Screw Compressor Basics

Screw Compressor
Modelling, Design and Use
Professor N. Stosic

Chair in Positive Displacement Compressor Technology

Centre for
Positive Displacement Compressor Technology
City University London, U.K.
INTRODUCTION

Screw Compressor Today

Highly competitive market, specially in air compression and refrigeration

Continuous improvement: more compact, efficient and cost effective compressors

New rotor generation, rotors optimized for certain compressor duty, specialized design

Scope for innovation, improvement and development
INTRODUCTION

Basics

View from Front and Top

Top and front: Admission
Bottom: Change of volume
Bottom and Rear: Discharge

View from Bottom and Rear
INTRODUCTION

Basics
Swedish company SRM, pioneer and leader

Screw compressor profiles, Symmetric, Asymmetric, ‘D’ and ‘G’
Screw compressor design
Screw compressor technology

Licence system left many screw compressor manufacturers at margins of research and development
Many companies started their own development

Gardner Denver, Atlas Copco, Compair, Kaeser, GHH
Trane, Ingersol-Rand, Hitachi, Fu Sheng, Hanbel, Refcomp
Holroyd

Many more or less successful rotor profiles

Many more or less successful screw compressor designs

Need exists to concentrate efforts in R&D
INTRODUCTION

Improved methods of analysis
Experimental validation
Design of critical components
Complete machine design
Product development
Training in machine design
Methods Applied

Advanced Computerized design tools
Machine process modelling
2-D and 3-D Computational Fluid Dynamics

Modern experimental technique
Computerized data acquisition

Users

Renown and new companies,
large and small manufacturers
in the U.K and abroad
SCREW COMPRESSOR GEOMETRY

Before modelling the physical process, the rotor lobe profiles must be defined together with the remaining parameters with which the rotor and housing geometry can be fully specified.

Rotor profile: x and y coordinates, pressure angle

Helix/lead angle, rotor length

Interlobe, end and radial clearance

Suction/discharge ports
General case: non-parallel and non-intersecting

Given profile

\[ \mathbf{r}_1 = \mathbf{r}_1(t, \theta) = [x_1, y_1, z_1] = [x_{01}\cos \theta - y_{01}\sin \theta, x_{01}\sin \theta + y_{01}\cos \theta, p_1]\]

\[ \frac{\partial \mathbf{r}_1}{\partial t} = \begin{bmatrix} \frac{\partial x_{01}}{\partial t}\cos \theta - \frac{\partial y_{01}}{\partial t}\sin \theta, \frac{\partial x_{01}}{\partial t}\sin \theta + \frac{\partial y_{01}}{\partial t}\cos \theta, 0 \end{bmatrix} \]

\[ \frac{\partial \mathbf{r}_1}{\partial \theta} = \begin{bmatrix} \frac{\partial x_{01}}{\partial \theta}, \frac{\partial y_{01}}{\partial \theta}, 0 \end{bmatrix} = [-y_{01}, x_{01}, 0] \]
General case: non-parallel and non-intersecting

Meshed profile

\[ r_2 = r_2(t, \theta, \tau) = [x_2, y_2, z_2] = [x_1 - C, y_1 \cos \Sigma - z_1 \sin \Sigma, y_1 \sin \Sigma + z_1 \cos \Sigma] = \]
\[ [x_{02} \cos \tau - y_{02} \sin \tau, x_{02} \sin \tau + y_{02} \cos \tau, p_2 \tau] \]

\[ \frac{\partial r_2}{\partial \tau} = [-y_2, x_2, p_2] = [x_{02} \sin \tau + y_{02} \cos \tau, x_{02} \cos \tau - y_{02} \sin \tau, p_2] = \]
\[ [p_1 \theta \sin \Sigma - y_1 \cos \Sigma, p_2 \sin \Sigma + (x_1 - C) \cos \Sigma, p_2 \cos \Sigma - (x_1 - C) \sin \Sigma] \]

