Analysis of cycloid type vacuum compressors for water vapor compression systems at sub-atmospheric pressures

T W Moesch, T Janitschke, C Thomas and U Hesse
Bitzer Chair for Refrigeration, Cryogenics and Compressor Technology, Institute for Energy Technologies, Technische Universität Dresden, 01062 Dresden

thomas.moesch@tu-dresden.de

Abstract. Current refrigerant regulations lead to a transition towards low GWP refrigerants and natural refrigerants. Water vapor (R718) has been introduced in different vapor compression chiller and heat pump systems. Current R718 systems are based on turbo type compressors due to the resulting sub-atmospheric operating pressure and the low volumetric cooling capacity. This leads to a rather limited application range due to the pressure ratio limitations of turbo compressors. The application range of R718 may be extended towards heat pump operation by using positive displacement compressors such as state of the art vacuum compressors. This paper analyses the application of an oil free cycloid type variable pitch screw vacuum compressor for a heat pump at 2...15 °C evaporation temperature and 25...85 °C condensing temperature. A chamber model was used for the compressor analysis. During the analysis the compressor geometry, its speed and leakage gap sizes were varied. The results include the compressor isentropic efficiency, its volumetric efficiency and the resulting discharge temperature.

1. Introduction
The currently defined quotas of fluorinated refrigerants in European regulations [1] lead to an increase in refrigerant prices and shift the focus for refrigeration and air condition systems towards low GWP and natural refrigerants. The comparative investigation of Kilicarslan and Müller [2] shows the efficiency benefits of water (R718) compared to other refrigerants at evaporation temperatures above 35°C and for lower evaporation temperatures with a temperature lift of ΔT≤10K and ΔT≥30K. Bamigbetan et al. [4] show benefits of R718 for high temperature heat pumps in industrial processes. The application of R718 in chillers or heat pumps with low temperature heat sources is challenging due to sub-atmospheric pressures, high volume flow rates and increased pressure ratios as well as high discharge temperatures. Recent investigations on R718 vapor compression chillers ([5], [6]) and heat pumps ([7], [8]) utilize centrifugal compressors. However, one stage of this compressor type is limited to a pressure ratio of π ≤ 2.45 for applications with R718 at evaporation temperatures of θ₀ = 5 °C assuming typical radial centrifugal compressor design parameters (Maᵣ = 1, ψ = 1.5, φ = 0.1). This reduces the operation envelope to temperature lifts of ΔT ≤ 13.5 K for single stage compression cycles and thus limits possible high temperature lift heat pump applications.

The authors therefor suggest an alternative approach as suggested by Chamoun et al. [9] by using a screw compressor or more specifically a variable pitch type screw compressor which is commonly applied in vacuum technology ([3], [10]). This paper describes the developed compressor chamber.
model and presents the results for the performance prediction for a given compressor geometry and its operating envelope.

2. Compressor chamber model
The investigated screw compressor is a double shaft oil free vacuum compressor. It has a symmetric cycloid type rotor profile and a multi-pitch helix as shown in Figure 1. The profiles of each rotor enclose a gap volume with the surrounding cylinder $V_{\text{chamber}}$. The compressor was modelled in a quasi-stationary multi-chamber approach assuming a system of several compressor chambers in serial. The approach is based on the model of Rohe [12] and a simplified model schematic is shown in Figure 2.

