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# Modelling of reciprocating and scroll compressors

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**Abstract**

This paper presents simple and thermodynamically realistic models of two types of compressors widely used in domestic heat pumps (reciprocating and scroll compressors). These models calculate the mass flow rate of refrigerant and the power consumption from the knowledge of operating conditions and parameters. Some of these parameters may be found in the technical datasheets of compressors whereas others are determined in such a way that the calculated mass flow rate and electrical power match those given in these datasheets.

The two models have been tested on five reciprocating compressors and five scroll compressors. This study has been limited to compressors with a maximum electrical power of 10 kW and for the following operating conditions: evaporating temperatures ranging from −20 to 15 °C and condensing temperatures ranging from 15 to 60 °C.

The average discrepancies on mass flow rate and power for reciprocating compressors are 1.10 and 1.69% (for different refrigerants: R134a, R404A, R22, R12 and R407C). For scroll compressors, the average discrepancies on mass flow rate and power are 2.42 and 1.04% (for different refrigerants: R134a, R404A, R407C and R22).
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*Keywords:* Refrigeration; Air conditioning; Modelling; Performance; Reciprocating piston; Scroll compressor; R134a; R404A; R22; R407C

# Modélisation des compresseurs à piston et à spirale

*Mots clés :* Réfrigération ; Conditionnement d'air ; Modélisation ; Performance ; Compresseur à piston ; Compresseur à spirale ; R134a ; R404A ; R22 ; R407C

## 1. Introduction

Since the beginning of the 1990s, under the pressure of different laws concerning the limitation of $CO_2$ emissions, the heat pumps market has known a new expansion in the domestic sector.

In the year 2000, there were 5750 heat pumps sold in Germany for house heating, 2750 in Finland, 3000 in Norway (for both water and space heating), 23,000 in Sweden, 7200 in Switzerland and 7500 in France [1]. The

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**Nomenclature**

| | |
|---|---|
| $d$ | diameter (m) |
| $h$ | specific enthalpy (J kg$^{-1}$) |
| HP | high pressure (Pa) |
| IP | intermediate pressure (Pa) |
| LP | low pressure (Pa) |
| $m$ | mass (kg) |
| $N$ | compressor rotation speed (t min$^{-1}$) |
| $p$ | pressure (Pa) |
| $P$ | power (W) |
| $p_{vsat}$ | saturated vapour pressure (Pa) |
| $q_m$ | mass flow rate (kg s$^{-1}$) |
| $q_v$ | volume flow rate (m$^3$ s$^{-1}$) |
| $s$ | specific entropy (J kg$^{-1}$ K$^{-1}$) |
| $T$ | temperature (K) |
| $u$ | specific internal energy (J kg$^{-1}$) |
| UA | global heat transfer coefficient (W K$^{-1}$) |
| $v$ | specific volume (m$^3$ kg$^{-1}$) |
| $V$ | volume (m$^3$) |
| $W$ | work (J) |
| $\Delta p_{suc}$ | pressure drop in the suction valve (Pa) |
| $\Delta T_{log}$ | log-mean difference temperature (K) |
| $\Delta T_{sup}$ | superheating (K) |
| $\varepsilon$ | ratio between the dead space and the swept volume (−) |
| $\eta$ | efficiency |
| $\rho$ | density (kg m$^{-3}$) |

*Subscripts*

| | |
|---|---|
| c | circulated |
| calc | calculated |
| cond | condensation |
| constr | constructor data |
| d | dead space |
| el | electrical |
| evap | evaporation |
| ex | exhaust |
| i | inlet of the compressor |
| iso-s | isentropic |
| pseudo-iso-s | pseudo-isentropic |
| mecha | mechanical |
| s | swept |
| suc | suction |
| w | fictitious wall |

average growth in terms of the number of installed heat pumps between 1997 and 2000 is about 15% a year.

It is important to generalize the use of domestic heat pumps in order to decrease the primary energy consumption in the dwelling sector. For each heat pump project, the type of heat pump must be correctly chosen; the design and installation steps must be carried out carefully. For optimization purpose, it would be interesting to dispose a calculation tool able to simulate the behaviour of the heat pump integrated to the residence so that the energy consumption could be predicted and the design of the heat pump (and/or of the house) could be adapted in such a way that the environmental impact is minimized. This simulation tool should remain as simple as possible so that its use could be widespread.

In this aim of modelling such a complete system, it is important to have the simplest and the most accurate models of its components. The compressor is one of the main part of a heat pump as it sets its mass flow rate which governs the heat flows. It is thus the first component of the heat pump to be modelled.

The types of compressors used in domestic heat pumps are usually the reciprocating and scroll ones. Reciprocating compressors are mainly used as far as low thermal power is concerned (heating water) whereas scroll compressors are widespread for space heating.

Many different models of those two types of compressors with different degrees of complexity are found in the literature.

On the one hand, there are models of reciprocating compressors in which the compressor is divided in several volumes (elements such as compression chamber, valves…). Those models require input data very difficult to obtain or known only by the constructor and non-available in the datasheets. The volumes of the different elements and the effective area of valves are also required. The transient fluid conservation equations (continuity, momentum and energy) are integrated in the whole compressor domain and the energy balance for the refrigerant inside the cylinder is computed for each time step during the operating cycle [2–7].

On the other hand, models in which thermodynamic assumptions are made are also found. In those models, data are not very difficult to obtain: e.g. refrigerant inlet state, outlet refrigerant pressure, clearance volume, motor speed. In Ref. [8] eight input data are sufficient to determine mass flow rate and required compressor power. In Refs. [9–11] (model derived from Ref. [8]) the refrigerant mass flow rate is affected by the clearance volume re-expansion, by a pressure drop in the suction valve and by a heat transfer from a fictitious isothermal wall. The friction power loss is composed of a constant contribution and another one proportional to the isentropic power. Ref. [12] presents a simple thermodynamic model for reciprocating compressors used in domestic appliances. It required the knowledge of five parameters easy to determine. The main difference between this model and the one presented in Ref. [9] is the fact that the heat transfer phenomena in the compressor are not considered in Ref. [12].

The same two categories may be defined for scroll compressors.

Models [13–16] require the knowledge of pocket volumes and perimeters for every six degrees of rotation, the height, thickness and pitch of the scrolls that are quite difficult to obtain. In those models, the whole compressor is divided in several chambers and the compression process is simulated for every gas pockets. It requires the evaluation of areas, volumes, pressures, temperatures and specific volumes for every crank angle. Mass and energy conservation equations were developed for each chamber.

Model [17] is exclusively thermodynamic and has the same philosophy as [9–11] used for reciprocating compressors. The refrigerant mass flow rate is affected by a suction temperature increase and the compression process is considered isentropic to the "adapted" pressure and isochoric until the discharge pressure.