Meshing condition

\[ \left( \frac{\partial r_1}{\partial t} \times \frac{\partial r_1}{\partial \theta} \right) \cdot \frac{\partial r_1}{\partial \tau} = -\left( \frac{\partial r_1}{\partial t} \times \frac{\partial r_1}{\partial \theta} \right) \cdot \frac{\partial r_2}{\partial \tau} = 0 \]

\[ \left[ C - x_1 + (p_1 - p_2 \cot \Sigma) \right] \left( x_1 \frac{\partial x_1}{\partial t} + y_1 \frac{\partial y_1}{\partial t} \right) + p_1 \left[ p_1 \theta \frac{\partial y_1}{\partial t} + (p_2 - C \cot \Sigma) \frac{\partial x_1}{\partial t} \right] = 0 \]
SCREW COMPRESSOR GEOMETRY

General case: non-parallel and non-intersecting corresponds to the rotor – hobbing tool relation

Special cases:

\( p_2=0 \), rotor - plate milling tool, grinding tool relation

\( \Sigma=0 \), screw compressor rotors
SCREW COMPRESSOR GEOMETRY

Rotor profile, $\Sigma=0$, $i=p_2/p_1$, $k=1-1/i$

Meshing condition

$$\frac{dy_{01}}{dx_{01}} \left( ky_{01} - \frac{C}{i} \sin \theta \right) + kx_{01} + \frac{C}{i} \cos \theta = 0$$

Meshed profile

$$x_{02} = x_{01} \cos k\theta - y_{01} \sin k\theta - C \cos \frac{\theta}{i}$$

$$y_{02} = x_{01} \sin k\theta + y_{01} \cos k\theta + C \sin \frac{\theta}{i}$$

Rack profile

$$x_{0r} = x_{01} \cos \theta - y_{01} \sin \theta$$

$$y_{0r} = x_{01} \sin \theta + y_{01} \cos \theta - r_1 \theta$$
Numerical solution of the meshing condition

Task: to find θ for a zero function

Simple iteration method:
Fast and reliable, but valid only for certain function

Additional complication
In certain areas two or more θ are the zero function
Only one is valid, additional values found by half interval method
Demonstrator profile
‘N’ ROTOR PROFILE

- Rack generation procedure
- Straight line on the rack - involute rotor contact
- Small torque transmitted
- Large displacement
- Strong gate rotor,

- Short sealing line

SCREW COMPRESSOR GEOMETRY

E-F Circle
F-G Straight Line
G-H Undercut by the Gate Rotor
H-A Undercut by the Main Rotor
A-B Arc: p=0.43, q=1
B-C Straight Line
C-D Circle
D-E Straight Line
SCREW COMPRESSOR THERMODYNAMICS

Differential approach:

Set of differential equations solved simultaneously
Equations of continuity, momentum and energy

Preintegrated equations inadequate and inaccurate
if high leakage rate and heat transfer is involved
SCREW COMPRESSOR THERMODYNAMICS

Internal Energy

\[ \omega \frac{dU}{d\theta} = m_{in} h_{in} - m_{out} h_{out} + Q - \omega p \frac{dV}{d\theta} \]

\[ m_{in} h_{in} = m_{suc} h_{suc} + m_{l,g} h_{l,g} + m_{oil} h_{oil} \quad m_{out} h_{out} = m_{dis} h_{dis} + m_{l,l} h_{l,l} \]

Continuity

\[ \omega \frac{dm}{d\theta} = m_{in} - m_{out} \]

\[ m_{in} = m_{suc} + m_{l,g} + m_{oil} \quad m_{out} = m_{dis} + m_{l,l} \quad \dot{m} = \rho w A \]

Leakage Flow

Momentum

\[ w_{l} dw_{l} + \frac{dp}{\rho} + f \frac{w_{l}^2}{2} \frac{dx}{D_{g}} = 0 \]

\[ m_{l} = \rho_{l} w_{l} A_{g} = A_{g} \sqrt{\frac{p_{2}^2 - p_{1}^2}{a^2 \left( \zeta + 2 \ln \frac{p_{2}}{p_{1}} \right)}} \]
Oil injection

\[
\frac{dT_{oil}}{d\theta} = \frac{h_o A_o (T - T_{oil})}{\omega m_{oil} c_{oil}} \quad T_{oil} = \frac{T - kT_{oil,p}}{1 + k}
\]

\[
k = \frac{\omega m_{oil} c_{oil}}{h_o A_o \Delta \theta} = \frac{\omega d_s c_{oil}}{6 h_o \Delta \theta}
\]

