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

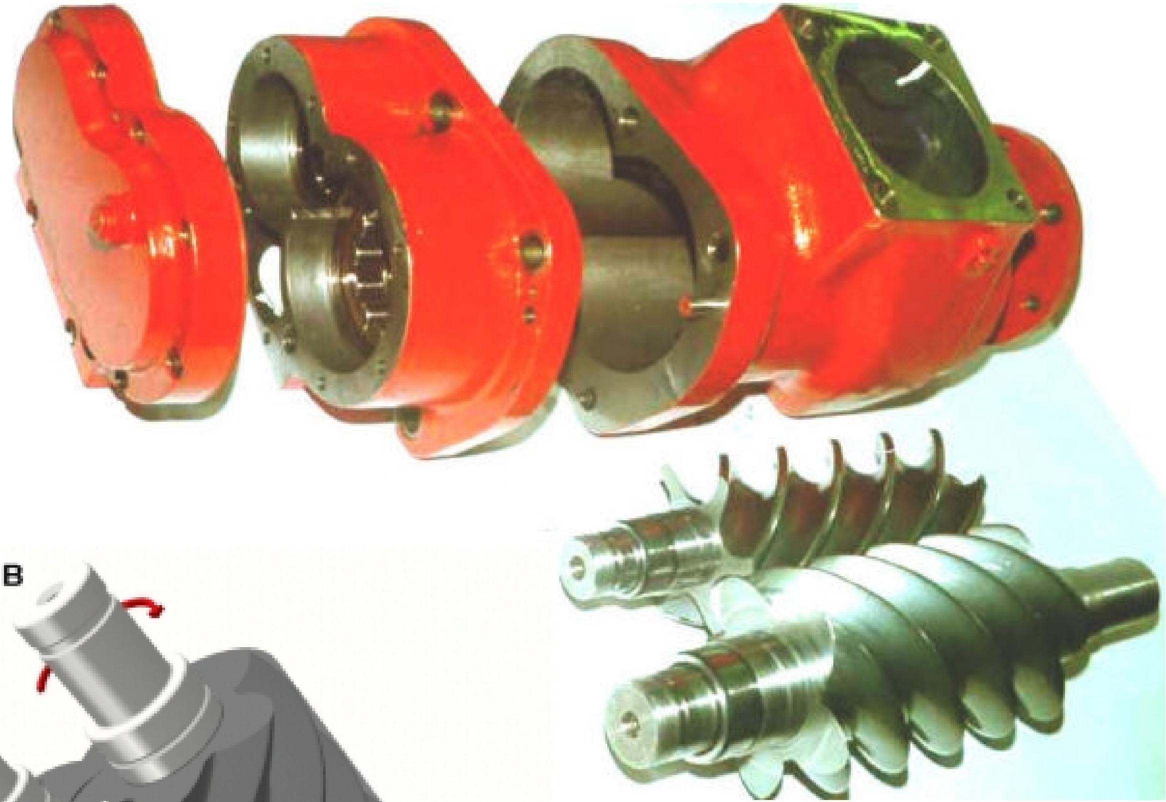
Highly competitive market, specially in air compression and refrigeration

Continuous improvement:
more compact, efficient and cost effective compressors

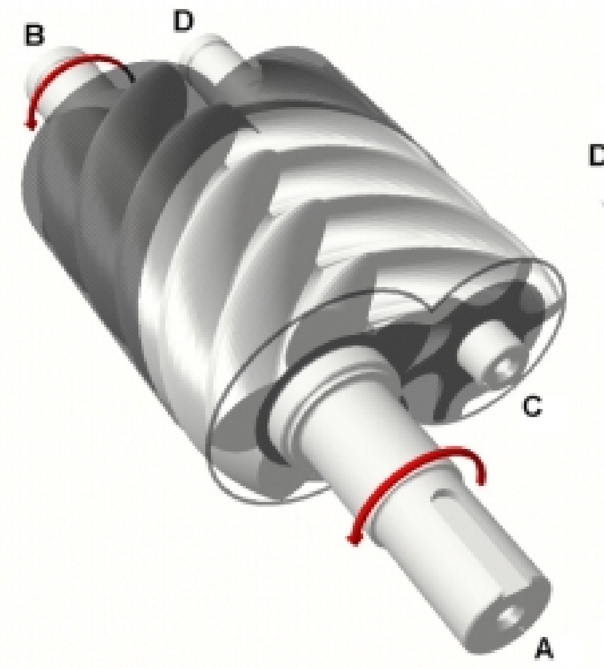
New rotor generation, rotors optimized for certain compressor duty, specialized design

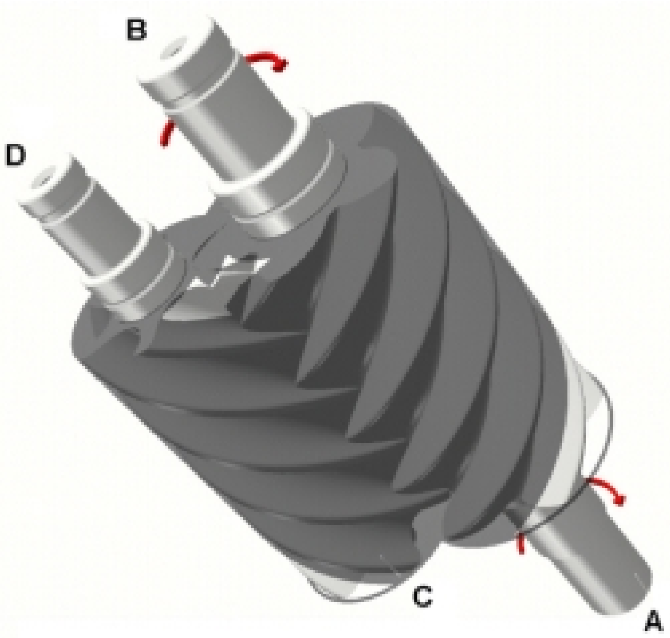
Scope for innovation, improvement and development

Basics



View from Front and Top

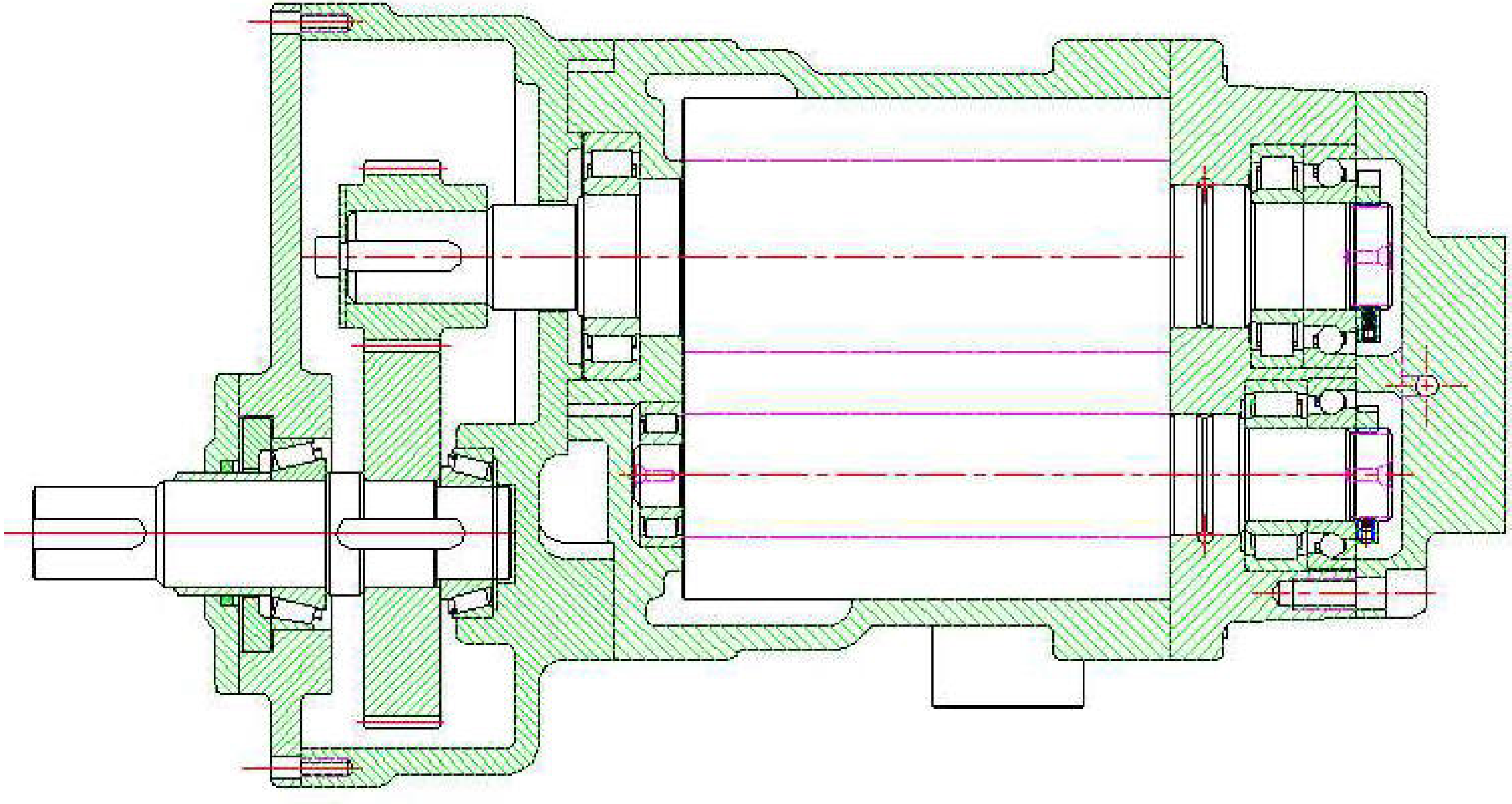




Top and front: Admission
Bottom: Change of volume
Bottom and Rear: Discharge

View from Bottom and Rear

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

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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 COMPRRESSOR 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.

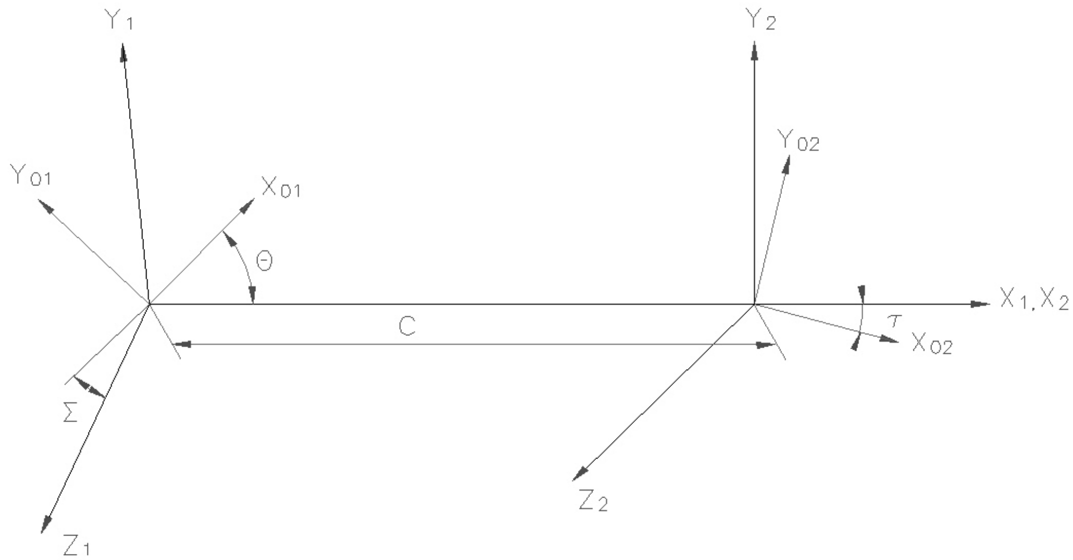
Rotor profile: x and y coordinates, pressure angle

Helix/lead angle, rotor length

Interlobe, end and radial clearance

Suction/discharge ports

General case: non-parallel and non-intersecting



Given profile

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

$$\frac{\partial \mathbf{r}_1}{\partial t} = \left[\frac{\partial x_1}{\partial t}, \frac{\partial y_1}{\partial t}, 0\right] = \left[\frac{\partial x_{01}}{\partial t}\cos\theta - \frac{\partial y_{01}}{\partial t}\sin\theta, \frac{\partial x_{01}}{\partial t}\sin\theta + \frac{\partial y_{01}}{\partial t}\cos\theta, 0\right]$$

$$\frac{\partial \mathbf{r}_1}{\partial \theta} = \left[\frac{\partial x_1}{\partial \theta}, \frac{\partial y_1}{\partial \theta}, 0\right] = [-y_{01}, x_{01}, 0]$$

General case: non-parallel and non-intersecting

Meshed profile

$$\mathbf{r}_2 = \mathbf{r}_2(t,\theta,\tau) = [x_2, y_2, z_2] = [x_1 - C, y_1\cos\Sigma - z_1\sin\Sigma, y_1\sin\Sigma + z_1\cos\Sigma] =$$
$$[x_{02}\cos\tau - y_{02}\sin\tau, x_{02}\sin\tau + y_{02}\cos\tau, p_2\tau]$$

$$\frac{\partial\mathbf{r}_2}{\partial\tau} = [-y_2, x_2, p_2] = [x_{02}\sin\tau + y_{02}\cos\tau, x_{02}\cos\tau - y_{02}\sin\tau, p_2] =$$
$$\left[p_1\theta\sin\Sigma - y_1\cos\Sigma, p_2\sin\Sigma + (x_1 - C)\cos\Sigma, p_2\cos\Sigma - (x_1 - C)\sin\Sigma\right]$$