The ARI Standard 540 [18] for positive displacement compressors recommends the use of third-degree-equations of 10 coefficients for the calculation of power input, mass flow rate of refrigerant, current or compressor efficiency. Those coefficients have no physical meaning so that the extrapolation of the performances outside the operating range used for the fitting leads to unrealistic performances values.

The purpose of this study is not the development of a complex model that can relate the performances of the compressor to its detailed geometry and that could be used for technological developments. A simple and thermodynamically realistic model is needed. It should be accurate enough to give mass flow rates and power values that can be used in a global model of a heat pump. Parameters appearing in such a model should be found in the technical datasheets of the compressors or should be determined in such a way that the calculated mass flow rate and electrical power match those given in these datasheets.

## 2. Reciprocating compressors

### 2.1. Modelling of reciprocating compressors

The model developed by Lebrun and coworkers [9–11] has been adapted given the particular requirements mentioned above.

The evolution of the thermodynamic state of the refrigerant through the compressor is presented in Fig. 1.

The compression process is divided in three steps.

- Isenthalpic pressure drop in the suction valve (i–1).
- Isobaric heating up in the suction pipe due to a heat transfer with a fictitious wall at temperature $T_w$ (1–2).
- Isentropic compression (2–3).

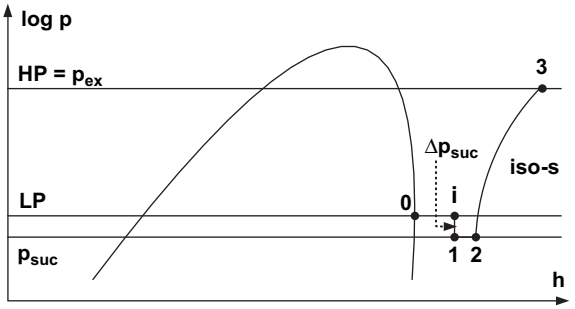


Fig. 1. Diagram (log $p$, $h$) of the compression.

#### 2.1.1. Prediction of the refrigerant mass flow rate

The data are:

- Evaporation temperature $T_{evap}$.
- Condensation temperature $T_{cond}$.
- Temperature at the compressor inlet (point i, Fig. 1) $T_i$ or superheating $\Delta T_{sup}$.

The parameters of the model are:

- The suction line diameter $d_{suc}$.
- The heat transfer coefficient multiplied by the heat transfer surface during the isobaric heating process in the suction line $UA_{suc}$.
- The compressor rotation speed $N$.
- The swept volume $V_s$.
- The ratio between the dead space $V_d$ and swept volumes $\varepsilon$.
- The temperature of the fictitious wall $T_w$.

Both high and low pressures (HP and LP) can be calculated from the phase change temperatures using Ref. [19] (Refprop 7.0®).

$$LP = p_{vsat}(T_{evap}) \quad (1)$$

$$HP = p_{vsat}(T_{cond}) \quad (2)$$

in which $p_{vsat}(T)$ is the saturation vapour pressure law of the refrigerant.

Point 0, corresponding to the saturated vapour at low pressure (Fig. 1), is thus determined.

Temperature $T_i$ at the inlet of the compressor (point i) is either directly given or calculated by Eq. (3).

$$T_i = T_{evap} + \Delta T_{sup} \quad (3)$$

$T_i$ and LP allow the calculation of $h_i$ and $\rho_i$ respectively, the specific enthalpy and density of the refrigerant at the compressor inlet using Ref. [19].

The suction pressure $p_{suc}$ (point 1) is given by Eq. (4).

$$p_{suc} = LP - \Delta p_{suc} \quad (4)$$

in which $\Delta p_{suc}$ is the pressure drop in the suction valve; it can be deduced from Eq. (5).

$$q_m = \frac{\pi d_{suc}^2}{4}\sqrt{2\Delta p_{suc}\rho_i} \tag{5}$$

in which:

- $q_m$ is the mass flow rate of refrigerant.
- $d_{suc}$ is the valve coefficient considered as a pseudo dimension.

The thermodynamic state of the refrigerant at point 1 is determined by Eq. (6) using Ref. [19].

$$h_i(T_i, \mathrm{LP}) = h_1(T_1, p_{suc}) \tag{6}$$

The temperature at the end of the isobaric heating process (transformation 1–2) $T_2$ may be deduced from Eq. (7).

$$UA_{suc}\Delta T_{log\ suc} = q_m(h_2 - h_1) \tag{7}$$

with the log-mean difference temperature defined by Eq. (8).

$$\Delta T_{log\ suc} = \frac{(T_w - T_1) - (T_w - T_2)}{\ln\frac{T_w - T_1}{T_w - T_2}} \tag{8}$$

In Eqs. (5) and (7), the refrigerant mass flow rate is calculated by Eq. (9).

$$q_m = \frac{1}{v_2} q_{vc} \tag{9}$$

in which:

- $v_2$ is the specific volume of refrigerant at point 2.
- $q_{vc}$ is the circulated volume flow rate.

$q_{vc}$, is determined by Eq. (10).

$$q_{vc} = V_c \frac{N}{60} \tag{10}$$

The circulated volume, $V_c$, is the difference between volumes $V_2$ and $V_{3''}$ defined in the Crank diagram (Fig. 2).

$V_2$ is given by Eq. (11):

$$V_2 = V_d + V_s = \varepsilon V_s + V_s \tag{11}$$

The determination of $V_{3''}$ requires the mass of gas ($m_{3''}$) and the specific volume of refrigerant ($v_{3''}$) at point 3″ (Eq. (12)).

$$V_{3''} = v_{3''} m_{3''} \tag{12}$$

Considering the expansion 3′–3″ as isentropic, the specific volume $v_{3''}$ can be calculated knowing the pressure $p_{3''}$ and the specific entropy $s_{3''}$ at point 3″.

$$p_{3''} = p_{suc} \tag{13}$$

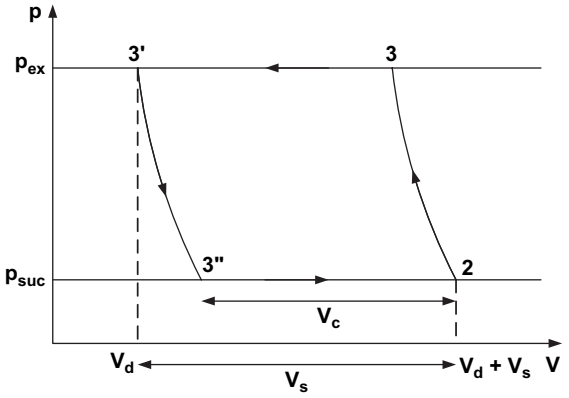


Fig. 2. Crank diagram.

$$s_{3''} = s_{3'} \tag{14}$$

$s_{3'}$ is calculated knowing the pressure $p_{3'}$ and the specific volume $v_{3'}$ at point 3′.

$$p_{3'} = \mathrm{HP} \tag{15}$$

$$v_{3'} = v_3 \tag{16}$$

In the same way, considering the compression step as isentropic, $v_3$ may be calculated from pressure $p_3$ and specific entropy $s_3$ by:

$$p_3 = \mathrm{HP} \tag{17}$$

$$s_3 = s_2 \tag{18}$$

$m_{3''}$ in Eq. (12) is calculated by:

$$m_{3''} = \frac{V_d}{v_{3'}} \tag{19}$$

Assuming that $T_{evap}$, $T_{cond}$ and $T_i$ or $\Delta T_{sup}$ are given, Eqs. (1)–(19) lead to the determination of $q_m$ once the parameters of the model are available.