Numerical solution, Runge-Kutta IV order solver

\( U(\theta), \ m(\theta), \ V(\theta), \ v = \frac{V}{m}, \ U = (mu) + (mu)_{oil}, \ u = \frac{U - (mcT)_{oil}}{m} \)

Ideal Gas

\( T = (\gamma - 1) \frac{u}{R} \quad p = \frac{RT}{v} \)

Real gas

\( p = f_1(T,v) \quad u = f_2(T,v) \)

Wet vapour

\( u = (1 - x) u_f + xu_g \quad v = (1 - x) v_f + xv_g \)
Compressor integral parameters

\[ m = m_{in} - m_{out} \]
\[ \dot{m} = mz_1 n / 60 \]
\[ \dot{V} = 60m / \rho_0 \]
\[ \dot{m}_t = \frac{(F_{1n} + F_{2n}) \ln z_1 \rho}{60} \]
\[ W_{ind} = \int_{cycle} Vdp \]
\[ P_{ind} = \frac{W_{ind} z_1 n}{60} \]
\[ W_{sind} = \int_{cycle} \frac{V}{m} dp \]
\[ \eta_v = \frac{\dot{m}}{\dot{m}_t} \]
\[ \eta_t = \frac{W_t}{W_{ind}} \]
\[ \eta_a = \frac{W_a}{W_{ind}} \]
\[ P_{sind} = \frac{P}{\dot{V}} \]
\[ W_t = RT_1 \ln \frac{p_2}{p_1} \]
\[ W_a = \frac{\gamma}{\gamma - 1} R(T_2 - T_1) \]
SCREW COMPRESSOR THERMODYNAMICS

Calculation of pressure loads
On compressor rotors

Radial forces

\[ R_x = -p \int_{A}^{B} dy = -p \left( y_B - y_A \right) \]
\[ R_y = -p \int_{A}^{B} dx = -p \left( x_B - x_A \right) \]

Rotor torque

\[ T = p \int_{A}^{B} x dx + p \int_{A}^{B} y dy - 0.5p \left( x_B^2 - x_A^2 + y_B^2 - y_A^2 \right) \]
Bearing reactions and rotor deflections

\[
\frac{d^2 \delta}{dz^2} = \frac{M}{EI}
\]
Optimization variables and target function

Single stage:
Rotor variables:
- $r_0$ Female rotor addendum
- $r_1$ Male rotor lobe radius
- $r_2$ Male rotor tip radius
- $r_3$ Female rotor tip radius
Compressor variables:
- Built-in volume ratio
Operation variables:
- Shaft speed
- Oil flow
- Injection position
- Oil temperature
9 Variables

Multistage:
9 Variables x Number of stages
+ Interstage pressures

Target function:
Specific power combined with compressor price

$F = w_1 L + w_2 C$
Box constrained simplex method for efficient and reliable multivariable optimization

\[ f(x_1, x_2, \ldots, x_n) \quad F = w_1 L + w_2 C \]

\[ g_i \leq x_i \leq h_i, \ i = 1, n \quad g_i \leq y_i \leq h_i, \ i = n + 1, m \quad y_{n+1}, \ldots, y_m \]

\[ f(x^h) = \max f(x^1), f(x^2), \ldots, f(x^k) \]

\[ f(x^g) = \min f(x^1), f(x^2), \ldots, f(x^k) \]

\[
\bar{x} = \frac{1}{k-1} \sum_{i=1}^{k} x^i, \quad x^i \neq x^l \quad x^r = \bar{x} + \alpha(\bar{x} - x^l) 
\]

\[ x_r^{\text{(new)}} = 0.5 \left[ x_r^{\text{(old)}} + c\bar{x} + (1-c)x^h \right] + (\bar{x} - x^h)(1-c)(2R-1) \]

