pressure measurements. The ranges used are: 0.2–9 bar (absolute) for low pressure side and 1–31 bar for high pressure side of the test apparatus. An error of 0.1 bar is estimated after calibration. Temperatures are measured by $T$ type thermocouples. The thermocouples are installed inside copper gloves inserted inside refrigerant and water pipes excepted for temperature measurements inside the cylinder head (see next section). Maximum error on temperature is estimated to 0.3 K.

Refrigerant mass flow rate is calculated from temperature and flow measurement on the secondary side of the gas cooler assuming steady state operation. Secondary side fluid (water) is measured by weighting. The final error on the refrigerant mass flow rate is estimated to ±3%.

The following measurements are specifically made on the compressor:

- the machine supply and exhaust pressures and temperatures: $p_{su}$, $t_{su}$, $p_{ex}$ and $t_{ex}$,
- the cylinder supply and exhaust temperatures and pressures (just before the suction valves and just after the discharge valves): $t_{su*}$, $p_{su*}$, $t_{ex*}$ and $p_{ex*}$,
- the rotation speed: $N$,
- the electric consumption of the motor: $\dot{W}_{mot}$,
- the compressor shaft torque: $T_{sh}$.

In order to measure the shaft torque, the compressor casing had to be put on external bearings, in such a way to release the fixation reactions. Fig. 2 shows the test bench, where the compressor casing, the bearing and the rigid rod that acts on the load cell can be seen. Two weights had to be fixed to the compressor base to avoid vibrations. Flexible piping was placed in the refrigerant network to release the piping reactions as much as possible. Nevertheless, the torque measurement had to be calibrated in order to account for the parasite torque ($T_p$) effects generated by the piping.
3. Test results

Two types of results are presented hereafter:

(1) Global results, where all the input and output variables were measured simultaneously. The suction heating-up and discharge cooling down in the cylinder head were also measured.

(2) Partial results of the tests that were carried out apart in order to identify the pressure drop in the suction manifold.

3.1. Global tests

A series of 25 tests was carried out at \( N = 366, 495 \) and 665 rpm. The test results are given in Table 1.

**Heat transfer in the manifold.** Thanks to the thermocouples located in the cylinder head just at the supply and exhaust valves, it was possible to measure directly the heating-up and cooling down in the manifold.

The most important factor that affects the heat transfer in the manifold is the pressure ratio as it can be seen in Fig. 5. When the pressure ratio increases, the difference between exhaust and suction temperatures increases and this give higher supply heating-up and exhaust cooling down. The supply fluid heating-up is much higher than the exhaust cooling-down (these temperature differences reach 32 K and 5 K, respectively).

**Volumetric and compressor isentropic efficiencies.** In order to analyse the compressor performance at different operating conditions, the volumetric and compressor isentropic efficiencies were computed for the tests presented before.

The volumetric efficiency is given by

\[
ε_v = \frac{\dot{V}_{su}}{NV_s} \tag{1}
\]

where \( V_s \) is the swept volume and \( \dot{V}_{su} \) is the volume flow in the suction inlet:

\[
\dot{V}_{su} = \dot{M}v_{su} \tag{2}
\]

The compressor isentropic efficiency is given by:

\[
ε_{comp,s} = \frac{w_s}{w_{sh}} \tag{3}
\]
Table 1
Tests results

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Fig. 6. Volumetric efficiency versus pressure ratio.

where $w_{sh}$ is the isentropic work of compression and $w_{sh}$ is the specific shaft work:

$$w_{sh} = \frac{W_{sh}}{M}$$  \hspace{1cm} (4)$$

Figs. 6 and 7 present the volumetric and compressor isentropic efficiencies versus the pressure ratio obtained for the tests. From these figures, it could be misleadingly thought that the measurement uncertainties are relatively high. But actually, the experimental points in these figures do not have to fit a single curve since $\varepsilon_v$ and $\varepsilon_{comp.a}$ depend not only on the pressure ratio, but also mainly on the fluid inlet conditions (temperature, pressure).

The decrease of the volumetric efficiency versus the pressure ratio is typical of the clearance volume re-expansion effect. Note that there is no significant influence of the rotation speed on the results for the range of speed considered.

On the other hand, Fig. 7 clearly shows that speed lowers the compressor isentropic efficiency. The decrease of the isentropic efficiency at low pressure ratio is due to the stronger effect of the valve losses and relatively stronger effect of the constant mechanical losses.
3.2. Throttling through the suction manifold

The pressure drop in the suction manifold was measured by the difference of the pressures given by the sensors installed in machine and cylinder supplies respectively, as described in Fig. 4.

Two series of tests were carried out, one at 366 rpm and the other at 665 rpm. The results are shown in Fig. 8. The total (stagnation) pressure difference was considered in order to account for the change of transversal area between machine and cylinder supplies.