The parameters are:

- The temperature of the fictitious wall $T_w$.
- The heat transfer coefficient multiplied by the heat transfer surface during the isobaric heating process in the suction line $UA_{suc}$.
- The ratio between dead space and swept volumes $\varepsilon$.
- The diameter of the suction pipe $d_{suc}$.
- The rotation speed of the compressor $N$.
- The swept volume $V_s$.

The last two ones are given by the constructor.

The calculation procedure is given in Fig. 3.

### 2.1.2. Prediction of the electrical power

Once the mass flow rate is calculated, each point in Fig. 1 is known.

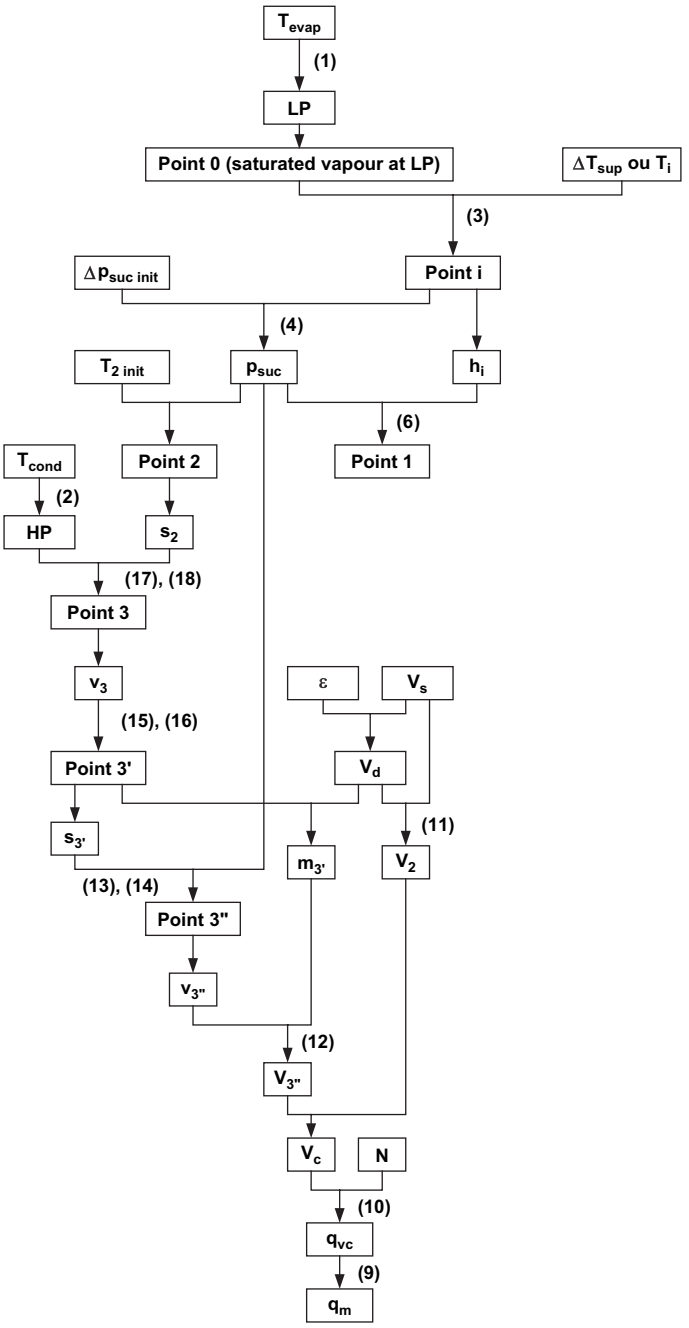


Fig. 3. Mass flow rate calculation.

In the previous section, compression (2–3) was supposed to be isentropic. The mechanical power is then written (Eq. (20)).

$$P_{\text{mecha}} = q_{\text{m}}(h_3 - h_2) \tag{20}$$

The calculation of the actual electric power consumption provided to the compressor requires the knowledge of the factor ($\eta_{\text{el}}\eta_{\text{iso-s}}$), which is the product of the electrical and isentropic efficiencies.

This factor allows us to take into account both the electrical losses and the losses due to the non-isentropic compression.

This term was considered to be a polynomial function of the compression ratio (HP/LP) (Eq. (21)).

$$\eta_{el}\eta_{iso\text{-}s} = a\left(\frac{HP}{LP}\right)^6 + b\left(\frac{HP}{LP}\right)^5 + c\left(\frac{HP}{LP}\right)^4 + d\left(\frac{HP}{LP}\right)^3 + e\left(\frac{HP}{LP}\right)^2 + f\left(\frac{HP}{LP}\right) + g \quad (21)$$

Knowing the coefficients of the polynomial function it is possible to calculate the electrical power (Eq. (22)).

$$P_{elcalc} = \frac{P_{mecha}}{\eta_{iso\text{-}s}\eta_{el}} \quad (22)$$

The prediction of this power consumption requires new parameters ($a$, $b$, $c$, $d$, $e$, $f$, $g$) but no new operating data: HP and LP were needed for the mass flow rate calculation.

### 2.2. Determination of the parameters

#### 2.2.1. Determination of the parameters: mass flow rate calculation

The four unknown parameters required for the calculation of the mass flow rate are: $T_w$, $UA_{suc}$, $\varepsilon$ and $d_{suc}$. The assumption that they are constant for a given compressor and a given refrigerant is made.

Their values are found by minimizing the discrepancies between the calculated mass flow rates and the ones provided in the datasheets for given operating conditions (LP, HP, $T_i$, $\Delta T_{sup}$).

The fictitious wall temperature, $T_w$, has not a significant influence on the mass flow rate so its value was set to 50 °C.

#### 2.2.2. Determination of parameters: electrical power calculation

The parameters needed for the determination of the electrical power are the coefficients of the polynomial equation linking the compression ratio HP/LP to the product of efficiencies ($\eta_{el}\eta_{iso\text{-}s}$). First the isentropic mechanical power is calculated (Eq. (20)). The ratio $P_{mecha}/P_{electrical}$ is then deduced from the electrical power provided by the manufacturer in the same operating conditions.