\[ c = \left( \frac{n_r}{n_r + k_r - 1} \right)^{n_r + k_r - 1} \]
<table>
<thead>
<tr>
<th></th>
<th>Dry</th>
<th>Oil-Flooded</th>
<th>Refrigeration</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_0 ) [mm]</td>
<td>2.62</td>
<td>0.74</td>
<td>0.83</td>
</tr>
<tr>
<td>( r_1 ) [mm]</td>
<td>19.9</td>
<td>17.8</td>
<td>19.3</td>
</tr>
<tr>
<td>( r_2 ) [mm]</td>
<td>6.9</td>
<td>5.3</td>
<td>4.5</td>
</tr>
<tr>
<td>( r_3 ) [mm]</td>
<td>11.2</td>
<td>5.5</td>
<td>5.2</td>
</tr>
<tr>
<td>Built-in volume ratio</td>
<td>1.83</td>
<td>4.1</td>
<td>3.7</td>
</tr>
<tr>
<td>Rotor speed [rpm]</td>
<td>7560</td>
<td>3690</td>
<td>3570</td>
</tr>
<tr>
<td>Oil flow [lit/min]</td>
<td>-</td>
<td>12</td>
<td>8</td>
</tr>
<tr>
<td>Injection position [°]</td>
<td>-</td>
<td>65</td>
<td>61</td>
</tr>
<tr>
<td>Oil temperature [°]</td>
<td>-</td>
<td>33</td>
<td>32</td>
</tr>
</tbody>
</table>
EXAMPLE OF CALCULATION

5-6-128 mm Oil-Flooded Air Compressor

7 m³/min, max 10 m³/min at 8 bar (abs)

5-14 bar (abs), max 15 bar (max)
Rotors optimized for oil flooded operated air compression

5/6-128 mm, L/D 1.65
Displacement 1.56 l/rev
Interlobe sealing line 0.13 m
Blow-hole area 1.85 mm²
EXAMPLES OF CALCULATION

CAD Interface: Compressor ports
EXAMPLES OF CALCULATION

Experimental verification of the model
Compressor in the test bed
EXAMPLES OF CALCULATION

Comparison of the calculated and test results Flow-Power

5-6 City 90 mm Compressor
Air Inlet 1 b, 20 degC
Air Outlet 6-14 b
Oil Inlet 60 degC

![Graph showing Flow-Power comparison](image)
EXAMPLES OF 3-D CFD CALCULATION

Majority of design problems can be solved by the one-dimensional approach, some of them require the two dimensional calculation, however, there are situations where 3-D CFD must be applied.

Such are

Oil flow distribution,
Fluid-Solid Interaction
Grid generation - 1

- Grid topology
  - polyhedral
  - O - grid
  - H - grid
  - C - grid

- Cell shape
Grid generation - 2

- Grid topology strongly affects accuracy, efficiency and ease of calculation
- Full structured block generated hexahedral 3D-O mesh
- Screw compressor sub-domains:
  - **Male rotor** - **Female rotor**
  - **End clearances**
  - Rotor connections, clearances, leakage paths
  - **Suction port** - **Discharge port**
  - **Suction and discharge receivers**

Automatic discretization process:
- The rack generating procedure
- Rack - a rotor with an infinite radius
- Divides working domain in two parts
  male and female rotor,
Screw Compressor FSI calculations

Mathematical model for screw compressor is based on conservation laws of continuity, momentum, energy, concentration and space:

\[
\frac{d}{dt} \int \rho dV + \int \rho (\mathbf{v} - \mathbf{v}_s) \cdot ds = 0
\]

\[
\frac{d}{dt} \int \rho \mathbf{v} dV + \int \rho \mathbf{v} (\mathbf{v} - \mathbf{v}_s) \cdot ds = \int \mathbf{T} \cdot ds + \int \mathbf{f}_b dV
\]

\[
\frac{d}{dt} \int \rho h dV + \int \rho h (\mathbf{v} - \mathbf{v}_s) \cdot ds = \int \mathbf{q}_h \cdot ds + \int \mathbf{q} dV - \int \mathbf{q}_s \cdot ds + \int \rho dV
\]

\[
\frac{d}{dt} \int \rho c_o dV + \int \rho c_o (\mathbf{v} - \mathbf{v}_s) \cdot ds = \int \mathbf{q}_{c_o} \cdot ds + \int \mathbf{S}_{c_o} dV
\]

\[
\frac{d}{dt} \int dV - \int \mathbf{v}_s \cdot ds = 0
\]