![Screw compressor with a cycloid type rotor profile (left) and multi-pitch helix (right)](image)

**Figure 1:** Screw compressor with a cycloid type rotor profile (left) and multi-pitch helix (right)

The volume of each chamber changes periodically and each chamber transfers fluid mass to adjacent chambers by turning into the consecutive chamber as well as through leakage passages $\Delta L_m$. The heat $\Delta Q$ is transferred to and from the compressor housing which is actively cooled. The model uses the energy balance in equation (1) and the mass balance in equation (2) for each chamber at the time step $t_{i+1} = t_i + \Delta t$.

$$U_{i+1} = U_i + \Delta W_{n,i} + \Delta Q_i + \sum_k (\dot{m}_{L,i,k} \cdot \Delta t \cdot h_{i,k})$$  \hspace{1cm} (1)
\[ m_{i+1} = m_i + \sum_k (\dot{m}_{L,ik} \cdot \Delta t) \]  
\[ (2) \]

The isentropic compressor work \( \Delta W_{is} \) is defined by equation (3) which is based on ideal gas behaviour.

\[ \Delta W_{is} = \frac{k}{k-1} \cdot \rho_i \cdot V_i \left( \frac{V_i}{V_{i+1}} \right)^{k-1} - 1 \]  
\[ (3) \]

Due to the periodicity of the chamber volumes, each chamber is rotated by 180° until it turns into its subsequent chamber. The state of the chamber is evaluated for each rotation step. The chamber state variables \( T, p, \) and \( v \) are calculated using equations (4) through (6) for each rotation step.

\[ T_{i+1} = \frac{u_{i+1} - u_{ref}}{c_v(T_i)} + T_{ref} \]  
\[ (4) \]
\[ p_{i+1} = \frac{m_{i+1} \cdot R \cdot T_{i+1}}{V_{i+1}} \]  
\[ (5) \]
\[ v_{i+1} = \frac{V_{i+1}}{m_{i+1}} \]  
\[ (6) \]

The state of each chamber at its last rotation is used as an inlet state for the subsequent chamber. During the chamber states iteration the discharge temperature of the compressor is reassumed for each iteration cycle. The stop criterion is defined by the differences in the energy and mass balances. Furthermore an ideal flow at the inlet and outlet of the compressor is assumed and therefore pressure losses are neglected.

The detailed modelling of the screw compressor geometry, the occurring leakage flows, the heat transfer and the performance parameters is presented in the following subsections.

2.1. Compressor geometry
The symmetric two teeth profile of the screw rotors and the corresponding geometric parameters are shown in Figure 1. The measured geometrical profile parameters are listed in Table 1.

| Parameter               | Symbol | Value | Unit |
|-------------------------|--------|-------|------|
| crown circle diameter   | \( d_C \) | 0.116 | m    |
| root circle diameter    | \( d_R \) | 0.085 | m    |
| crown angle             | \( \Delta \tau_C \) | 73.5  | °     |
| wrap-angle              | \( \tau_W \) | 1800  | °     |
| number of teeth         | \( N_Z \) | 2     | -    |
| rotor length            | \( L \) | 0.210 | m    |

The enclosed gap volume of each chamber results from a numerical integration of the cross sectional chamber area as defined in equation (7).

\[ V(\tau) = \int_{\tau_{i+1}}^{\tau_{i}} A(z)dz = \int_{\tau_{i+1}}^{\tau_i} A(\tau) \cdot \frac{dz(\tau)}{d\tau} d\tau \]  
\[ (7) \]
Where $A(\tau)$ is the cross sectional chamber area, $z$ is the position on the $z$-axis, and $\tau$ is the angle of rotation.

The cross sectional chamber area $A(\tau)$ is determined in a geometrical approach by evaluating the enclosed polygon for each discrete rotational angle step $\Delta \tau$. The rotor helix and the measured helix data is shown in Figure 3 and consists of three parts: two with a constant pitch and one with a variable pitch which was found to be linear. The $z$-position of the helix for the constant pitch and variable pitch is defined by equations (8) through (10).

$$h(\tau) = 2\pi \cdot \frac{dz(\tau)}{d\tau}$$  \hspace{1cm} (8)

$$z_{\text{conv}}(\tau) = h \cdot \frac{\tau - \tau_0}{2\pi} + z_0$$  \hspace{1cm} (9)

$$z_{\text{var}}(\tau) = h_0 \left[ \frac{\tau - \tau_0}{2\pi} - \lambda \cdot \left( \frac{\tau - \tau_0}{2\pi} \right)^2 \right] + z_0$$  \hspace{1cm} (10)