Meshing condition

$$\left(\frac{\partial\mathbf{r}_1}{\partial t}\times\frac{\partial\mathbf{r}_1}{\partial\theta}\right)\cdot\frac{\partial\mathbf{r}_1}{\partial\tau} = -\left(\frac{\partial\mathbf{r}_1}{\partial t}\times\frac{\partial\mathbf{r}_1}{\partial\theta}\right)\cdot\frac{\partial\mathbf{r}_2}{\partial\tau} = 0$$

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

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

Σ=0, screw compressor rotors

Rotor profile, Σ=0, i=p2/p1, 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

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

$$y_{0r}=x_{01}\sin\theta+y_{01}\cos\theta-r_1\theta$$

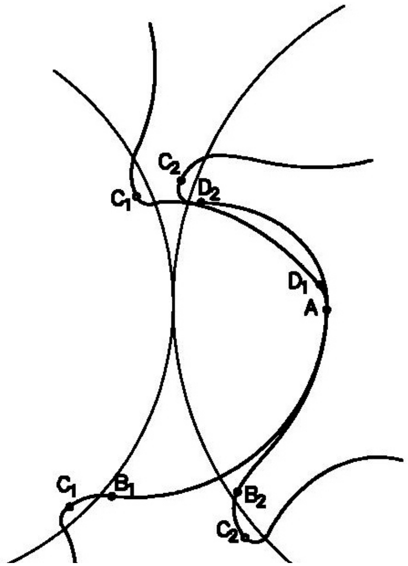
Numerical solution of the meshing condition

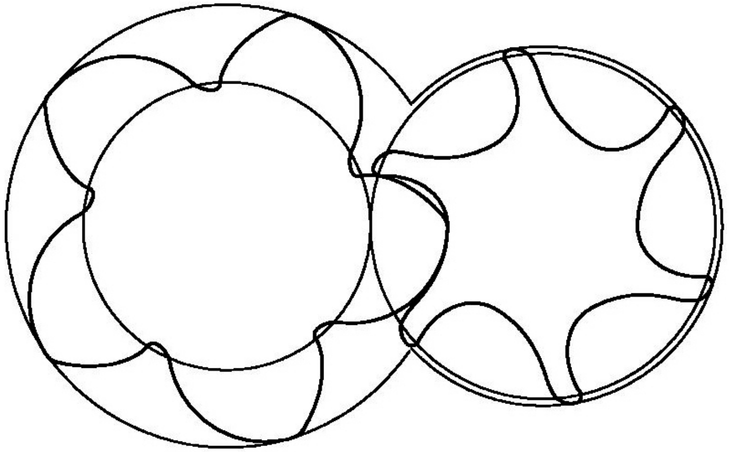
Task: to find θ for a zero function

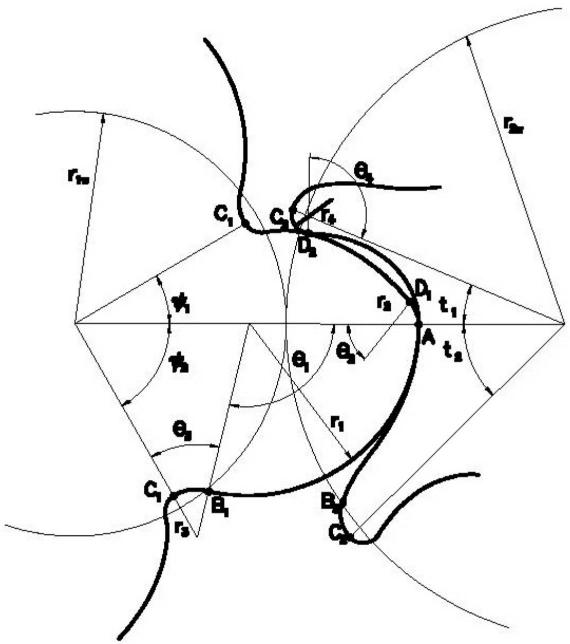
Simple iteration method:
Fast and reliable, but valid only for certain function

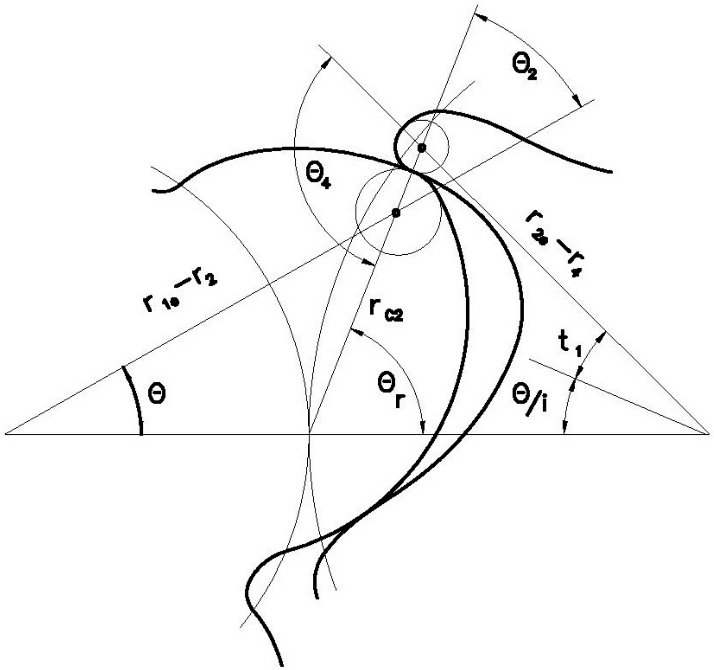
Additional complication
In certain areas two or more θ are the zero function
Only one is valid, additional values found by half interval method

Demonstrator profile



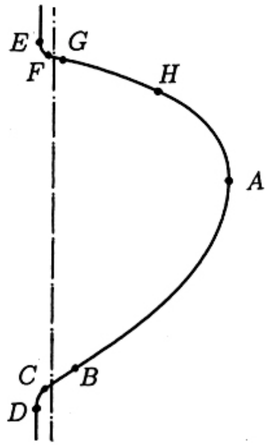




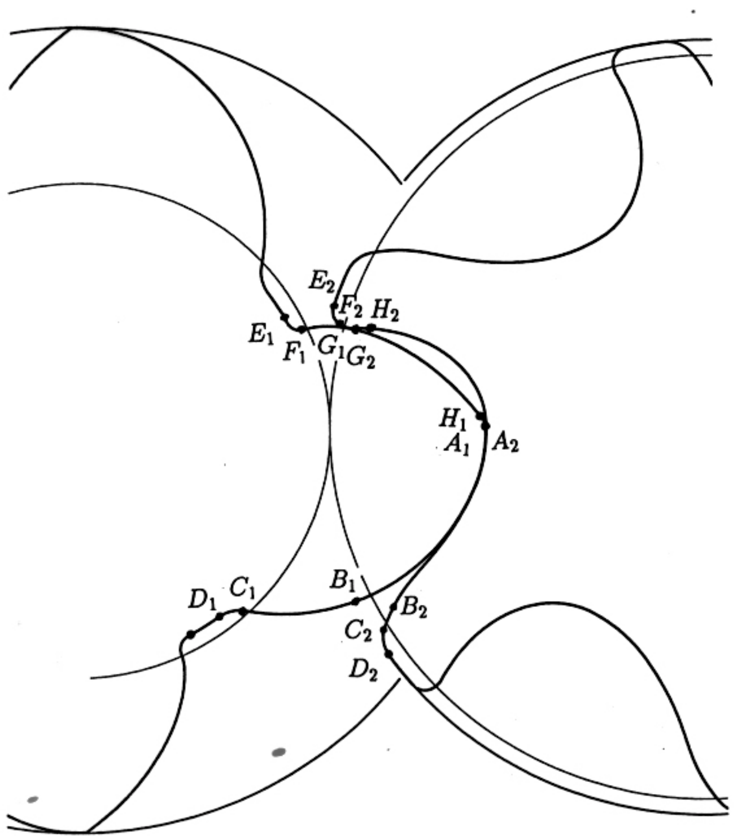


'N' ROTOR PROFILE

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



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 COMPRRESSOR 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

Internal Energy

$$\omega \frac{dU}{d\theta} = \dot{m}_{in} h_{in} - \dot{m}_{out} h_{out} + Q - \omega p \frac{dV}{d\theta}$$

$$\dot{m}_{in} h_{in} = \dot{m}_{suc} h_{suc} + \dot{m}_{l,g} h_{l,g} + \dot{m}_{oil} h_{oil} \qquad \dot{m}_{out} h_{out} = \dot{m}_{dis} h_{dis} + \dot{m}_{l,l} h_{l,l}$$

Continuity

$$\omega \frac{dm}{d\theta} = \dot{m}_{in} - \dot{m}_{out}$$

$$\dot{m}_{in} = \dot{m}_{suc} + \dot{m}_{l,g} + \dot{m}_{oil} \qquad \dot{m}_{out} = \dot{m}_{dis} + \dot{m}_{l,l} \qquad \dot{m} = \rho w A$$

Leakage Flow

Momentum

$$w_l dw_l + \frac{dp}{\rho} + f \frac{w_l^2}{2} \frac{dx}{D_g} = 0$$

$$\dot{m}_l = \rho_l w_l A_g = A_g \sqrt{\frac{p_2^2 - p_1^2}{a^2 \left(\zeta + 2 \ln \frac{p_2}{p_1} \right)}}$$

Oil injection

$$\frac{dT_{oil}}{d\theta} = \frac{h_o A_o \left(T - T_{oil}\right)}{\omega m_{oil} c_{oil}} \qquad T_{oil} = \frac{T - kT_{oil,p}}{1+k}$$

$$k = \frac{\omega m_{oil} c_{oil}}{h_o A_o \Delta\theta} = \frac{\omega d_S c_{oil}}{6 h_o \Delta\theta}$$

Numerical solution, Runge-Kutta IV order solver

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

Ideal Gas

$$T = \left(\gamma - 1\right)\frac{u}{R} \qquad p = \frac{RT}{v}$$

Real gas

$p=f_1(T,v)$ $u=f_2(T,v)$

Wet vapour

$$u = \left(1 - x\right)u_f + xu_g \qquad v = \left(1 - x\right)v_f + xv_g$$

Compressor integral parameters

$$m = m_{in} - m_{out}$$

$$\dot{m} = m z_1 n / 60$$

$$\dot{V} = 60 m / \rho_0$$

$$\dot{m}_t = \frac{\left(F_{1n} + F_{2n}\right) L n z_1 \rho}{60}$$

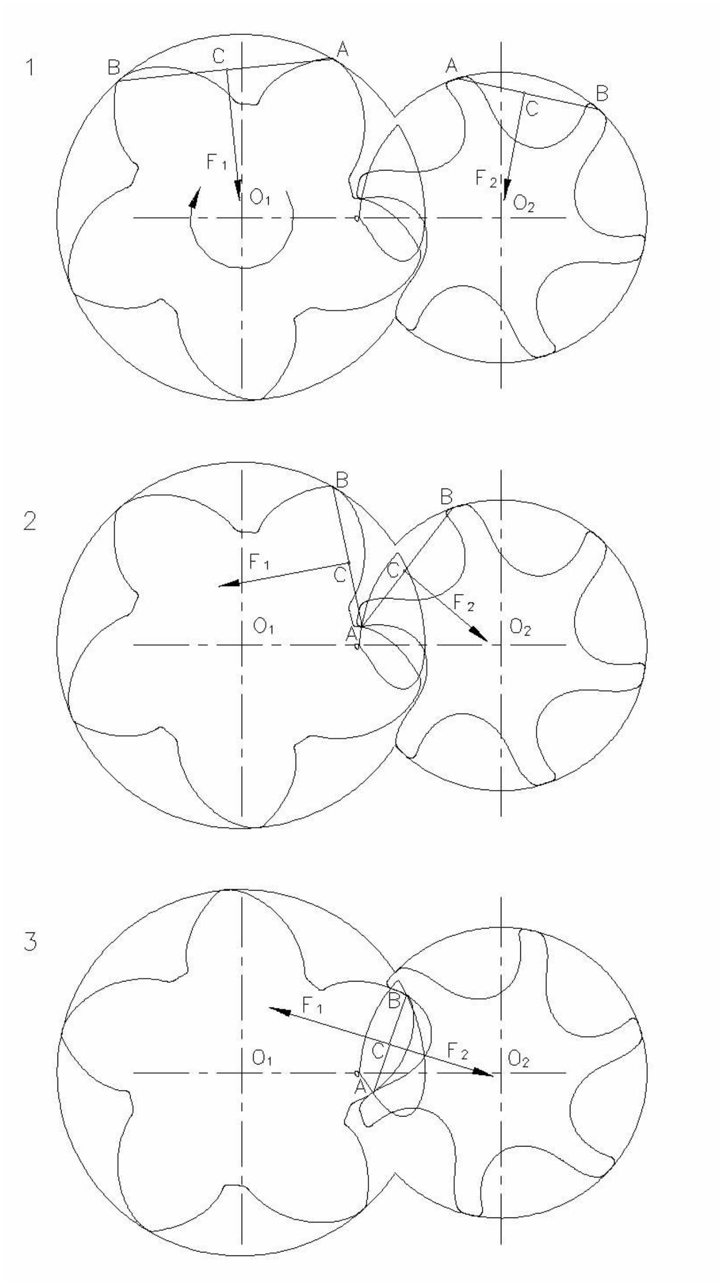
$$W_{ind} = \int_{cycle} V dp$$

$$P_{ind} = \frac{W_{ind} z_1 n}{60}$$

$$W_{sind} = \int_{cycle} \frac{V}{m} dp$$

$$\eta_v = \frac{\dot{m}}{\dot{m}_t} \qquad \eta_t = \frac{W_t}{W_{ind}} \qquad \eta_a = \frac{W_a}{W_{ind}}$$

$$P_{\sin d} = \frac{P}{\dot{V}} \qquad W_t = R T_1 \ln \frac{p_2}{p_1} \qquad W_a = \frac{\gamma}{\gamma - 1} R \left(T_2 - T_1\right)$$