Closed by constitutive relations and equation of state and accompanied by turbulence model.

\[
\frac{d}{dt} \int \rho k dV + \int \rho k (\mathbf{v} - \mathbf{v}_s) \cdot ds = \int \mathbf{q}_k \cdot ds + \int (P - \rho \varepsilon) dV,
\]

\[
\frac{d}{dt} \int \rho \varepsilon dV + \int \rho \varepsilon (\mathbf{v} - \mathbf{v}_s) \cdot ds = \int \mathbf{q}_\varepsilon \cdot ds + \int (C_1 P \frac{\varepsilon}{k} - C_2 \rho \frac{\varepsilon^2}{k} + C_3 \rho \varepsilon \nabla \cdot \mathbf{v}) dV
\]
Thermodynamic properties of real fluids

- $p$-$v$-$T$ equation
  compressibility factor $z$
- $z$ is assumed to change linearly with pressure err<2%

- Antoine equation for saturation temperature

- Clapeyron equation for latent heat

- Specific heat for constant pressure

- Density of mixture

- Coefficient in the pressure correction equation

\[
\frac{p}{\rho} = z \cdot RT = z(p) \cdot RT
\]

\[
z = p \cdot B_1 + B_2
\]

\[
T_{sat} = \frac{A_2}{A_1 - \log p}
\]

\[
h_L = T \left( \frac{1}{\rho_v} - \frac{1}{\rho_l} \right) \frac{dP_{sat}}{dT_{sat}}
\]

\[
c_{pv} = C_0 + C_1 \cdot T + C_2 \cdot T^2 + C_3 \cdot T^3
\]

\[
\rho = \frac{1}{1 - \frac{c_{o_2}}{\rho_v} - \frac{c_{o_2}}{\rho_l}}
\]

\[
C_\rho = \left( \frac{dp}{d\rho} \right)_T = \left( \frac{1}{zRT} - \frac{\rho_v \cdot b_1}{z} \right) \frac{\rho_v}{\rho}
\]
**Multiphase flow**

Euler-Lagrange approach

\[ \frac{d(m_o h_o)}{dt} = m_o \frac{dh_o}{dt} + h_{ol} \frac{dm_o}{dt} = \dot{Q}_{con} + \dot{Q}_{mass} \]

- **Oil** is assumed to be a passive ‘species’
  - Mass calculated from the concentration
  - Oil drag force influence concentration
  - Energy source due to heat transfer between working fluid and oil is:

\[ m_o = m \cdot C_o \]
\[ f_{drag} = -\frac{1}{2} \rho A_o C_{drag} |v_o - v|(v_o - v) \]
\[ \dot{Q}_{con} = m_o C_{p_o} \frac{dT}{dt} \approx m_o C_{p_o} \frac{T^k - T^{k-1}}{\delta t} \]

- **Liquid phase** is assumed to be an active ‘species’
  - Mass source
    - evaporated/condensed mass
  - Energy source
    - energy of evaporation/condensation

\[ \dot{m}_L = \frac{m \cdot C_{pm} \cdot (T - T_s)}{h_L} \]
\[ \dot{Q}_{mass} = h_L \frac{dm_L}{dt} \approx h_L \frac{m_{ol} - m_{ol}^s}{\delta t} = h_L \dot{m}_L \]
Boundary conditions

- Wall boundaries with wall functions are introduced on the housing and rotors.

- Compressor positioned between suction and discharge receivers of small volume

- Inlet & outlet receivers and oil port are treated as boundary domains:

- Mass equation corrected by mass source to maintain constant pressure

- Energy equation corrected by energy source to update energy balance

\[
\dot{m}_{\text{add}} \approx \left( \frac{dm}{dt} \right)_{p=\text{const}} = \frac{p_{\text{const}} - p}{V} \cdot \frac{\rho}{\delta t}
\]

\[
\dot{Q}_{\text{add}} = h_{\text{add}} \left( \frac{d\dot{m}_{\text{add}}}{dt} \right)_{p=\text{const}} = \dot{m}_{\text{add}} \cdot h_{\text{add}}
\]
Screw Compressor performance

- Volume flow (inlet and outlet)
- Mass flow (inlet, outlet and oil)
- Boundary forces
- Restraint Forces and Torque
- Compressor shaft power
- Specific power
- Efficiency