The resulting cross sectional area function $A(\tau)$ for the inlet and the outlet of the compressor as well as the volume function $V(\tau)$ of all chambers are shown in Figure 4. Each chamber has a helix angle range of $\Delta \tau_{\text{chamber}} = 425^\circ$. It should be noted that the helix angle range of both inlet chamber (S) and outlet chambers (D) will vary from $0 \leq \Delta \tau_{S,D} \leq \Delta \tau_{\text{chamber}}$ due to the opening or closing of the chamber.

The volume function of both rotors lead to a theoretical suction volume of 469 cm$^3$ considering both teeth on each profile. The calculation of the inner volume ratio yield 6.8 for the investigated compressor.

2.2. Leakage model

The leakage model of each chamber is based on a model used by Rohe [12]. The leakage is assumed to be an isentropic and adiabatic continuum flow and differentiates between subsonic and supersonic flow:

$$\dot{m}_{\text{Leak}, \text{Max}^{\text{I}}} = \zeta \cdot A_L \cdot \frac{p_1}{R \cdot T_i} \cdot \left( \frac{p_2}{p_1} \right)^{\frac{1}{\kappa-1}} \cdot \sqrt{\frac{2 \cdot \kappa}{\kappa - 1}} \cdot R \cdot T_i \cdot \left[ 1 - \left( \frac{p_2}{p_1} \right)^{\frac{\kappa}{\kappa-1}} \right]$$  \hspace{1cm} (11)

$$\dot{m}_{\text{Leak}, \text{Max}^{\text{II}}} = \zeta \cdot A_L \cdot \left( \frac{2}{\kappa+1} \right)^{\frac{1}{\kappa+1}} \cdot \frac{p_1}{T_i} \cdot R \cdot \sqrt{\frac{2 \cdot \kappa}{\kappa+1}} \cdot R \cdot T_i$$  \hspace{1cm} (12)
The criterion for the transition to supersonic flow is defined by a critical pressure ratio in equation (13).

\[
\pi_{\text{crit}} = \frac{p_{\text{out}}}{p_1} = \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}}
\]  

(13)

In case of an inversion of the pressure gradient due to large leakage flows, the leakage flow is recalculated by using a pressure equalizing flow approach introduced by Rohe [12]. This approach assumes a certain mass exchange \( \Delta m_{\text{balance}} \) that leads to a balance in pressure. The exchanged mass is defined in equation (14)

\[
\Delta m_{\text{balance}} = m_2 - \left( \frac{T_1}{T_2} \cdot m_1 \cdot m_2^{\frac{\kappa}{\kappa - 1}} + m_2 \right)^{\frac{1}{\kappa}} \cdot \left( \frac{v_1}{v_2} + 1 \right)^{\frac{1}{\kappa}}
\]  

(14)

According to the investigations of Peveling [11] the pressure loss coefficient used in equation (11) and (12) depends on multiple parameters such as the area of the leakage path, the velocity of the gas flow the pressure difference, the height and the three dimensional geometry of the of the leakage gap. This model assumes constant pressure loss coefficients for each leakage type respectively. The various leakage types and paths of the screw compressor have been identified by Ohbayashi et al. [13] and Rohe [12] and are shown in Figure 5.