Calculation of pressure loads On compressor rotors

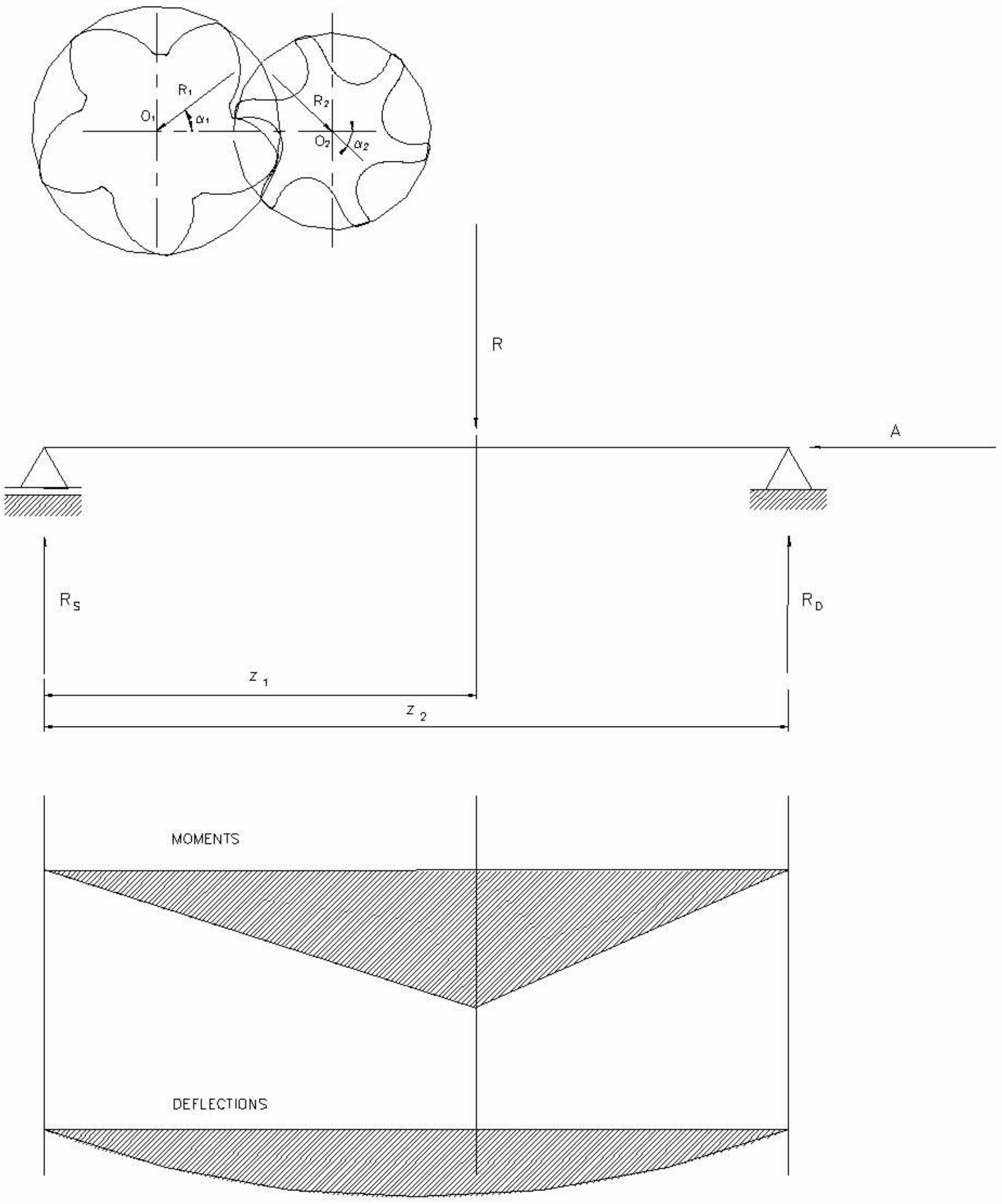
Radial forces

$$R_x = -p\int_A^B dy = -p\left(y_B - y_A\right)$$

$$R_y = -p\int_A^B dx = -p\left(x_B - x_A\right)$$

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

$$F=w_1L+w_2C$$

Box constrained simplex method for efficient and reliable multivariable optimization

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

$$g_i \le x_i \le h_i,\ i = 1, n \quad g_i \le y_i \le h_i,\ i = n+1, m \qquad y_{n+1}, \ldots, y_m$$

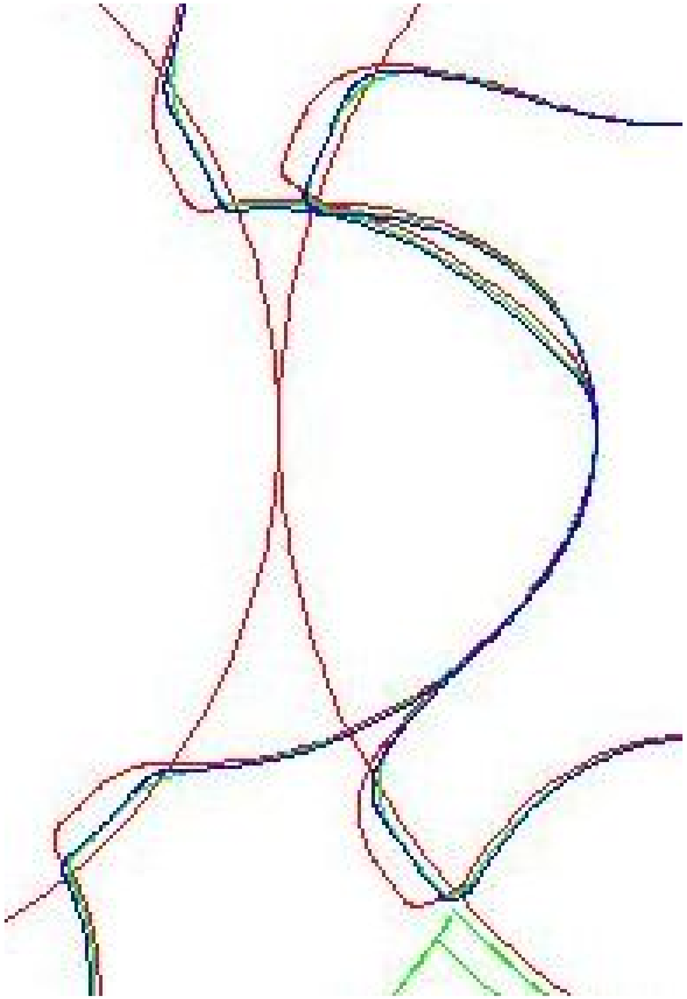
$$f(x^h) = \max\ f(x^1), f(x^2), \ldots, f(x^k)$$

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

$$\overline{x} = \frac{1}{k-1}\sum_{i=1}^{k} x_j^i, \quad x^i \ne x^l \qquad x^r = \overline{x} + \alpha(\overline{x} - x^l)$$

$$x^{r(new)} = 0.5\left[x^{r(old)} + c\overline{x} + (1-c)x^h\right] + (\overline{x} - x^h)(1-c)(2R-1)$$

$$c = \left(\frac{n_r}{n_r + k_r - 1}\right)^{\frac{n_r + k_r - 1}{n_r}}$$



	Dry	Oil-Flooded	Refrig.
r_0 [mm]	2.62	0.74	0.83
r_1 [mm]	19.9	17.8	19.3
r_2 [mm]	6.9	5.3	4.5
r_3 [mm]	11.2	5.5	5.2
Built-in volume ratio	1.83	4.1	3.7
Rotor speed [rpm]	7560	3690	3570
Oil flow [lit/min]	-	12	8
Injection position [º]	-	65	61
Oil temperature [º]	-	33	32

Oil Free

Oil Flooded

Refrigeration

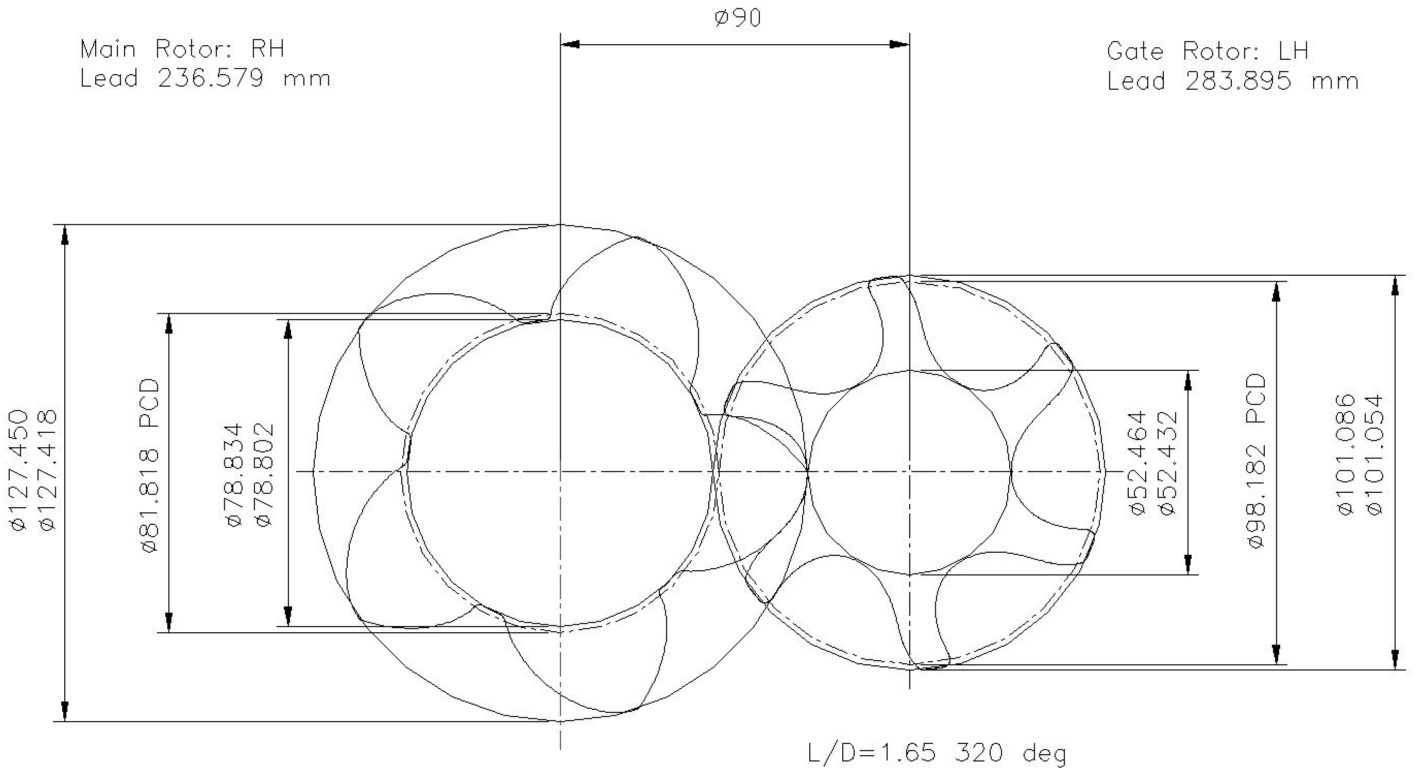
EXAMPLE OF CALCULATION

5-6-128 mm Oil-Flooded Air Compressor

7 m^3/min, max 10 m^3/min at 8 bar (abs)