Volumetric and adiabatic

\[ \dot{V} = 60 \cdot \sum_{t=t_{\text{start}}}^{t_{\text{end}}} \dot{V}_f^{(t)} \left[ \frac{m^3}{\text{min}} \right], \quad \dot{V}_f^{(t)} = \sum_{i=1}^{I} v_{fi} S_{fi} \]

\[ \dot{m} = \sum_{t=t_{\text{start}}}^{t_{\text{end}}} \dot{V}_f^{(t)} \cdot \bar{\rho}^{(t)} \left[ \text{kg/sec} \right] \]

\[ F_x = p_b \cdot A_{xb}; \quad F_y = p_b \cdot A_{yb}; \quad F_z = p_b \cdot A_{zb} \]

\[ F_{rS} = \sum_{i=1}^{I} F_{rs}(i), [N]; \quad F_{rD} = \sum_{i=1}^{I} F_{rd}(i), [N] \]

\[ F_a = \sum_{i=1}^{I} F_a(i), [N]; \quad T = \sum_{i=1}^{I} T(i), [Nm] \]

\[ P = 2 \cdot \pi \cdot n \cdot (T_M + T_F) \left[ \text{W} \right] \]

\[ P_{\text{spec}} = \frac{P}{\dot{V} \cdot 1000 \left[ \frac{kW}{m^3 \text{min}} \right]} \]

\[ \eta_v = \frac{\dot{V}}{V_d}; \quad \eta_i = \frac{P_{ad}}{P} \]
Oil injected - Pressure in axial section

'N' rotors 5/6
Velocity

Pressure

-1.000e+04
5.600e+04
1.220e+05
1.880e+05
2.540e+05
3.200e+05
3.860e+05
4.520e+05
5.180e+05
5.840e+05
6.500e+05
300
Oil injected - Pressure and velocity
Oil injected - Pressure 3D view

'N' rotors 5/6

Pressure

6.400e+05
5.750e+05
5.100e+05
4.450e+05
3.800e+05
3.150e+05
2.500e+05
1.850e+05
1.200e+05
5.500e+04
-1.000e+04
Real fluid - Ammonia – pressure
Experimental verification – P-α diagram

P-α diagram for the Screw Compressor
‘N’ profile, 5/6, 128mm, 5000rpm
Oil injected – Deformation

Pressure

\[ P_{\text{in}} = 1 \text{ b} \quad P_{\text{out}} = 7 \text{ b} \quad n = 5000 \text{ rpm} \]

\[ t_{\text{in}} = 20 \degree \text{C} \quad t_{\text{out}} = 40 \degree \text{C} \]
Oil injected – Deformation
Pressure-1

\[ P_{\text{ini}} = 1 \text{ b} \quad P_{\text{out}} = 7 \text{ b} \quad n = 5000 \text{ rpm} \]

\[ t_{\text{ini}} = 20 \, ^\circ\text{C} \quad t_{\text{out}} = 40 \, ^\circ\text{C} \]
Oil injected – Deformation
Pressure-2

\[ P_{\text{in}} = 1 \text{ b} \quad P_{\text{out}} = 7 \text{ b} \quad n = 5000 \text{ rpm} \]
\[ t_{\text{in}} = 20^\circ \text{C} \quad t_{\text{out}} = 40^\circ \text{C} \quad \text{mag} = 20,000 \times \]
Oil injected – Deformation
Temperature

\[ P_{\text{in}} = 1 \text{ b} \quad P_{\text{out}} = 3 \text{ b} \quad n = 5000 \text{ rpm} \]
\[ t_{\text{in}} = 20^\circ \text{C} \quad t_{\text{out}} = 150^\circ \text{C} \quad \text{mag} = 1,000 \times \]
Oil injected – Deformation
Pressure + Temperature

\( P_{\text{in}} = 30 \text{ b} \quad P_{\text{out}} = 90 \text{ b} \quad n = 5000 \text{ rpm} \)
\( t_{\text{in}} = 0 \degree \text{C} \quad t_{\text{out}} = 40 \degree \text{C} \quad \text{mag} = 2,000 \times \)
DESIGN EXAMPLES