This ratio is the product of the electrical and isentropic efficiencies ($\eta_{el}\eta_{iso\text{-}s}$) (Eq. (23)).

$$\frac{P_{mecha}}{P_{electrical}} = \eta_{iso\text{-}s}\eta_{el} \quad (23)$$

A sixth degree polynomial law is used to correlate $\eta_{el}\eta_{iso\text{-}s}$ to the compression ratio. The values of the parameters are determined by an optimization procedure.

## 3. Scroll compressors

### 3.1. Modelling of scroll compressors

The model is an adaptation of the work by Lebrun et al. [17].

The evolution of the thermodynamic state of the refrigerant through the compressor is presented in Fig. 4.

The compression process is divided in three steps.

- Isobaric heating up in the suction pipe due to a heat transfer with a fictitious wall at temperature $T_w$ (1–2).
- First part of the compression (assumed isentropic, 2–3): the gas is compressed until the volume created by the scrolls matches exhaust volume, $V_{ex}$. The pressure at point 3 is called "intermediate pressure, IP"; it can be lower or greater than the high pressure.
- End of the compression at $V_{ex}$ (constant volume) by mass accumulation in the exhaust chamber until the pressure is equal to the exhaust one (3–4).

#### 3.1.1. Prediction of the refrigerant mass flow rate

The data are:

- Evaporation temperature $T_{evap}$.
- Condensation temperature $T_{cond}$.
- Temperature at the compressor inlet (point 1, Fig. 4) $T_1$ or superheating $\Delta T_{sup}$.

The parameters of the model are:

- The temperature of the fictitious wall $T_w$.
- The heat transfer coefficient multiplied by the heat transfer surface during the isobaric heating process in the suction line $UA_{suc}$.
- The swept volume $V_s$.
- The compressor rotation speed $N$.

Both high and low pressures (HP and LP) can be calculated from the phase change temperatures using Ref. [19].

$$LP = p_{vsat}(T_{evap}) \quad (24)$$

$$HP = p_{vsat}(T_{cond}) \quad (25)$$

in which $p_{vsat}(T)$ is the saturation vapour pressure law of the refrigerant.

Point 0, corresponding to the saturated vapour at low pressure (see Fig. 4), is thus determined.

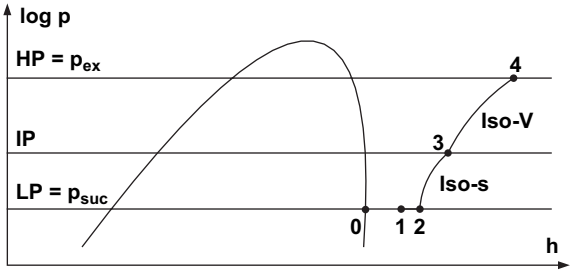


Fig. 4. Diagram (log $p$, $h$) of the thermodynamic process.

Temperature $T_1$ at the inlet of the compressor (point 1) is either directly given or calculated by Eq. (26).

$$T_1 = T_{evap} + \Delta T_{sup} \tag{26}$$

$T_1$ and LP allow the calculation of $h_1$, specific enthalpy of the refrigerant at the compressor inlet using Ref. [19].

The temperature at the end of the isobaric heating process (transformation 1–2) $T_2$ may be deduced from Eq. (27).

$$UA_{suc}\Delta T_{log\ suc} = q_m(h_2 - h_1) \tag{27}$$

in which $q_m$ is the mass flow rate of refrigerant.

With the log-mean difference temperature defined by Eq. (28).

$$\Delta T_{log\ suc} = \frac{(T_w - T_1) - (T_w - T_2)}{\ln \frac{T_w - T_1}{T_w - T_2}} \tag{28}$$

In Eq. (27), the refrigerant mass flow rate is calculated by Eq. (29).

$$q_m = \frac{1}{v_2} V_s N \tag{29}$$

in which $v_2$ is the specific volume of refrigerant at point 2.

Assuming $T_{evap}$, $T_{cond}$ and $T_1$ or $\Delta T_{sup}$ are given, Eqs. (24)–(29) lead to the determination of $q_m$ once the parameters of the model are available.

The parameters are:

- The temperature of the fictitious wall $T_w$.
- The heat transfer coefficient multiplied by the heat transfer surface during the isobaric heating process in the suction line $UA_{suc}$.
- The swept volume $V_s$.
- The rotation speed of the compressor $N$.

The last two ones are given by the constructor.

The calculation procedure is given in Fig. 5.

### 3.1.2. Prediction of the electrical power

The compression 2–3 is supposed to be isentropic. It occurs between the low pressure and the intermediate pressure for which the exhaust volume is obtained. This intermediate

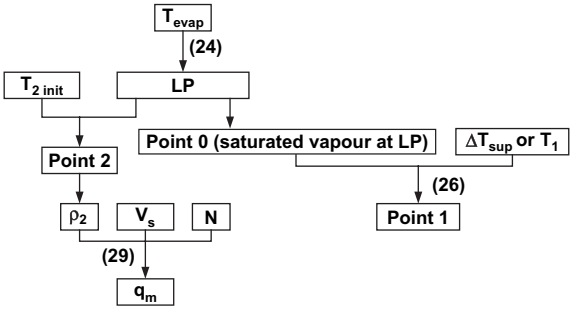


Fig. 5. Mass flow rate calculation.

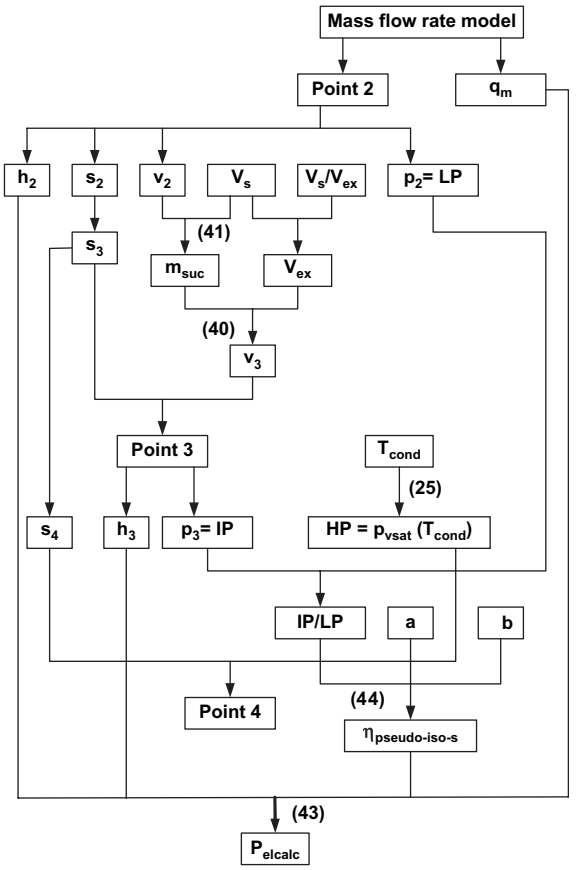


Fig. 6. Determination of power consumption, diagram.

pressure, IP, can be either higher or lower than the high pressure.