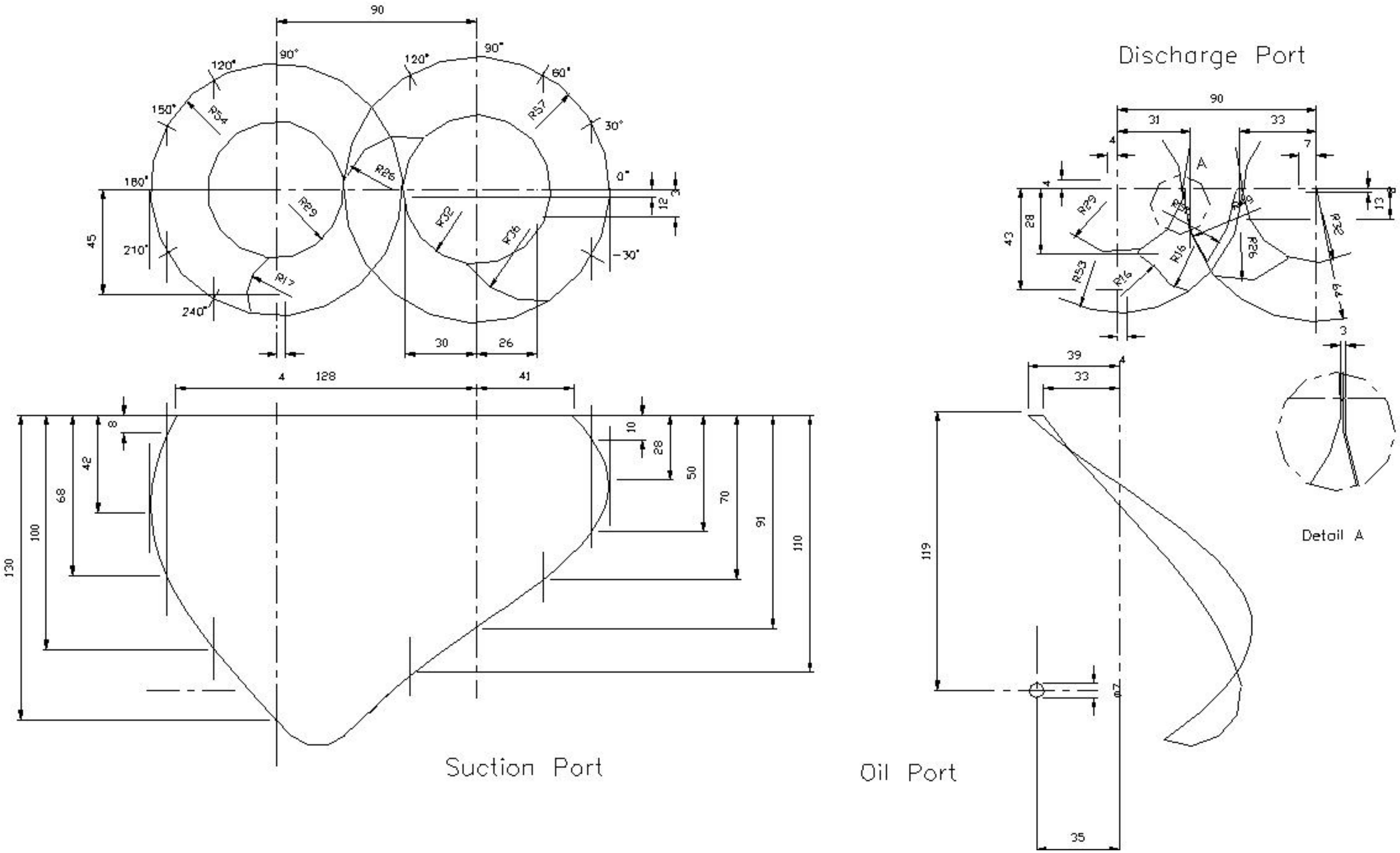
5-14 bar (abs), max 15 bar (max)

EXAMPLES OF CALCULATION



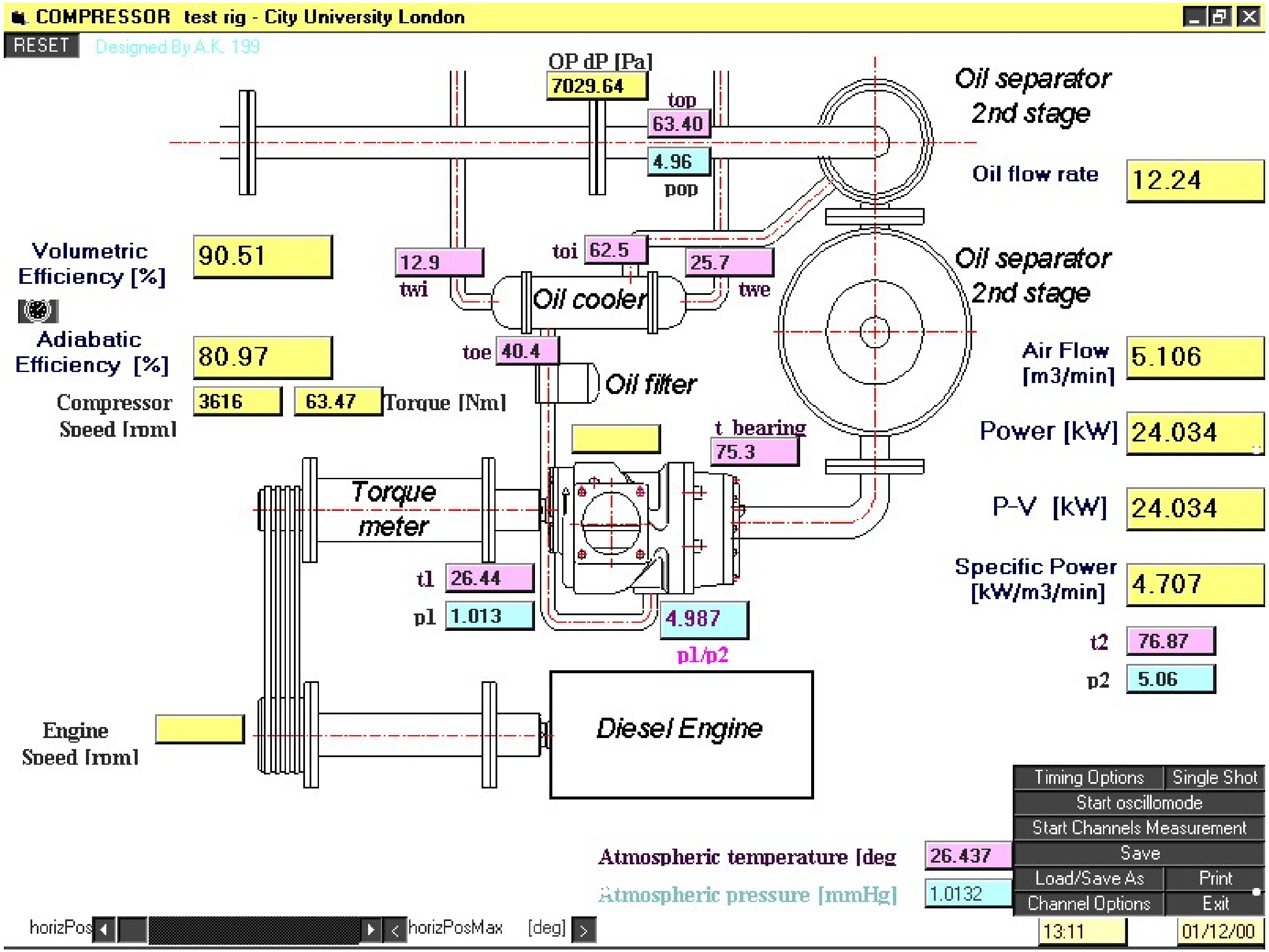
Rotors optimized for oil flooded operated air compression

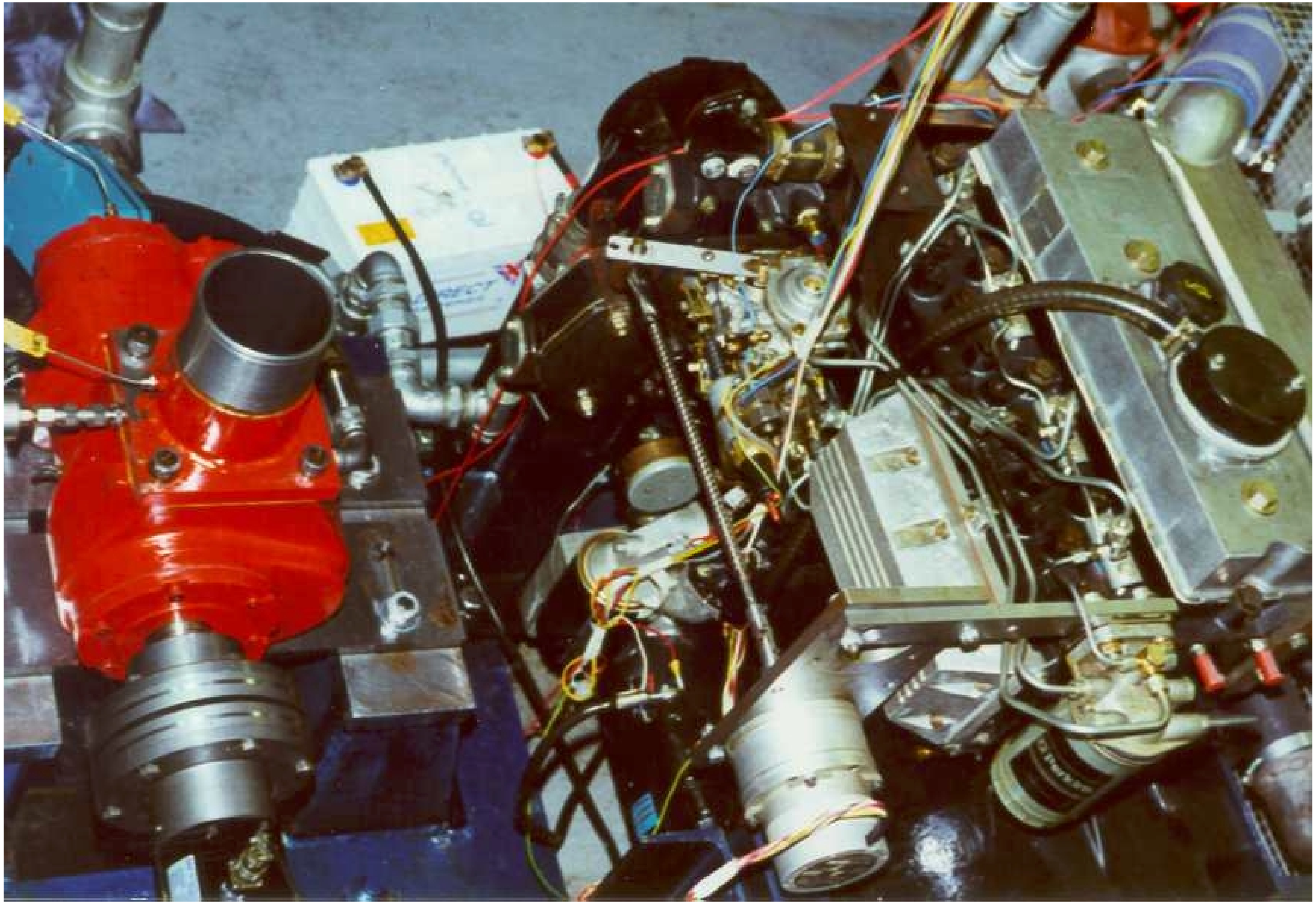
5/6-128 mm, L/D 1.65
Displacement 1.56 l/rev
Interlobe sealing line 0.13 m
Blow-hole area 1.85 mm^2