These results obtained for different conditions show that a fixed area obstacle model is convenient for manifold pressure drop prediction. This fixed area obstacle is comparable to an orifice plate.

4. Compressor modelling

The conceptual schema of the compressor is presented in Fig. 9. The evolution of the refrigerant is decomposed into five steps [15]:

(i) Pressure drop (su → su1).
(ii) Heating-up (su1 → su2).
(iii) Isentropic compression (su2 → ex2).
(iv) Cooling down (ex2 → ex1).
(v) Pressure drop (ex1 → ex).

The pressure drop is represented according to the conceptual schema of Fig. 10 [13]:

(1) Isentropic expansion su → su1s in the nozzle,
(2) Isobaric diffusion su1s → su1.

This decomposition of the refrigerant evolution permits one to obtain a relationship between the pressure drop and the obstacle area through a globally isenthalpic process.

If the pressure drop is much smaller than the entrance pressure, the fluid may be considered quasi-incompressible and we obtain:

\[ \dot{M} = \frac{\pi d_{su}^2}{4} \sqrt{\frac{2\Delta P_{su} \rho_{su}}{\Delta P_{ex}}} \tag{5} \]

where \( \dot{M} \) is the refrigerant mass flow rate, \( \Delta P_{su} \) the pressure drop at the suction, \( \rho_{su} \) the density and \( d_{su} \) is the fictitious nozzle throat diameter (m). A dimensionless diameter can be defined as:

\[ \delta_{su} = \frac{d_{su}}{\sqrt{v_s}} \tag{6} \]

The high discharge valve stiffness makes almost constant the corresponding pressure drop. Hiller and Glicksman [17] give a range of discharge pressure drop \( \Delta P_{ex} \) between
0.7 and 2.1 bar. A first modelling approximation consists in identifying a constant exhaust pressure drop. If more precision is required, a term proportional to the quadratic mass flow rate should be used.

**Heat transfer.** In general, the different heat transfers in a compressor include the one to the suction gas $\dot{Q}_{su}$, the heating-up due to the electromechanical losses $\dot{W}_{loss}$, the heat given by the high temperature discharge gas $\dot{Q}_{ex}$ and the heat flow to the ambient $\dot{Q}_{amb}$.

We assume that the definition of a fictitious wall uniform temperature $t_w$ is sufficient to represent all the heat transfer modes mentioned above, as shown in the conceptual schema of Fig. 9. Steady state balance for this wall gives:

$$\dot{W}_{loss} + \dot{Q}_{ex} - \dot{Q}_{su} - \dot{Q}_{amb} = 0$$  \hspace{1cm} (7)

The equations for the fictitious supply heat exchanger are [18]:

$$\dot{Q}_{su} = C_p(t_{su2} - t_{su1})$$  \hspace{1cm} (8)\hspace{1cm} and

$$\dot{Q}_{su} = \varepsilon_{su} \dot{M} C_p(t_w - t_{su1})$$  \hspace{1cm} (9)

where

$$\varepsilon_{su} = 1 - e^{-AU_{su}/\dot{M} C_p}$$  \hspace{1cm} (10)

A similar set of equations is used for the exhaust heat transfer. Parameters $AU_{su}$ and $AU_{ex}$ may be considered equals for similar refrigerants from the point of view of thermo-physical properties [15,16].

The ambient losses are given by:

$$\dot{Q}_{amb} = AU_{amb}(t_w - t_{amb})$$  \hspace{1cm} (11)

**Clearance volume re-expansion.** The compressor volumetric efficiency is mainly affected by the re-expansion of the fluid trapped in the clearance volume. This effect is very well identified for reciprocating compressors and it can be described by:

$$\frac{\dot{M} v_{su2}^2}{N} \equiv V_c = V_{s} \left( \frac{v_{su2}}{v_{ex2}} - 1 \right)$$ \hspace{1cm} (12)

where

$$V_c = C_t V_s$$  \hspace{1cm} (13)

with $C_t$, the clearance factor.

4.2. **Prediction of the exhaust temperature**

The compression is regarded as isentropic. So the cylinder exhaust temperature can easily be computed by:

$$t_{ex2} = t_{(\text{fluid}; \ p_{ex2}; \ p_{su2}; \ t_{su2}; \ s = s_{su2})}$$ \hspace{1cm} (14)

The mass flow rate and temperature characteristics are thus defined by means of the following parameters: $V_s$, $C_t$, $AU_{su}$, $AU_{ex}$, $AU_{amb}$, $\delta_{su}$ and $\Delta P_{ex}$.

4.3. **Prediction of the compressor shaft power**

Following the ASHRAE Toolkit [13] approach for the calculation of compressor shaft power, we split it into several terms:

- An internal isentropic compression power:
  \[ \dot{W}_{in} = \dot{M} (h_{ex2} - h_{su2}) \]  \hspace{1cm} (15)

- Mechanical losses $\dot{W}_{loss}$ divided in:
  - losses proportional to the isentropic compression power: $\alpha\dot{W}_{in}$,
  - hydrodynamic mechanical losses $\dot{W}_{loss0}(N/N_0)^2$

where $N_0$ is a reference (arbitrary) revolution speed.