**Figure 5.** Leakage paths of a screw compressor at a rotation of 45° and 60°

The leakage areas where computed based on the given rotor profile geometry. Their calculation is based on equation (15). It assumes a constant clearance \( h_G \) and \( h_R \) for the housing leakage (A) and radial leakage (B) and a varying gap and the gap for the profile intermesh leakage (C) \( h_P(\tau) \) and the blow hole leakage (D) \( h_B(\tau) \).

\[
A_{LX} = \int h_X(\tau) \frac{dz(\tau)}{d\tau} d\tau \quad \text{with} \quad X \in \{G, R, B, P\}
\]  

(15)

Since the leakage areas depend on the temperature of the housing and the rotor, the actual leakage gaps could not be measured. The clearances are based on the cold state of the machine. The pressure loss coefficients are assumed to constant and based on the investigations of (Peveling [11]). Both clearances and pressure loss coefficients are summarized in Table 2.
Table 2: Measured clearance and assumed loss coefficients for the given screw compressor

| Parameter                                     | Symbol | Value | Unit |
|-----------------------------------------------|--------|-------|------|
| Housing clearance                             | $h_\text{G}$ | 0.15  | mm  |
| Radial clearance                              | $h_\text{R}$ | 0.10  | mm  |
| Pressure loss coefficient (housing leakage)   | $\zeta_\text{G}$ | 0.2   | -    |
| Pressure loss coefficient (radial leakage)    | $\zeta_\text{R}$ | 0.8   | -    |
| Pressure loss coefficient (Intermesh leakage) | $\zeta_\text{P}$ | 0.8   | -    |
| Pressure loss coefficient (Blowhole)          | $\zeta_{\text{BH}}$ | 0.8   |      |

2.3. Heat transfer model

The investigated screw compressor has a water-cooled housing. The housing is assumed to have a constant wall temperature. The heat is transferred through two different paths as shown in Figure 6. The overall transferred heat of each chamber $\Delta Q_i$ was calculated as followed.

$$\Delta Q_i = \left[ \dot{Q}_{\text{c}}(\tau_i) + \dot{Q}_{\text{h}}(\tau_i) \right] \cdot (2\pi \cdot n)^{-1} \cdot \Delta \tau$$  \hspace{1cm} (16)

The heat transport mechanism between the enclosed chamber volume and the housing $\dot{Q}_{\text{c}}$ is assumed to be a solely convective and the transferred heat flow is defined by equation (17).

$$\dot{Q}_{\text{c}} = \alpha_{\text{avg}} \cdot A_w \cdot \Delta T_{\text{in}}$$  \hspace{1cm} (17)

$$\Delta T_{\text{in}} = \left[ (T_w - T_{\text{in}}) - (T_w - T_{\text{out}}) \right] \cdot \ln \left( \frac{T_w - T_{\text{in}}}{T_w - T_{\text{out}}} \right)^{-1}$$  \hspace{1cm} (18)

$$A_w = \sum \left[ z(\tau_i - \Delta \tau_c) - z(\tau_i - \Delta \tau_{\text{chamber}}) \right] \cdot \Delta \tau \cdot d_c$$  \hspace{1cm} (19)

Figure 6. Heat flow mechanism for each chamber (left) and chamber wall area (right) for heat transfer

The corresponding flow of the enclosed chamber volume is approximated by a ring gap flow and the Nusselt correlation of Gnielinski [15] is used for the average heat transfer coefficient $\alpha_{\text{avg}}$ as defined by equation (20).

$$\alpha_{\text{avg}} = \frac{Nu \cdot \lambda_{\text{avg}}}{d_{\text{hyd}}}$$  \hspace{1cm} (20)
Where $\lambda_{avg}$ is the average heat conductivity of the water vapor, Nu is the Nusselt correlation as defined in [15] and $d_{hyd}$ is the hydraulic diameter.

$$Nu = f \left( \frac{d_i}{d_u}, Re, Pr, \frac{d_{hyd}}{l} \right)$$ (21)

with $\frac{d_{hyd}(\tau)}{l} = \frac{d_c - d_R}{z(\tau + \Delta \tau_{chamber}) - z(\tau)}$; $d_l = d_R$; $Re = \frac{w_{in} \cdot d_{hyd}}{v}$

The velocity of the ring gap flow $w_{in}$ is based on the chambers median as defined by equation (22).

$$w_{in}(\tau) = \frac{z(\tau + \Delta \tau_{chamber} / 2 + 2 \pi \cdot \Delta t) - z(\tau + \Delta \tau_{chamber} / 2)}{\Delta t}$$ (22)

The fluid properties are calculated at the mean temperature $T_m = \frac{(T_{in} + T_{out})}{2}$ (using the thermophysical property library CoolProp by Bell et al. [16].