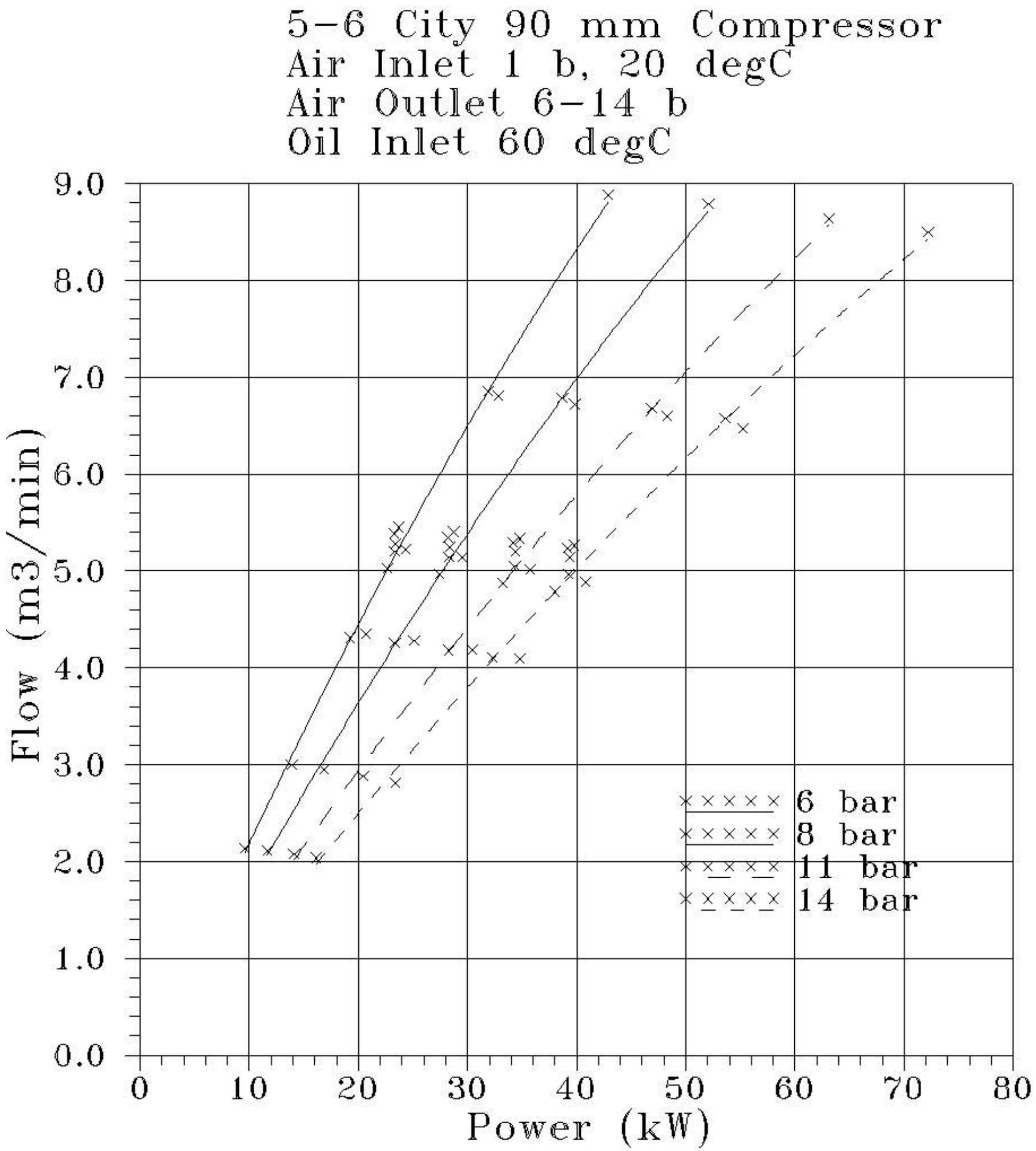
CAD Interface: Compressor ports

Experimental verification of the model





Compressor in the test bed



Comparison of the calculated and test results Flow-Power

EXAMPLES OF 3-D CFD CALCULATION

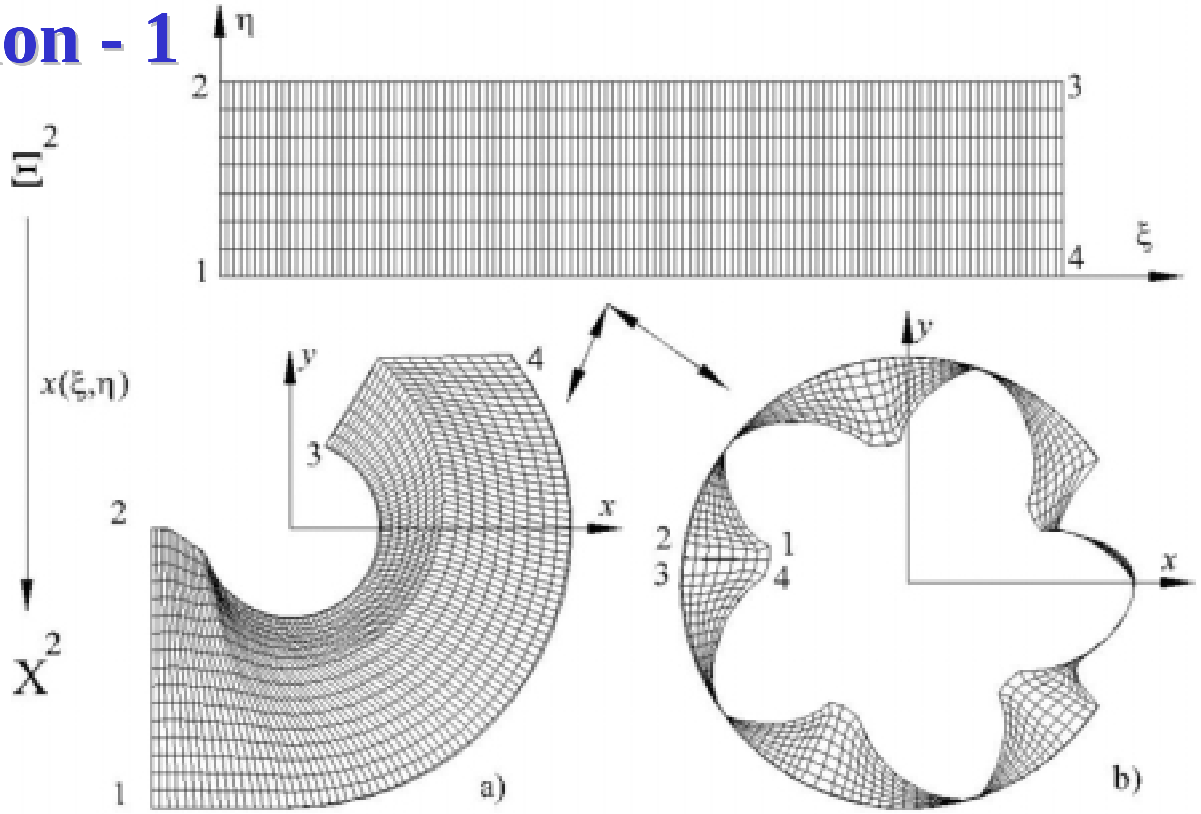
Majority of design problems can be solved by the one-dimensional approach, some of them require the two dimensional calculation, however, there are situations where 3-D CFD must be applied

Such are

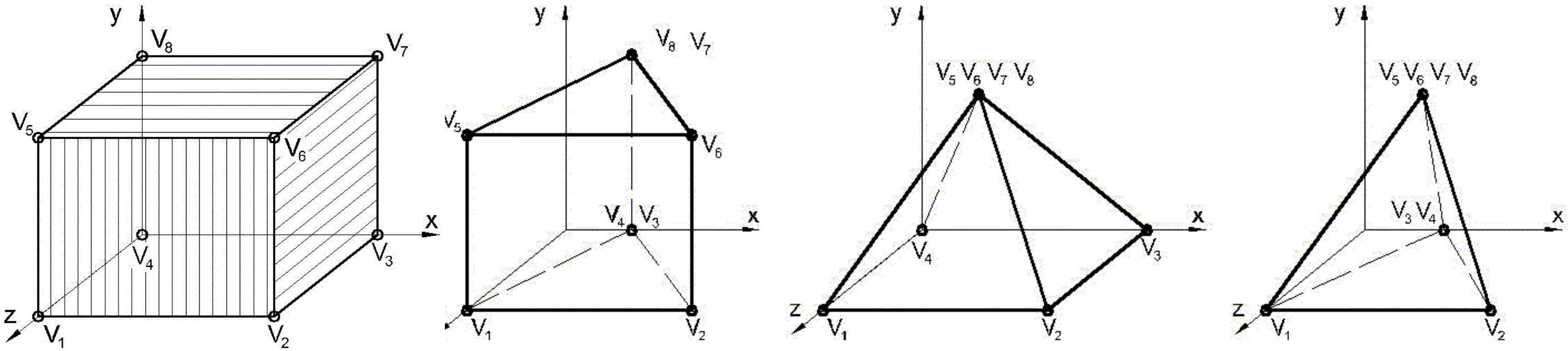
Oil flow distribution,
Fluid-Solid Interaction

Grid generation - 1

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



- **Cell shape**



Grid generation - 2

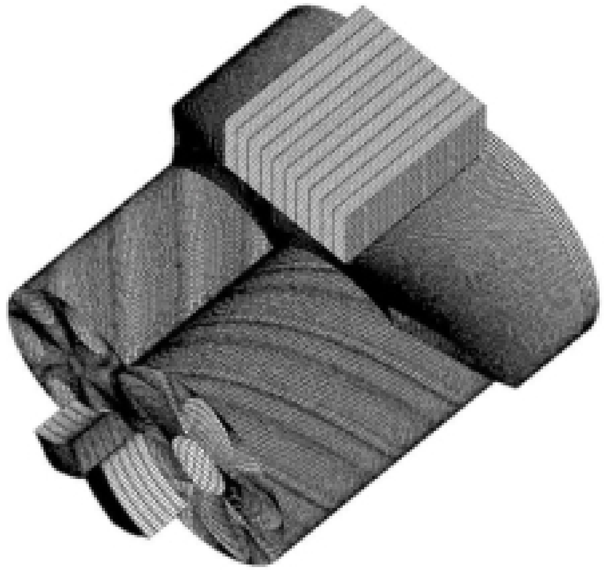
- Grid topology strongly affects accuracy, efficiency and ease of calculation
- Full structured block generated hexahedral 3D-O mesh
- Screw compressor sub-domains:

- **Male rotor** - **Female rotor**

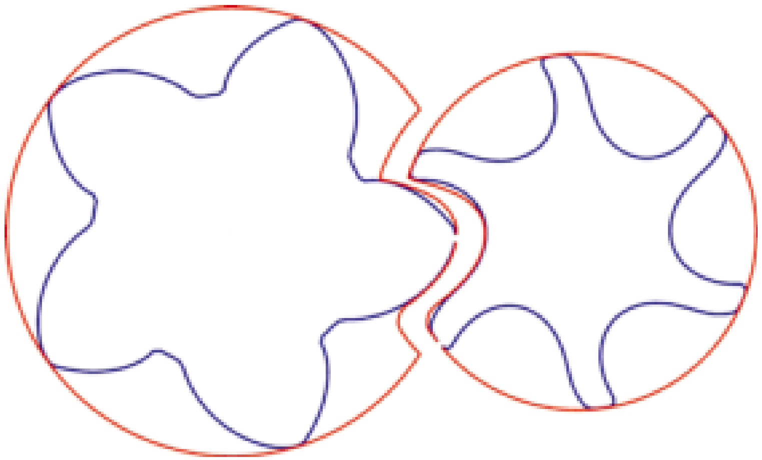
- End clearances

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,



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Screw Compressor FSI calculations

Comet

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

$$\frac{d}{dt}\int_V \rho dV + \int_S \rho(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = 0$$

$$\frac{d}{dt}\int_V \rho\mathbf{v} dV + \int_S \rho\mathbf{v}(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = \int_S \mathbf{T}\cdot d\mathbf{s} + \int_V \mathbf{f}_b dV$$

$$\frac{d}{dt}\int_V \rho h dV + \int_S \rho h(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = \int_S \mathbf{q}_h \cdot d\mathbf{s} +$$
$$\int_V s_h dV + \int_V (\mathbf{v}\cdot \mathrm{grad}\, p + \mathbf{S} : \mathrm{grad}\,\mathbf{v})\, dV - \int_S \mathrm{p}\mathbf{v}_s \cdot d\mathbf{s} + \frac{d}{dt}\int_V p dV$$

$$\frac{d}{dt}\int_V \rho c_o dV + \int_S \rho c_o(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = \int_S \mathbf{q}_{c_o}\cdot d\mathbf{s} + \int_V S_{c_o} dV$$

$$\frac{d}{dt}\int_V dV - \int_S \mathbf{v}_s \cdot d\mathbf{s} = 0$$

$$\rho = \rho(p,T), \quad e = e(p,T)$$

$$\mathbf{T} = 2\mu\dot{\mathbf{D}} - \frac{2}{3}\mu\, \mathrm{div}\,\mathbf{v}\mathbf{I} - p\mathbf{I}$$

$$\mathbf{T} = 2\eta\mathbf{D} + \lambda\, \mathrm{div}\,\mathbf{w}\mathbf{I} - (3\lambda + 2\eta)\alpha(T - T_r)\mathbf{I}$$

$$\mathbf{q}_{\mathrm{h}} = \kappa\, \mathrm{grad}\, T$$

$$\mathbf{q}_{\mathrm{c_o}} = \rho D_o\, \mathrm{grad}\, c_o$$

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

$$\frac{d}{dt}\int_V \rho k dV + \int_S \rho k(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = \int_S \mathbf{q}_k \cdot d\mathbf{s} + \int_V (P - \rho\varepsilon) dV,$$

$$\frac{d}{dt}\int_V \rho\varepsilon dV + \int_S \rho\varepsilon(\mathbf{v}-\mathbf{v}_s)\cdot d\mathbf{s} = \int_S \mathbf{q}_\varepsilon \cdot d\mathbf{s} + \int_V (C_1 P\frac{\varepsilon}{k} - C_2\rho\frac{\varepsilon^2}{k} + C_3\rho\varepsilon\nabla\cdot\mathbf{v}) dV$$

Thermodynamic properties of real fluids

- *p*-*v*-*T* equation
  compressibility factor *z*

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

- *z* is assumed to change linearly
  with pressure err<2%

$$z = p \cdot B_1 + B_2$$

- Antoine equation for saturation
  temperature

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

- Clapeyron equation for latent heat

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

- Specific heat for constant pressure

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

- Density of mixture

$$\rho = \frac{1}{\frac{1 - co_2}{\rho_v} - \frac{co_2}{\rho_l}}$$

- Coefficient in the pressure correction
  equation

$$C_\rho = \left(\frac{d\rho}{dp} \right)_T = \left(\frac{1}{zRT} - \frac{\rho_v \cdot b_1}{z} \right) \cdot \frac{\rho_v}{\rho}$$

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

$$m_o = m \cdot C_o$$

$$\mathbf{f}_{drag} = -\frac{1}{2} \rho A_o C_{drag} \left| \mathbf{v}_o - \mathbf{v} \right| (\mathbf{v}_o - \mathbf{v})$$

- Energy source due to heat transfer
  between working fluid and oil is:

$$\dot{Q}_{con} = m_o C_{p_o} \frac{dT}{dt} \approx m_o C_{p_o} \frac{T^k - T^{k-1}}{\delta t}$$

- **Liquid phase** is assumed to be an active 'species'

- Mass source
  evaporated/condensed mass

$$\dot{m}_L = \frac{m \cdot C_{pm} \cdot (T - T_s)}{h_L}$$

- Energy source
  energy of evaporation/condensation

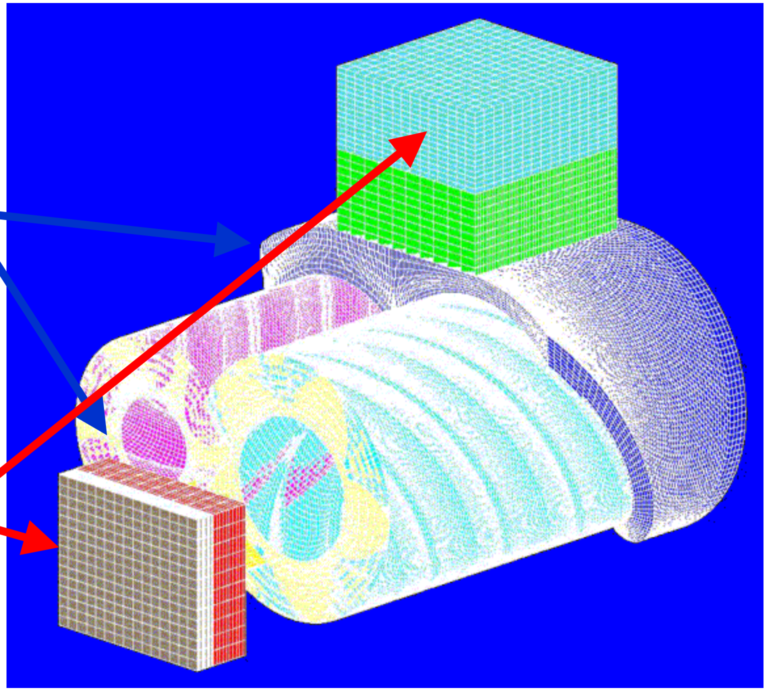
$$\dot{Q}_{mass} = h_L \frac{dm_L}{dt} \approx h_L \frac{m_{ol} - m_{ol}^{\ s}}{\delta t} = h_L \dot{m}_L$$