The shaft power model is obtained by combining all these terms:

$$\dot{W}_{sh} = \dot{W}_{in} + \dot{W}_{loss}$$ \hspace{1cm} (16)

where

$$\dot{W}_{loss} = \alpha \dot{W}_{in} + \dot{W}_{loss0} \left( \frac{N}{N_0} \right)^2$$ \hspace{1cm} (17)

and $\dot{W}_{loss0}$ and $\alpha$ are positive parameters to be identified.

5. **Results**

The global test results presented in the first part of this paper were used to identify the parameters related to the flow rate characteristic, heat transfers and power losses.

**Identification procedure.** The identification process itself was not considered as mathematically relevant here so it was carried out “manually”, i.e., iteratively considering the result trends at each step on EES software [19]. Default values were used as first guesses, then these values were modified in order to obtain results that fit to the orders of magnitude given in the literature (mainly [17]), for example for the supply and exhaust throttling, for the heating-up, etc. Finally the parameters were modified to obtain a better agreement with the experimental results.

First the swept volume was fixed to the theoretical one and the clearance factor to 5% as guess values. Then the throttling parameters and the $AU$ were tuned in order to give reasonable pressure drops, heating-up and cooling-down. The power losses parameters were fixed to 100 W for the constant losses and 0.2 for $\alpha$. Then the iterative identification process was carried out as described here before until the simulation fits to the experimental results. Fig. 11 illustrates the identification process.

**Results of the identification.** The identification gives the following results:

- $V_s = 710$ cm$^3$
Discussion. The swept volume identified is close to the theoretical one (4% higher). Note that this swept volume is a fictitious one since it takes into account not only the geometrical swept volume but also other processes due to valves dynamics like the suction and discharge back flow. The swept volume parameter depends also on the other parameters value. The clearance factor was identified to 4% which is, one more time, a fictitious value that includes also processes like leakage, back flow and any inaccuracy on the other parameters.

The throttling dimensionless parameter $\delta_{su}$ is of the same order of magnitude as found by Dirlea et al. [20] (they found an average of 0.14 in small car air conditioning compressors). The model leads to pressure drops at the compressor inlet varying from 0.05 bar to 0.40 bar which is in agreement with the orders of magnitude given by Hiller and Glicksmann [17]. A constant exhaust pressure drop was not satisfactory for the precision required. It was necessary to add a pressure drop varying with the refrigerant mass flow rate. Fig. 12 shows the pressures losses obtained by the model.

The different $AU$s were identified as constant for all the test points. A varying $AU$ with the mass flow rate lowers the model accuracy. Fig. 13 gives the computed heating-up and cooling-down obtained by simulation. Comparing it to Fig. 5, it can be seen that the computed cooling down is much more important than the measured one in the manifold. Note that the computed cooling down starts in the cylinder while the measured one is only in the manifold. These variables are quite sensitive on the errors on the other variables as mass flow rate and shaft power, particularly at low load.

Figs. 14, 15 and 16 give the most important variables as simulated by the model compared to the experimental ones. The relative error on the mass flow rate varies between
−6% to 6% except for one point at high mass flow rate, where the relative error increases up to 10% (Fig. 14). The relative error on the shaft power varies between −7% and 3% except for two points at low power where it increases up to 17% (Fig. 15). Nevertheless the highest values correspond to absolute error of the order of 180 W, which is the same order of magnitude as the measurement uncertainty. The absolute error on the exhaust temperature varies between −3 K and +6 K (Fig. 16).

Finally Fig. 17 gives the error on the ambient losses computed compared to the ones obtained by heat balance measurements. It gives in average an error of about 200 W with a peak at about 600 W for the test with the highest error on the shaft power.

6. Conclusions

The detailed experimental analysis carried out permits one to conclude that the main processes affecting the mass flow rate passing through the compressor are the clearance volume re-expansion, the heating-up and the throttling.

A model based on this observation has been developed. It needs seven parameters to compute the mass flow rate and the exhaust temperature: the swept volume $V_s$, the clearance factor $C_f$, two throttling parameters, the different heat transfer coefficients $AU_{su}$, $AU_{ex}$ and $AU_{amb}$ and two parameters to calculate the compressor shaft power: the constant power losses term $\dot{W}_{loss_0}$, and a power coefficient $\alpha$.

The model is able to compute variables of first importance like the mass flow rate, the mechanical power and the exhaust temperature as well as secondary variables as the suction heating-up, the discharge cooling-down, and the ambient losses.

This model is easy to integrate into a global simulation of the compression cycle.

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References