Oil-free air compressor
Oil-flooded air compressor
Retrofit rotors of an air compressor
Refrigeration compressor
Compressors for Oil-Free Air Delivery

Design aims:

Delivery: 350-700 and 700-1000 m³/h
Working pressure: 1-2.5 (2.7) bar

Volumetric efficiency 90 % +
Low specific power
Simple, reliable and compact machine
3/5 lobe rotors
Large displacement
Convenient gear ratio $5/3 = 1.67$
Features of 3/5 ‘N’ Rotors

- Highest possible displacement
- Higher delivery
- Better volumetric efficiency
- Better adiabatic efficiency
- Stronger gate rotor
- Long durability
- High reliability
- Easy manufacturing
- Easy compressor assembly
- Reasonable noise
New design, fully customized by the manufacturer

New rotors,

New, improved compressor

New concept, better screw compressor compared with competition
Comparison of Test Results

Air Inlet 1 bar, 20 degC
Air Outlet 3 bar

R1- GHH C80
R2- Drum D9000
R3- Mouvex Typhoon
R4- GHH CS-1000
Screw Compressor Family for Oil-Flooded Operation

Design aims:

Delivery: 0.6-60 m³/min
Working pressure: 5-13 bar

Volumetric efficiency 90 % +
Low specific power
Simple, reliable and compact machine
Improved capacity and efficiency
Lower power consumption
Reduced manufacturing cost

4/5 Rotors

Reasonable efficiency for moderate pressure ratios, at least the same as for 4/6 rotors
Improved capacity and efficiency
Lower power consumption
Reduced manufacturing cost
Five compressors, rotor diameters:
74, 102, 159, 225 and 285 mm, L/D 1.55
Performance of the Compressor Family

![Graphs showing the performance of the compressor family.](image)
DESIGN EXAMPLES

Proven design, fully customized to the manufacturer’s needs

New rotors,
New, improved compressor
Test Results, Compressors 73 and 159 mm

4/5-73 Compressor
Air Outlet 5-13 b gauge

4/5-159 Compressor
Air Outlet 5-13 b gauge
Retrofit ‘N’ Rotors

5.4 % more displacement
6.5 % higher delivery
4 % better volumetric efficiency
2.5 % better adiabatic efficiency
75 % less torque on the gate rotor
Asymmetric Rotors, the most common screw compressor rotors

‘N’ Rotor retrofit for more efficient screw compressors
Specific Power (kW/m³/min) vs. Discharge Pressure (bar)

- +++++ Asymmetric Rotors
- +++++ 'N' Rotors
Design of a semihermetic compressor for air-conditioning and refrigeration based on ‘N’ rotors

- Semihermetic, convertible to open
- All existing refrigerants
- Modern, better than competition
- 5/6-102 mm L/D=1.55
- Efficient rotors for air condition and refrigeration
5/6-102 mm L/D=1.55 compressor

- Theoretical displacement 0.72 l/rev
- Gear-box 1500-10000 rpm
- Delivery 0.920-5.80 m³/min
- R-22
- Air condition 5/40 °C 60-380 kW
- COP 4.20 at 3000 rpm
- Refrigeration -5/30 °C 33-210 kW
- COP 3.95 at 3000 rpm
Refrigeration Application
Application of Design Optimization: A Family of Two-Stage Compressors

Range: 22-250 kW
8-16 bar (abs)

Two-stages: 19 Variables
Target Function:
Minimum specific power
And only specific power

Number of frames
VFD vs gearbox
Stage size
Stage speed
Stage rotor profile
CONCLUSIONS

The screw compressor is a mature product at the millenium meeting point. Orchestrated efforts of a large number of companies driven by market forces resulted in the compact and efficient compressor machine. Every detail counts today. A small difference will give a small, but distinctive improvement which may be used as an individual advantage.

Optimization opens hidden spots still left for better compressors. They result in stronger but lighter rotors with higher displacement, more compact and more efficient compressor machines at all areas of their application.
The Centre for Positive Displacement Compressor Technology carries out research and provide a service to manufacturing companies in all aspects of compressor design and development. Through the everyday activity the Centre is gaining experience in the compressor research, development and design. Some of the recent experience is given in this presentation.