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

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

- Energy equation corrected by energy source to update energy balance

$$\dot{Q}_{add} = h_{add} \left(\frac{d\dot{m}_{add}}{dt}\right)_{p=const} = \dot{m}_{add} \cdot h_{add}$$

Screw Compressor performance

- Volume flow (inlet and outlet)

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

- Mass flow (inlet, outlet and oil)

$$\dot{m} = \sum_{t=t_{start}}^{t_{end}} \dot{V}_f^{(t)} \cdot \overline{\rho}^{(t)} \quad [kg/\sec]$$

- Boundary forces

$$F_x = p_b * A_{xb}; \quad F_y = p_b * A_{yb}; \quad F_z = p_b * A_{zb}$$

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

- Restraint Forces and Torque

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

- Compressor shaft power

$$P = 2 \cdot \pi \cdot n \cdot (T_M + T_F) \quad [W]$$

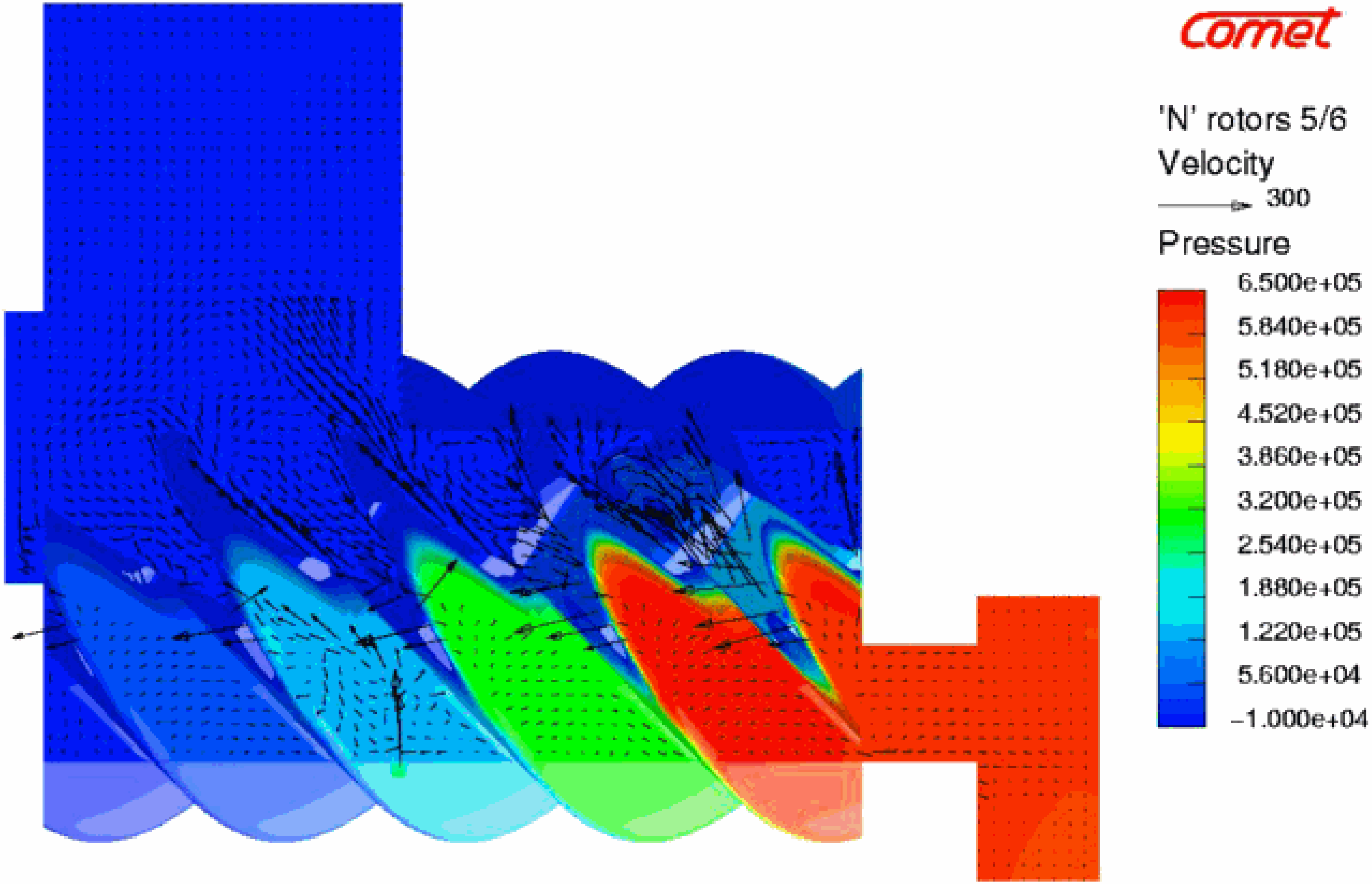
- Specific power

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

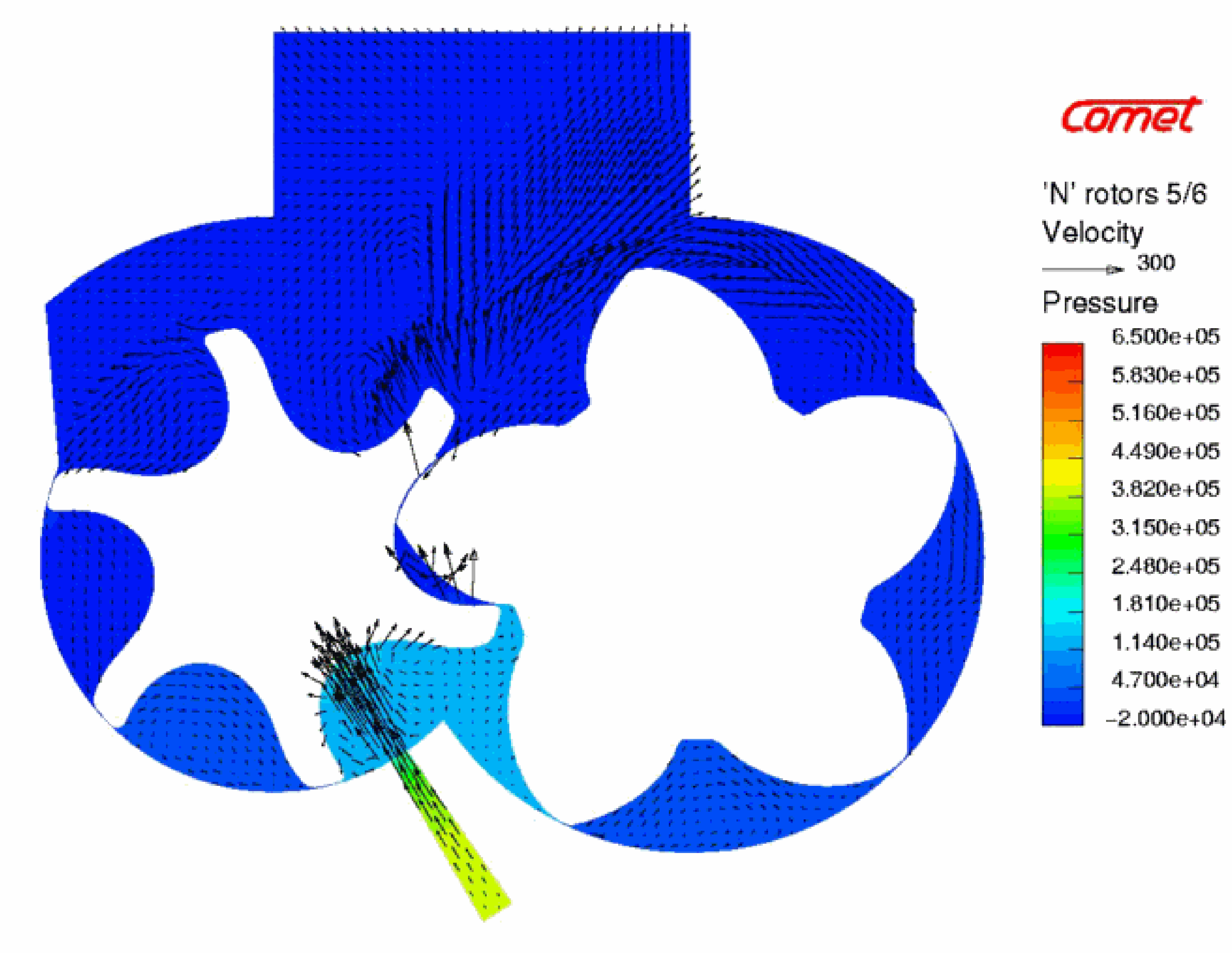
- Efficiency
  Volumetric and adiabatic

$$\eta_v = \dot{V} / V_d; \quad \eta_i = P_{ad} / P$$

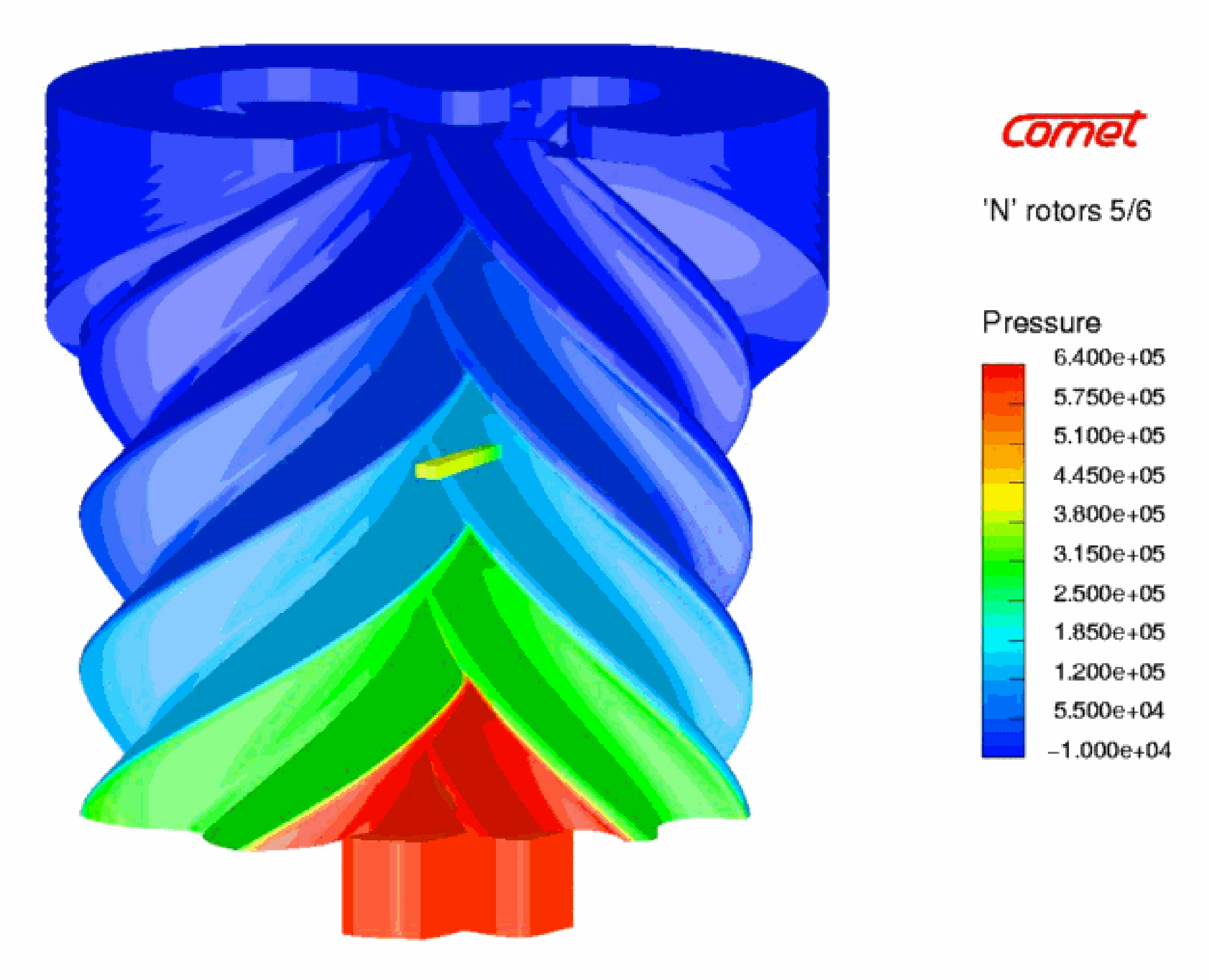
Oil injected - Pressure in axial section