The mechanical work $W_{23}$ corresponding to this compression step is given by Eq. (30).

$$W_{23} = m_{suc}(u_3 - u_2) \tag{30}$$

in which:

- $u$ is the specific internal energy of the gas.
- $m_{suc}$ is the suctioned mass.

Table 1
Characteristics of the studied reciprocating compressors

| Compressor | Volumic flow rate ($m^3\,h^{-1}$) | Number of cylinders | Bore (mm) | Stroke (mm) | $P$ (W) (R134a, $T_{cond} = 40\,^{\circ}C$, $T_{evap} = 0\,^{\circ}C$) |
|---|---|---|---|---|---|
| R1 | 39.36 | 4 | 60 | 40 | 5080 |
| R2 | 73.6 | 4 | 70 | 55 | 9460 |
| R3 | 55.99 | 4 | 63.5 | 50.8 | 7600 |
| R4 | 70.9 | 4 | 68.3 | 55.6 | 9150 |
| R5 | 14.9 | 2 | Not available | Not available | 2162 |

Table 2
Reciprocating compressors: results

| Compressor | Fluid | $UA_{suc}$ (W K$^{-1}$) | $d_{suc}$ (cm) | $\varepsilon$ (%) | Discrepancy $q_m$ (%) | Discrepancy $P$ (%) |
|---|---|---|---|---|---|---|
| Compressor R1: $V_s = 452.414$ cm$^3$, $N = 1450$ t min$^{-1}$ | R134a | 43.53 | 2.918 | 2.50 | 1.04 | 3.45 |
| | R404A | 48.46 | 1.772 | 2.36 | 0.99 | 2.37 |
| | R22 | 48.91 | 2.275 | 4.24 | 1.02 | 2.71 |
| | R12 | 50 | 2.457 | 4.22 | 1.11 | 1.66 |
| Compressor R2: $V_s = 845.977$ cm$^3$, $N = 1450$ t min$^{-1}$ | R134a | 15.66 | 2.781 | 3.41 | 0.53 | 1.30 |
| | R404A | 19.01 | 2.043 | 2.97 | 0.34 | 1.16 |
| | R22 | 19.77 | 2.405 | 4.28 | 0.50 | 2.30 |
| | R12 | 20.35 | 2.567 | 4.18 | 1.42 | 0.53 |
| Compressor R3: $V_s = 643.563$ cm$^3$, $N = 1450$ t min$^{-1}$ | R134a | 67.93 | 2.610 | 4.73 | 0.80 | 1.98 |
| | R404A | 40.09 | 2.105 | 4.59 | 0.75 | 1.45 |
| | R22 | 5.35 | 1.962 | 5.91 | 0.37 | 2.24 |
| Compressor R4: $V_s = 814.943$ cm$^3$, $N = 1450$ t min$^{-1}$ | R134a | 104.39 | 3.417 | 5.11 | 1.22 | 1.24 |
| | R404A | 95.44 | 2.678 | 4.79 | 1.09 | 0.86 |
| | R22 | 43.97 | 2.432 | 5.88 | 0.39 | 0.82 |
| Compressor R5: $V_s = 85.64$ cm$^3$, $N = 2900$ t min$^{-1}$ | R134a | 13.77 | 1.221 | 8.25 | 1.86 | 2.79 |
| | R404A | 19.38 | 1.017 | 6.76 | 3.06 | 1.65 |
| | R407C | 22.18 | 1.132 | 8.55 | 1.59 | 1.23 |

The mechanical work during suction, $W_{suc}$, is given by Eq. (31).

$$W_{suc} = -V_{suc}\mathrm{LP} \tag{31}$$

The work at the exhaust, $W_{ex}$, is given by Eq. (32).

$$W_{ex} = V_{ex}\mathrm{HP} \tag{32}$$

The total work between points 2 and 4 is obtained by the summation of the three works (relations (30)–(32)).

Volumes $V_{suc}$ and $V_{ex}$ can also be written as Eqs. (33) and (34).

$$V_{suc} = m_{suc}v_2 \tag{33}$$

$$V_{ex} = m_{suc}v_3 \tag{34}$$

which leads to Eq. (35).

$$W = m_{suc}(u_3 - u_2) - \mathrm{LP}m_{suc}v_2 + \mathrm{HP}m_{suc}v_3 \tag{35}$$

given

$$u_2 = h_2 - \mathrm{LP}v_2 \tag{36}$$

$$u_3 = h_3 - \mathrm{IP}v_3 \tag{37}$$

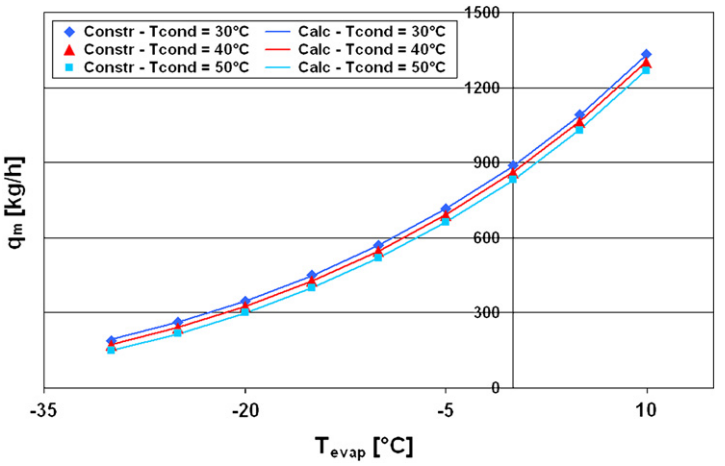


Fig. 7. Compressor R2, R134a – $q_m = f(T_{evap})$.

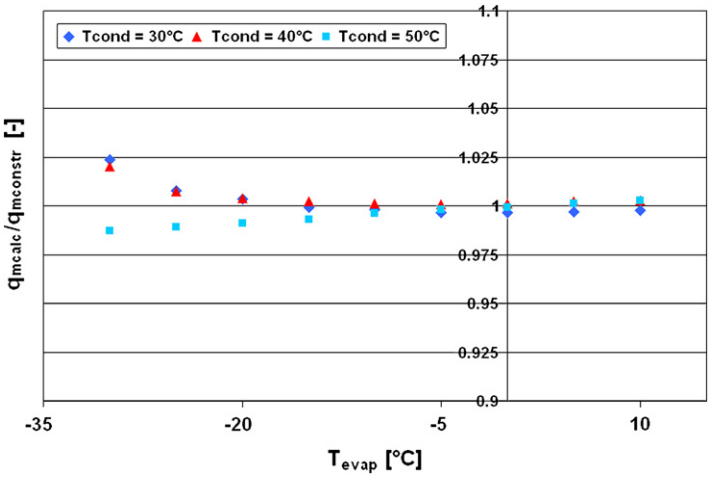


Fig. 8. Compressor R2, R134a – $q_{mcalc}/q_{mconstr} = f(T_{evap})$.