The leakage flow through the housing clearance is rather complex due to its moving boundaries, the unknown polytropic exponent and the acceleration of the flow towards the smallest gap diameter [12]. The model assumes a full heat transfer for the gap flow which leads to an outlet temperature at the gap that is equal to the wall temperature. The corresponding heat flow is defined by equation (23).

$$\dot{Q}_{le} = m_L \cdot c_{p,in} \cdot (T_{in} - T_{in})$$ (23)

2.4. Compressor performance indicators

The compressor performance of refrigeration compressors is usually characterized by its volumetric and isentropic efficiency as well as resulting discharge temperatures. The volumetric efficiency $\eta_v$ is based on a theoretical swept volume $V_{th}$ as defined in equation (24).

$$\eta_v = \frac{m_D}{V_{in} \cdot \rho_g} = \frac{1}{k \cdot V_{k,\max}} \cdot \frac{(T_0 + \Delta T_{sh}) \cdot R}{p_0} \cdot \frac{1}{2\pi \cdot n} \cdot \sum_{\tau=0}^{\tau} m_D(\tau) \cdot \Delta \tau$$ (24)

The isentropic efficiency is based on equation (25). The electrical efficiency and the mechanical efficiency were assumed to be constant for the entire operating envelope with $\eta_{el} \cdot \eta_{mech} = 0.9$. The internal efficiency $\eta_i$ is defined in equation (26) as the ratio of the isentropic work and the indicated work resulting from the p-V diagram. The isentropic work for an ideal gas is calculated with equation (27).

$$\eta_i = \eta_v \cdot \eta_{mech} \cdot \eta_{el}$$ (25)

$$\eta_i = \frac{W_{is}}{W_i} = \left[ \frac{1}{V} \int p(V) dV \right]^{-1} \cdot \frac{m_D \cdot W_{in}}{n}$$ (26)

$$W_{is} = \frac{\kappa}{\kappa - 1} \cdot R \cdot (T_0 + \Delta T_{sh}) \cdot \left[ \left( \frac{p_c}{p_0} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right]$$ (27)

3. Simulative analysis and results

The compressor model was applied for the given compressor geometry at various boundary conditions as summarized in Table 3.
**Table 3:** Boundary conditions for the compressor analysis

| Parameter                          | Symbol | Value / Range | Unit  |
|-----------------------------------|--------|---------------|-------|
| Evaporation temperature (pressure)| \( \vartheta_0 (p_0) \) | 2 to 15 (0.7 to 1.7) | °C (kPa) |
| Condensing temperature (pressure) | \( \vartheta_C (p_C) \) | 25 to 85 (3.2 to 57.8) | °C (kPa) |
| Suction Superheat                 | \( \Delta T_{SH} \) | 10 | K |
| Compressor Speed                  | \( n \) | 50 to 200 | Hz |
| Housing temperature               | \( \vartheta_W \) | 40 | °C |

The resulting p-V diagram for the reference case with \( \vartheta_0 = 5 \) °C (\( p_0 = 0.87 \) kPa), \( \vartheta_C = 50 \) °C (\( p_C = 12.35 \) kPa) and \( n = 100 \) Hz is shown in **Figure 7**. The result shows that the compressor is at its optimum concerning the internal volume ratio. Further, the pressure increase due to the internal leakage flow from the discharge chamber leads to an increase in indicated work load as compared to an isentropic compression. The isentropic efficiency is 0.68 and the volumetric efficiency is 0.78.