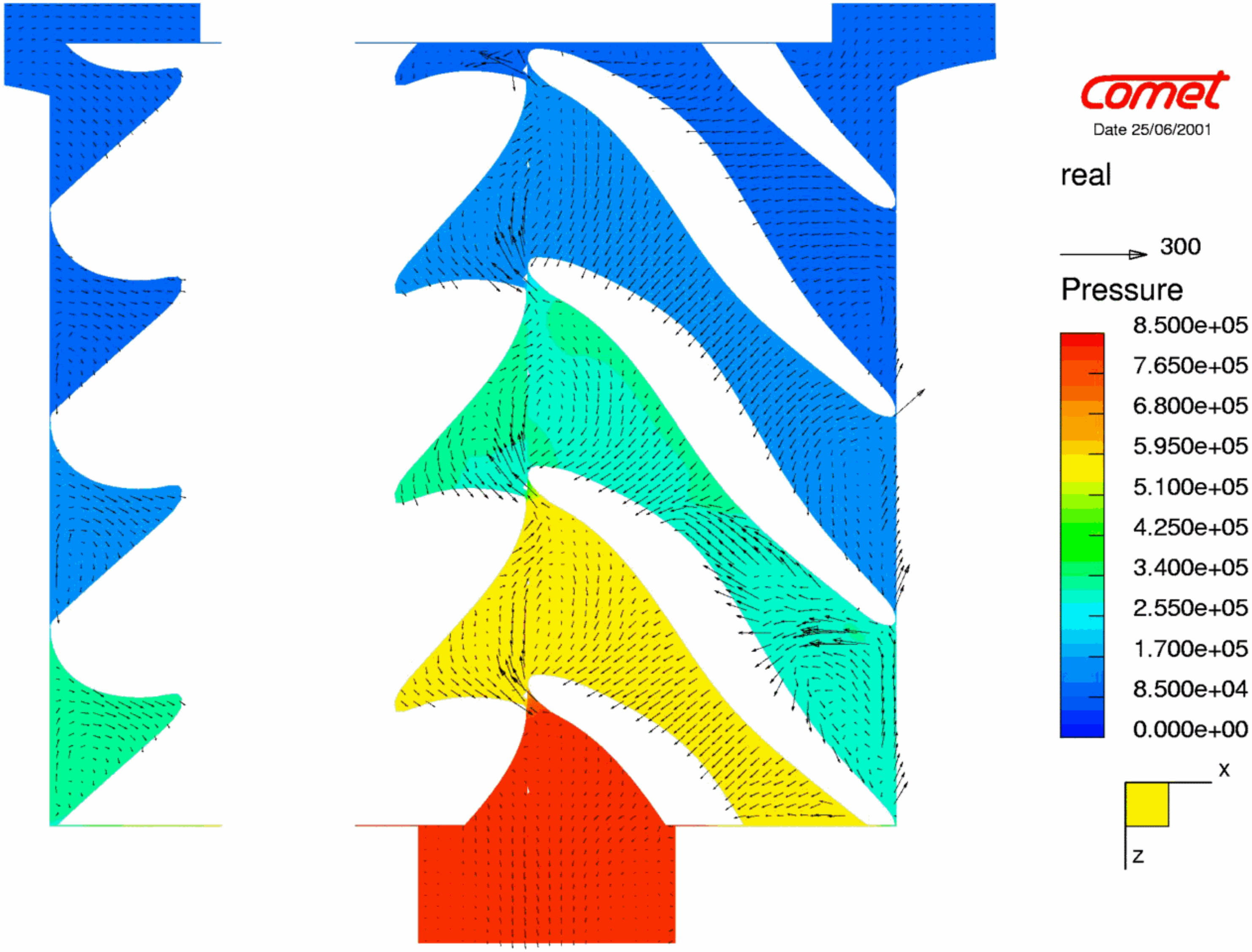
Oil injected - Pressure and velocity



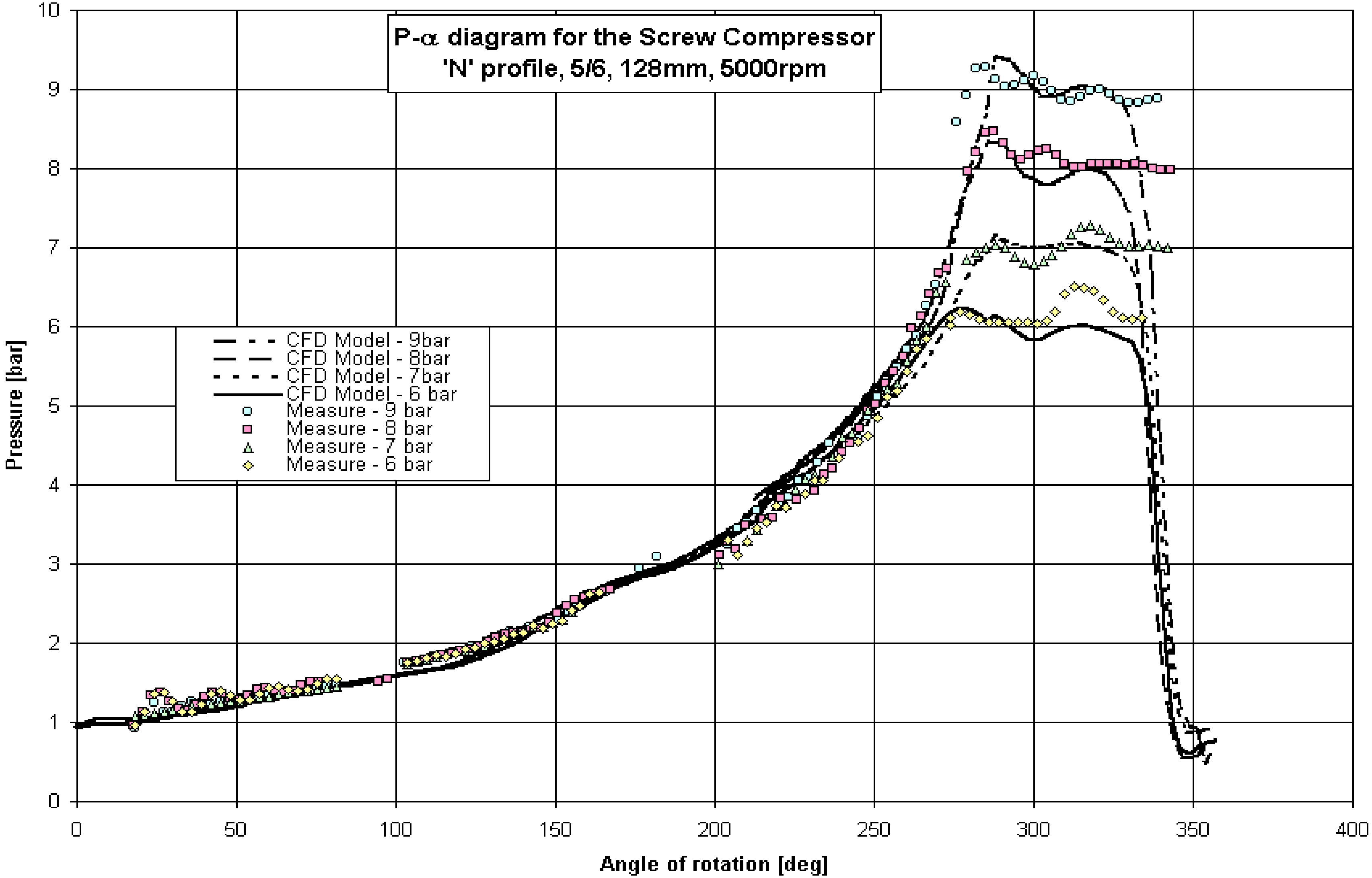
Oil injected - Pressure 3D view



Real fluid - Ammonia – pressure



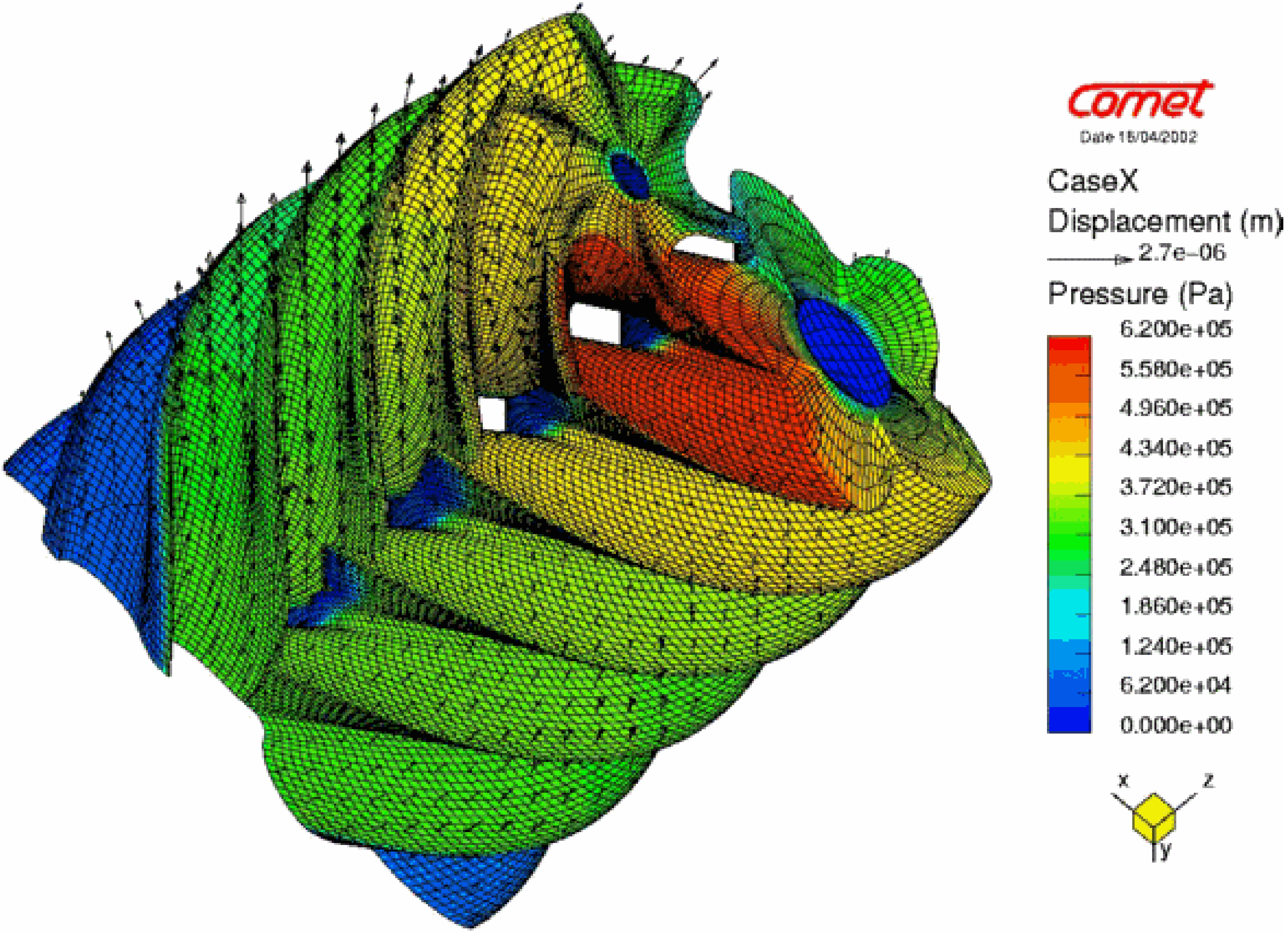
Experimental verification – P-α diagram



Oil injected – Deformation Pressure1

P_{inl}=1 b P_{out}=7 b n=5000 rpm

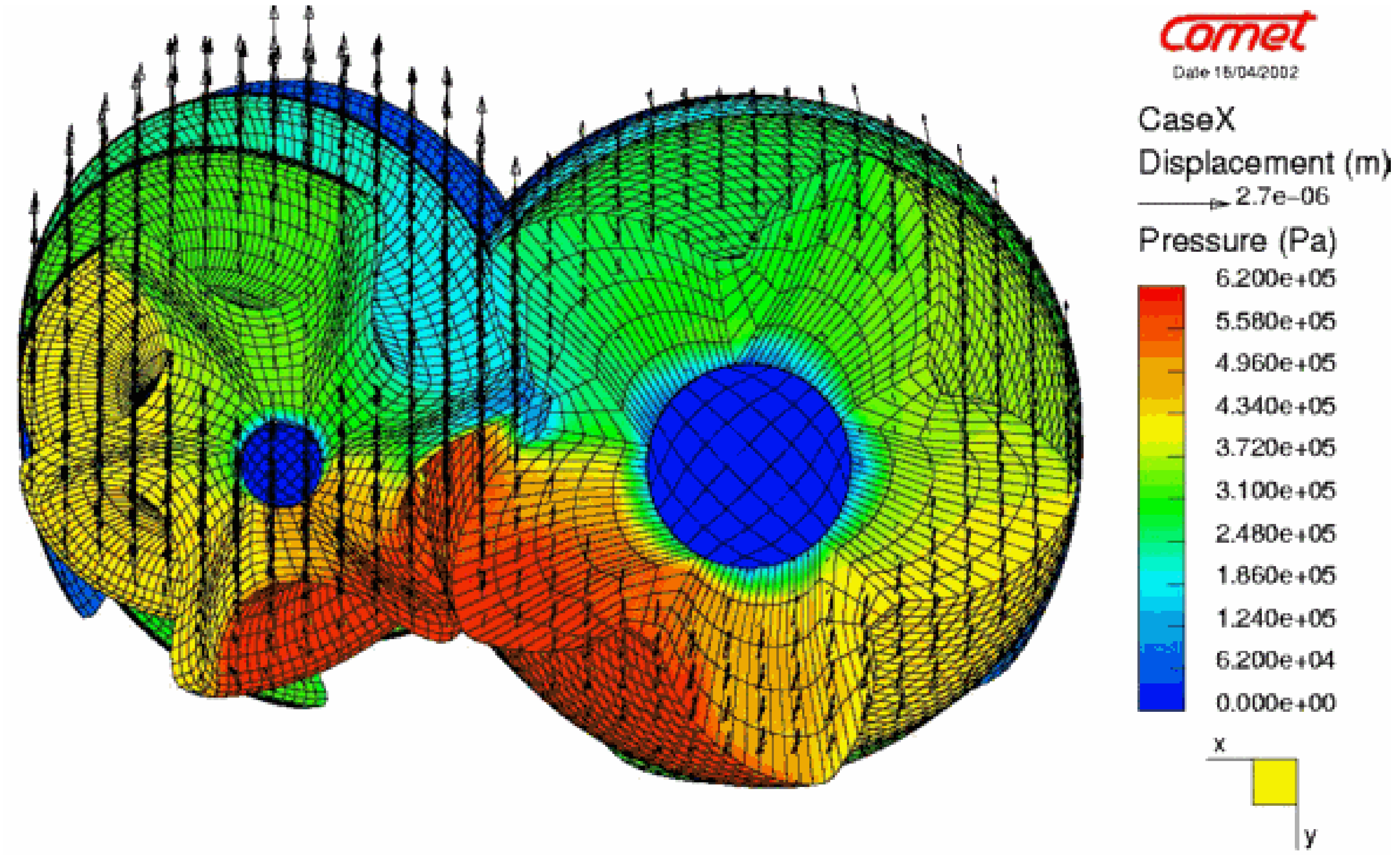
t_{inl}=20 ^oC t_{out}=40 ^oC



Oil injected – Deformation Pressure-1

P_{inl}=1 b P_{out}=7 b n=5000 rpm

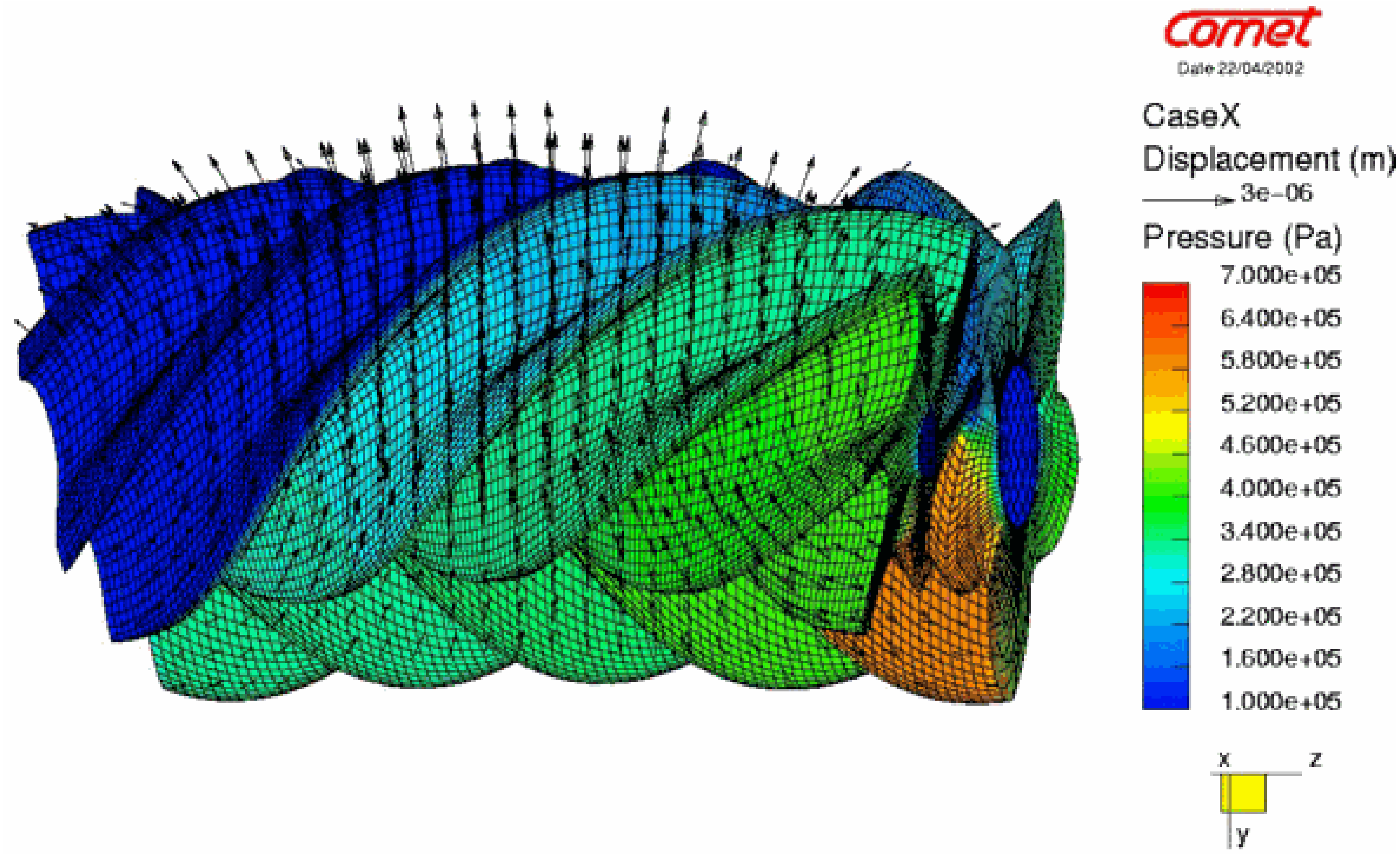