Eq. (35) becomes Eq. (38).

$$W = m_{suc}(h_3 - h_2) + (HP - IP)m_{suc}v_3 \quad (38)$$

which leads to the expression of the power (Eq. (39)).

$$P = q_m(h_3 - h_2) + (HP - IP)q_m v_3 \quad (39)$$

In Eq. (39) $q_m$ is determined by the procedure described in Section 3.1.1; $h_2$ is calculated by Ref. [19] knowing the temperature and pressure conditions at point 2; the high pressure HP is known via relation (24).

The specific volume $v_3$ is calculated by:

$$v_3 = \frac{V_{ex}}{m_{suc}} \quad (40)$$

in which:

- The exhaust volume $V_{ex}$ is determined from the suction volume (given by the constructor) and from the ratio between this volume and the exhaust volume ($V_{suc}/V_{ex}$) which is a parameter of the model.
- The suctioned mass $m_{suc}$ is calculated by:

$$m_{suc} = \frac{V_{suc}}{v_2} \quad (41)$$

with $v_2$ calculated by Ref. [19].

In Eq. (39), point 3 ($h_3$ and IP) is known because both the specific volume $v_3$ and the specific entropy $s_3$ are known (Eqs. (40) and (42)).

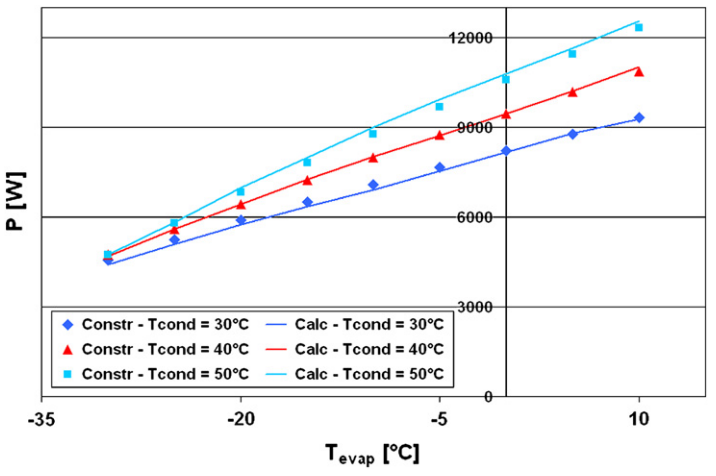


Fig. 9. Compressor R2, R134a – $P = f(T_{evap})$.

$$s_3 = s_2 \tag{42}$$

Knowing each term in relation (39), it is possible to calculate $P$ which is the thermodynamic mechanical power (with the first step considered as isentropic). The calculation of the electrical power $P_{elcalc}$ would require the introduction of an isentropic efficiency in the first term of the right-hand member of Eq. (39) as well as an electrical efficiency applied to each step of the compression. Due to the fact that these two efficiencies are not applied to the same terms it is not possible to gather them in one term ($\eta_{el}\eta_{iso\text{-}s}$) as it was the case for reciprocating compressors. It appeared that the introduction of two new parameters to be dealt with in the optimization procedure led to unrealistic parameter values and to too good performances compared to the experimental accuracy of the data. As a consequence, given the low influence of the second term of the right-hand term of Eq. (39), we only introduced one parameter (applied to the isentropic step) called the pseudo-isentropic efficiency and which takes into account both the irreversibilities and the electrical losses.

$$P_{elcalc} = \frac{q_m(h_3 - h_2)}{\eta_{pseudo\text{-}iso\text{-}s}} + (HP - IP) q_m v_{3real} \tag{43}$$

For a correct representation of the electrical power, it appeared that this pseudo-isentropic efficiency should be considered as a linear function of the ratio IP/LP. $v_{3real}$ is the specific volume at the "real" point 3 (non-isentropic).

The power calculation procedure is presented in Fig. 6.

### 3.2. Determination of the parameters

#### 3.2.1. Determination of the parameters: mass flow rate calculation

The two unknown parameters in the mass flow rate prediction are $T_w$ and $UA_{suc}$. The assumption that they are constant for a given compressor and a given refrigerant is made. Their values are found by minimizing the discrepancies between the calculated mass flow rates and the ones provided in the datasheets for given operating conditions.

The fictitious wall temperature, $T_w$, has no significant influence on the mass flow rate so that its value was set to 50 °C.

#### 3.2.2. Determination of the parameters: electrical power calculation

The influent parameters in the power prediction are the ratio between the suction and the exhaust volumes ($V_{suc}/V_{ex}$) and the coefficients of the linear law of the pseudo-isentropic efficiency ($\eta_{pseudo\text{-}iso\text{-}s}$) versus the compression ratio IP/LP.

The isentropic efficiency is expressed by Eq. (44).

$$\eta_{pseudo\text{-}iso\text{-}s} = a\frac{IP}{LP} + b \tag{44}$$

These three parameters are determined by minimizing the discrepancies between the calculated electrical power and the ones provided in the datasheets for given operating conditions.

## 4. Results

In this section, simulation results of mass flow rate and power for both types of compressors are presented.

### 4.1. Reciprocating compressors

Five reciprocating compressors were studied. Some of their characteristics are presented in Table 1 [20–22].

Table 2 presents values of the different parameters ($UA_{suc}$, $d_{suc}$ and $\varepsilon$) and the average discrepancies, respectively, on the mass flow rate and on the electrical power for each compressor and each studied fluid.

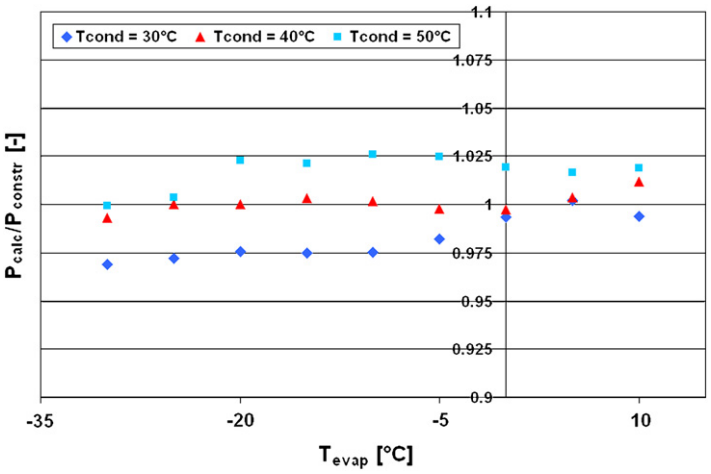


Fig. 10. Compressor R2, R134a – $P_{calc}/P_{constr} = f(T_{evap})$.