![Figure 7. Resulting p-V diagram for the screw compressor at \( \vartheta_0 = 5 \) °C, \( \vartheta_C = 50 \) °C and \( n = 100 \) Hz](image)

Figure 8 shows the result of the compressor performance for the entire operating envelope in regards to the isentropic efficiency, volumetric efficiency and discharge temperatures. It can be seen that the volumetric efficiency stays above 80 % for compressor speeds > 50 Hz. The isentropic efficiency reaches values > 80 %. However, these values are only achieved for compression ratios close to the design point. This shows the disadvantage of the under-compression and over-compression effect due to the internal volume ratio. The discharge temperature increases with the compression ratio and reaches temperature far beyond 500 °C. This is due to the work load which increases with higher mass flow rates and increasing pressure ratios. The rogue points at 50 Hz can be discarded as calculation errors due to the low mass flow rates at this operating condition.

**4. Conclusion**

The investigation shows that the vacuum compressor are thermodynamically capable of compressing water vapor even for higher temperature lifts. However, due to its internal volume ratio, the compressor may not operate efficiently at higher pressure ratios. It was shown that the compressor has a good volumetric efficiency for compressor speeds > 50Hz. The results for the high discharge temperatures suggests, that the cooling through the compressor housing is not sufficient for higher pressure ratios and needs to be substituted by other cooling means, e.g. liquid injection. Further experimental investigations are required to validate the presented model.
Figure 8. Performance diagrams for the investigated compressor: Compressor isentropic efficiency (upper left), Volumetric efficiency (upper right), discharge temperature (lower).

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**Nomenclature**

| Symbol | Description |
|--------|-------------|
| A      | Area (m²)   |
| cp     | Heat capacity at const. pressure (J/kgK) |
| cv     | Heat capacity of const. volumes (J/kgK) |
| d      | Diameter (m) |
| h      | Specific enthalpy (J/kgK), pitch of rotor helix (m), gap height (m) |
| h      | Length (m)   |
| Ma     | Mach number (-) |
| m, ṁ  | Mass (kg), mass flow rate (kg/s) |
| n      | Speed (1/s) |
| Nₙ     | Number of teeth |
| Nu     | Nusselt number (-) |
| p      | Pressure (Pa) |
| Pr     | Prandtl number (-) |
| R      | Specific gas constant (J/kgK) |
| Re     | Reynolds number (-) |
| Q      | Heat (J) |
| U      | Inner Energy (J) |
| v, V   | Specific Volume (m³/kg), Volume (m³) |
| W      | Work (J) |
| t, Δt  | Time, time difference (s) |
| ΔT     | Temperature difference (K) |
| z      | Position on the z-Axis (m) |
| β      | Helix pitch angle |
| ζ      | Pressure loss coefficient (-) |
| η      | Efficiency (-) |
| θ      | Temperature (°C) |
| κ      | Isentropic coefficient (-) |
| λ      | Linear coefficient of variable pitch (-), heat conductivity (W/mK) |
| ν      | Kinematic viscosity (m²/s) |

**Pressure ratio (-)**

**Rotational angle (°)**

**Flow coefficient (-)**

**Stage-loading coefficient (-)**

**Subscripts**

| Symbol | Description |
|--------|-------------|
| 0      | Evaporation or Initial |
| 1,2,...,n-1 | Chamber state points |
| a      | Outer |
| avg    | Average |
| B      | Blowhole |
| C      | Crown, convective, condensation |
| crit   | Critical |
| D      | Discharge |
| el     | Electric |
| i      | Index, inner, internal |
| is     | Isentropic |
| hydr.  | Hydraulic |
| k      | Chamber |
| L      | Leakage |
| ln     | Logarithmic |
| m      | Median |
| mech   | Mechanical |
| P      | Profile |
| R      | Root |
| ref    | Reference point @273.15K |
| s      | Isentropic |
| S      | Suction |
| SH     | Superheat |
| u      | Circumference |
| w      | Wall |