Table 3
Characteristics of the studied scroll compressors

| Compressor | Volumic flow rate ($m^3 h^{-1}$) | $P$ (W) (R134a, $T_{cond} = 40\ °C$, $T_{evap} = 0\ °C$) |
|---|---|---|
| S1 | 25 | 3770 |
| S2 | 36 | 5420 |
| S3 | 8.6 | 1340 |
| S4 | 35.56 | 5500 |
| S5 | 43.5 | 6866 |

The model is very good at predicting mass flow rate and power consumption of reciprocating compressors: average discrepancies are lower than 3%. In most cases, the points for which the discrepancy is higher than 5% correspond to evaporation temperatures lower than −20 °C and/or condensation temperature higher than 60 °C. In residential heat pumps, those ranges of temperatures are rare.

Figs. 7 and 8 present, respectively, the evolution of the mass flow rate (calculated and given by the manufacturer) and the related discrepancy as a function of the evaporation temperature for different condensation temperatures (30, 40 and 50 °C).

For evaporation temperatures higher than −20 °C the discrepancy on the mass flow rate is lower than 1%.

Figs. 9 and 10 present, respectively, the evolution of the electrical power (calculated and given by the manufacturer) and the related discrepancy as a function of the evaporation temperature for different condensation temperatures (30, 40 and 50 °C).

For evaporation temperatures higher than −20 °C the discrepancy on the mass flow rate is lower than 2.5%.

### 4.2. Scroll compressors

Five reciprocating compressors were studied. Some of their characteristics are presented in Table 3 [20–22].

Table 4 presents the values of the different parameters ($UA_{suc}$, $V_{suc}/V_{ex}$, $a$ and $b$) and the average discrepancies, respectively, on the mass flow rates and on the electrical powers for each compressor and each fluid.

The model is very good at predicting mass flow rate and power of scroll compressors. The mean discrepancies are lower than 3.5%. In most cases, points for which discrepancy is higher than 5% correspond to evaporation temperatures lower than −20 °C and/or condensation temperature higher than 60 °C. In the residential heat pumps, those ranges of temperatures are rare.

Figs. 11 and 12 present, respectively, the evolution of mass flow rate and the discrepancy on this mass flow rate as a function of the evaporation temperature for different condensation temperatures.

The discrepancy on the mass flow rate is generally lower than 5%. It can reach nearly 10% for high condensation temperatures (55 and 60 °C).

Figs. 13 and 14 present, respectively, the evolution of power and the discrepancy on this power as a function of the evaporation temperature for different condensation temperatures.

Table 4
Scroll compressors: results

| Compressor | Fluid | $UA_{suc}$ ($W K^{-1}$) | $V_{suc}/V_{ex}$ (−) | $a$ (−) | $b$ (−) | Discrepancy $q_m$ (%) | Discrepancy $P$ (%) |
|---|---|---|---|---|---|---|---|
| Compressor S1: $V_s = 143.678\ cm^3$, $N = 2900\ t\ min^{-1}$ | R134a | 19.01 | 2.379 | −0.777 | 2.585 | 1.70 | 1.12 |
| | R404A | 48.36 | 2.165 | 0.010 | 0.682 | 1.57 | 0.19 |
| | R407C | 42.06 | 2.219 | −0.312 | 1.463 | 2.31 | 0.46 |
| | R22 | 60.33 | 2.187 | −0.671 | 2.329 | 2.21 | 0.59 |
| Compressor S2: $V_s = 206.897\ cm^3$, $N = 2900\ t\ min^{-1}$ | R134a | 26.99 | 2.378 | −0.772 | 2.570 | 1.74 | 1.12 |
| | R404A | 69.59 | 2.166 | 0.003 | 0.698 | 1.59 | 0.16 |
| | R407C | 59.49 | 2.220 | −0.316 | 1.473 | 2.31 | 0.46 |
| | R22 | 87.09 | 2.189 | −0.677 | 2.345 | 2.18 | 0.59 |
| Compressor S3: $V_s = 47.778\ cm^3$, $N = 3000\ t\ min^{-1}$ | R134a | 5.39 | 2.302 | −0.473 | 1.766 | 1.47 | 0.85 |
| | R404A | 18.97 | 2.089 | −0.320 | 1.361 | 2.95 | 1.02 |
| | R22 | 22.67 | 2.128 | −0.642 | 2.142 | 3.15 | 2.26 |
| | R407C | 17.44 | 2.156 | −1.152 | 3.302 | 1.97 | 1.21 |
| Compressor S4: $V_s = 197.556\ cm^3$, $N = 3000\ t\ min^{-1}$ | R134a | 107.02 | 2.637 | −0.134 | 1.022 | 2.32 | 0.53 |
| | R404A | 44.93 | 2.667 | −0.113 | 0.954 | 2.95 | 1.49 |
| | R22 | 131.05 | 2.937 | −0.215 | 1.422 | 2.78 | 1.24 |
| | R407C | 146.36 | 2.799 | −0.194 | 1.273 | 2.95 | 1.08 |
| Compressor S5: $V_s = 24.99\ cm^3$, $N = 2900\ t\ min^{-1}$ | R134a | 34.64 | 2.466 | −0.826 | 2.752 | 2.02 | 0.66 |
| | R22 | 62.43 | 2.293 | −0.579 | 2.148 | 1.84 | 0.58 |
| | R407C | 99.14 | 2.239 | −0.465 | 1.787 | 2.23 | 0.55 |

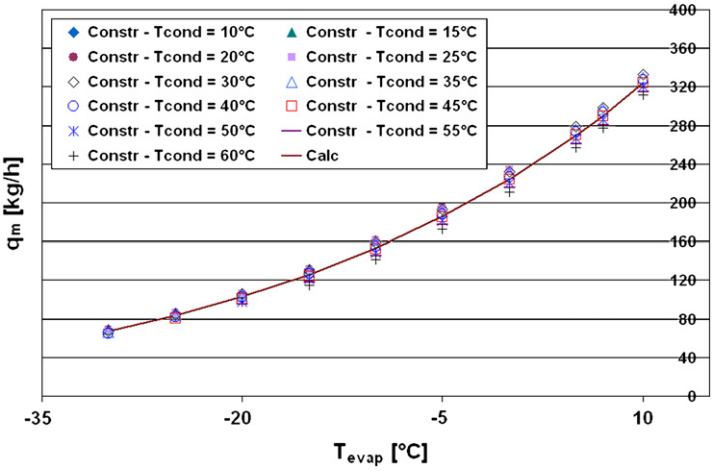


Fig. 11. Compressor S3, R404A – $q_m = f(T_{evap})$.

The discrepancy on the power is also generally lower than 5%. It can be higher than 5% for low condensation temperatures (10 and 15 °C).

## 5. Conclusion

In this paper, simple and thermodynamically realistic models of the two types of compressors found in the residential heat pumps (reciprocating and scroll compressors) have been presented. Those models calculate the mass flow rate of refrigerant and the power consumption from the knowledge of operating conditions (phase change temperatures and refrigerant temperature at the inlet of the compressor or superheating) and different parameters.

Parameters appearing in the models are found in the technical datasheets of the compressors or they are fitted in such a way that the calculated mass flow rate and electrical power match those given in these datasheets.

The reciprocating compressor model requires the knowledge of six parameters. Two of them are given in the datasheets: the swept volume ($V_s$) and the compressor rotation speed ($N$). The last four are: the temperature of the fictitious wall ($T_w$) which is actually set to a constant value (50 °C), the global heat transfer coefficient at suction ($UA_{suc}$), the diameter of the suction pipe ($d_{suc}$) and the ratio between the dead space and the swept volume ($\varepsilon$). Those three parameters have to be fitted in order that the calculated mass flow rates match the ones given in the datasheets. The determination of the powers requires the knowledge of the seven coefficients of the polynomial law relating the product of the electrical and isentropic efficiencies to the compression ratio.

This model has been tested on five compressors. The mean discrepancies values on the determination of, respectively, mass flow rate and power consumption are 1.15 and 2.04% (R134a), 1.38 and 1.43% (R404A),

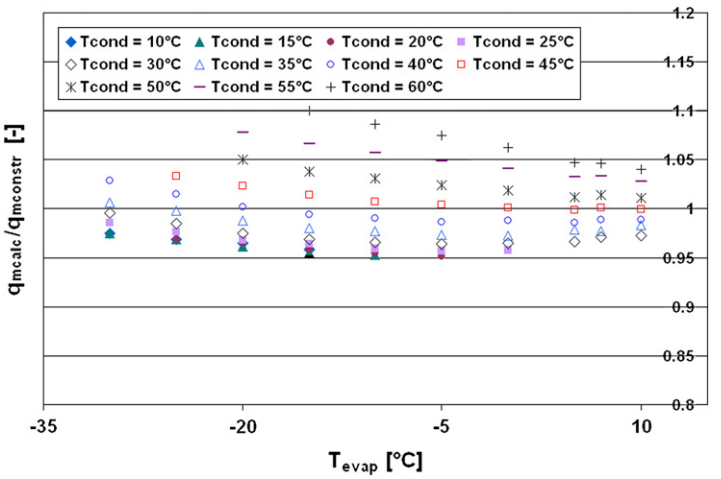


Fig. 12. Compressor S3, R404A – $q_{mcalc}/q_{mconstr} = f(T_{evap})$.

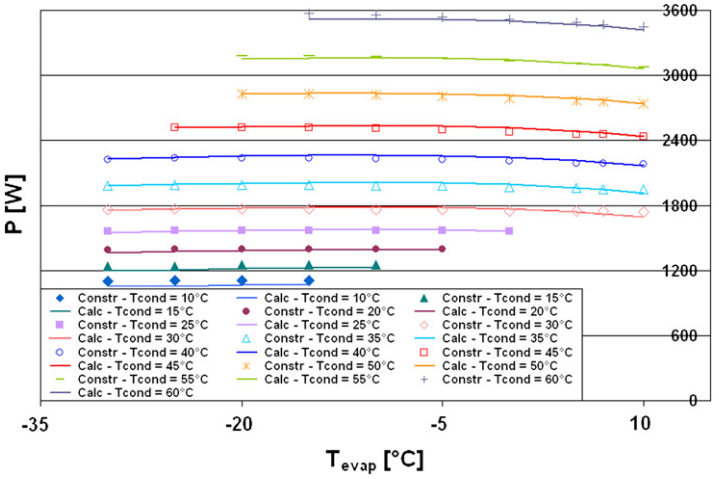


Fig. 13. Compressor S3, R404A – $P = f(T_{evap})$.

0.51 and 1.86% (R22), 1.27 and 1.09% (R12), 1.59 and 1.23% (R407C).

The scroll compressor model requires the knowledge of seven parameters. Two of them are given in the datasheets: the swept volume ($V_s$) and the compressor rotation speed ($N$). The last five are: the temperature of the fictitious wall ($T_w$) which is actually set to a constant value (50 °C), the global heat transfer coefficient at suction ($UA_{suc}$), the ratio between the swept volume and the exhaust volume ($V_{suc}/V_{ex}$) and the two parameters $a$ and $b$ of the linear law of the pseudo-isentropic efficiency. Those four parameters have to be fitted in order that the calculated mass flow rate and power matches those given in the datasheets. $UA_{suc}$ is fitted using the mass flow rate data while the three other parameters ($V_{suc}/V_{ex}$, a, and $b$) are fitted using power consumption data.

This model has been tested on five compressors. The mean discrepancy values on mass flow rate and power are 1.89 and 0.76% (R134a), 2.76 and 1.12% (R404A), 2.39 and 0.91% (R407C), 2.59 and 1.31% (R22).

Although, it was not hinted in this paper those models fail at predicting correct outlet temperatures (up to 50 °C difference depending on the compressor and fluid, usually the difference is around 20–25 °C). So the same philosophy at the outlet than at the inlet (taking into account a heat transfer with the fictitious wall) should be applied. The determination of the global heat transfer coefficient at the exhaust ($UA_{ex}$) requires values of exhaust temperature or values of heat flow rate at the condenser. As temperature $T_w$ has a significant influence on the outlet temperature of the compressor, it seems interesting to make it vary with the phase change temperatures and power consumption of the compressor.

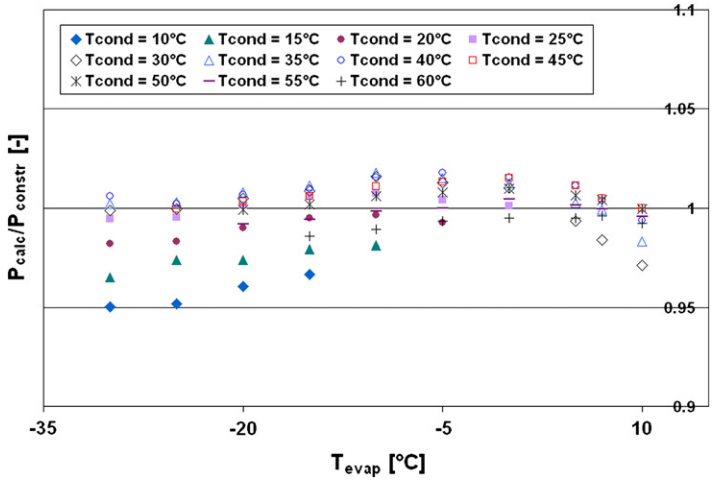


Fig. 14. Compressor S3, R404A – $P_{calc}/P_{constr} = f(T_{evap})$.

It is also possible to determine one set of parameters for all the refrigerants and for a given compressor.

It must also be noticed that the isentropic and electrical efficiencies that are used for the determination of the power have no real physical sense (the use of parameters with physical sense leads to an over-parameterised model). This lack of physical sense can explain the higher discrepancies at high temperatures.

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