t_{inl}=20 °C t_{out}=40 °C



Oil injected – Deformation Pressure-2

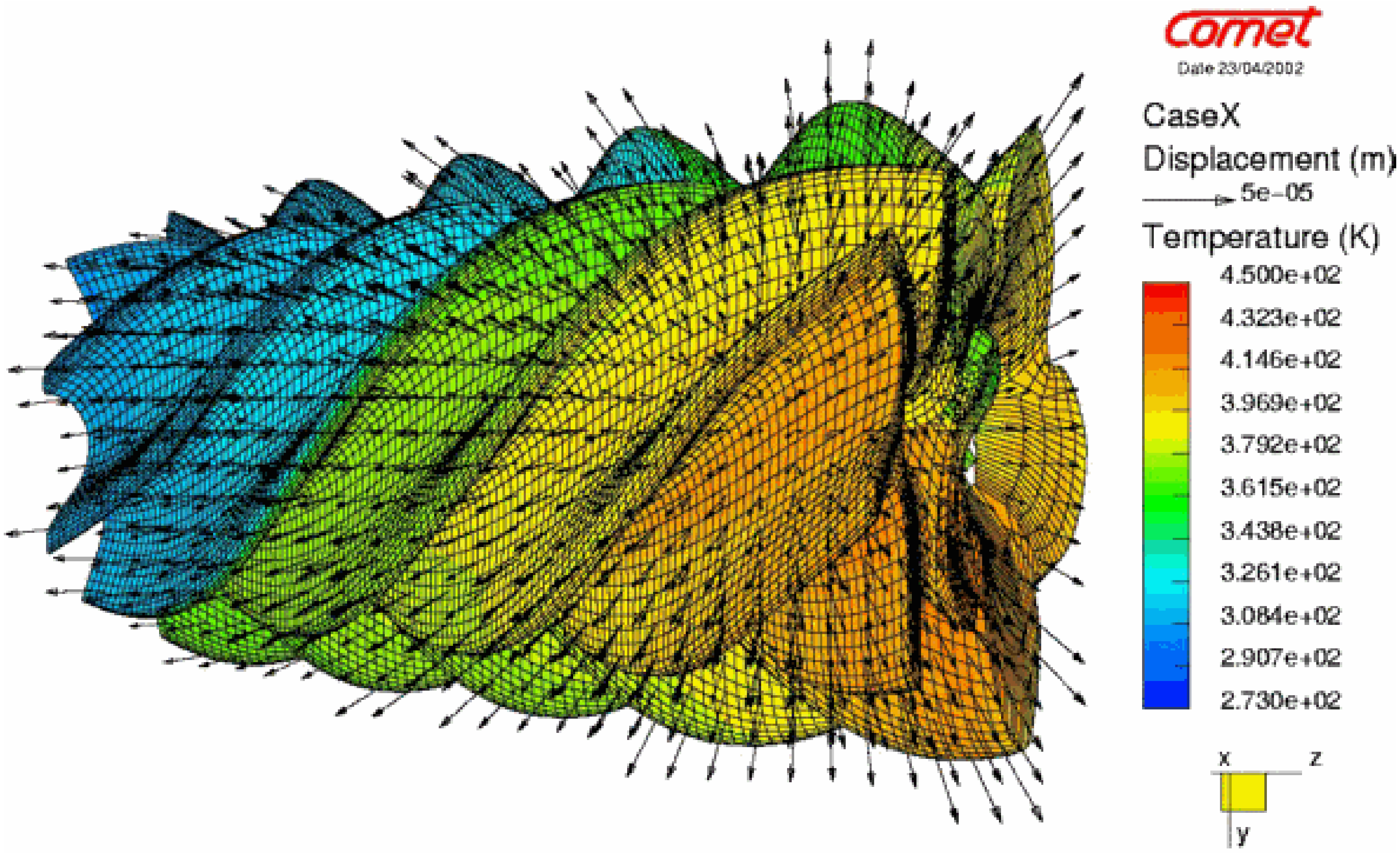
P_{inl}=1 b P_{out}=7 b n=5000 rpm

t_{inl}=20 ºC t_{out}=40 ºC mag=20,000x



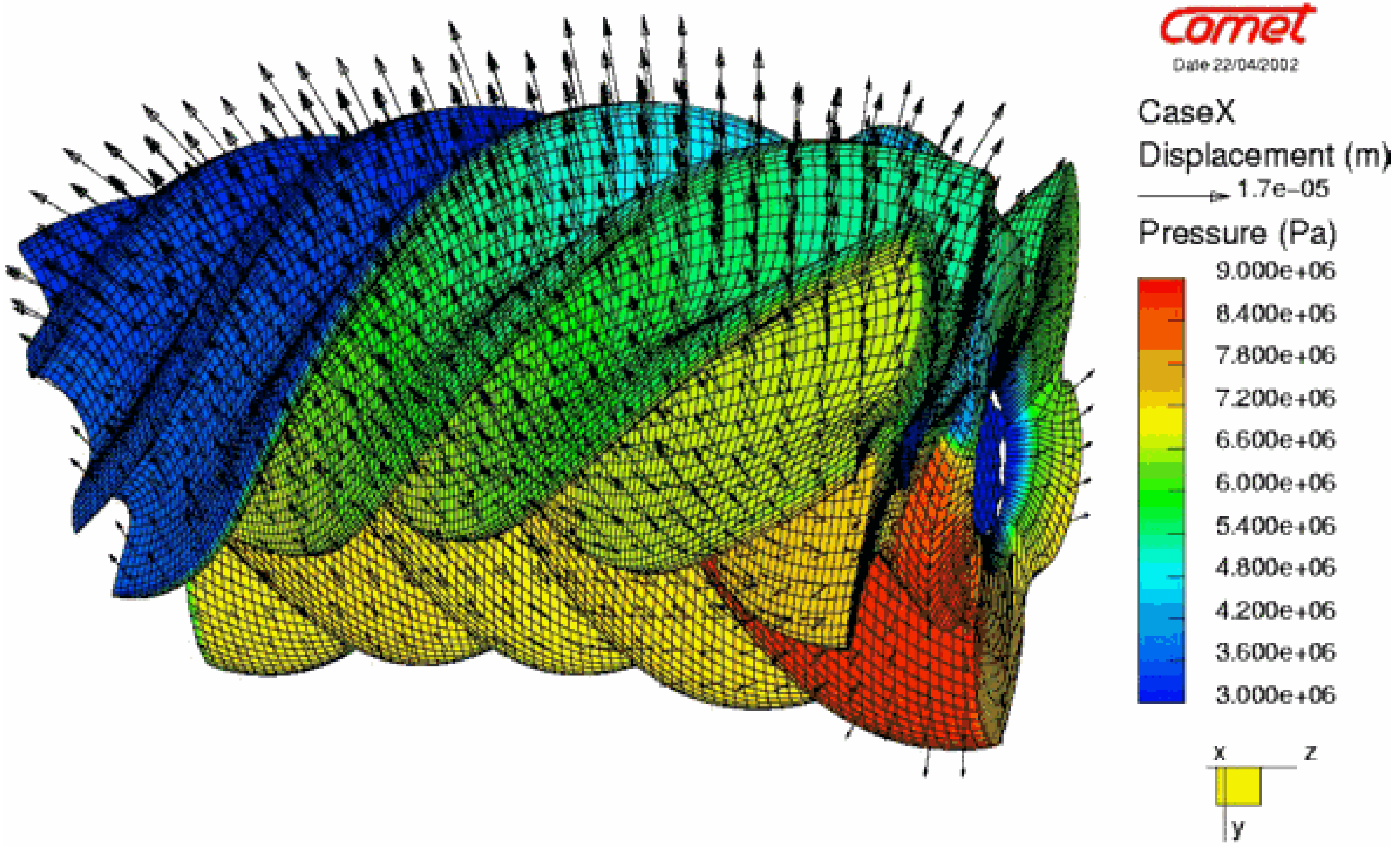
Oil injected – Deformation Temperature

P_{inl}=1 b P_{out}=3 b n=5000 rpm
t_{inl}=20 ºC t_{out}=150ºC mag=1,000x



Oil injected – Deformation Pressure+Temperature

P_{inl}=30 b P_{out}=90 b n=5000 rpm
t_{inl}=0 ºC t_{out}=40 ºC mag=2,000x



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:

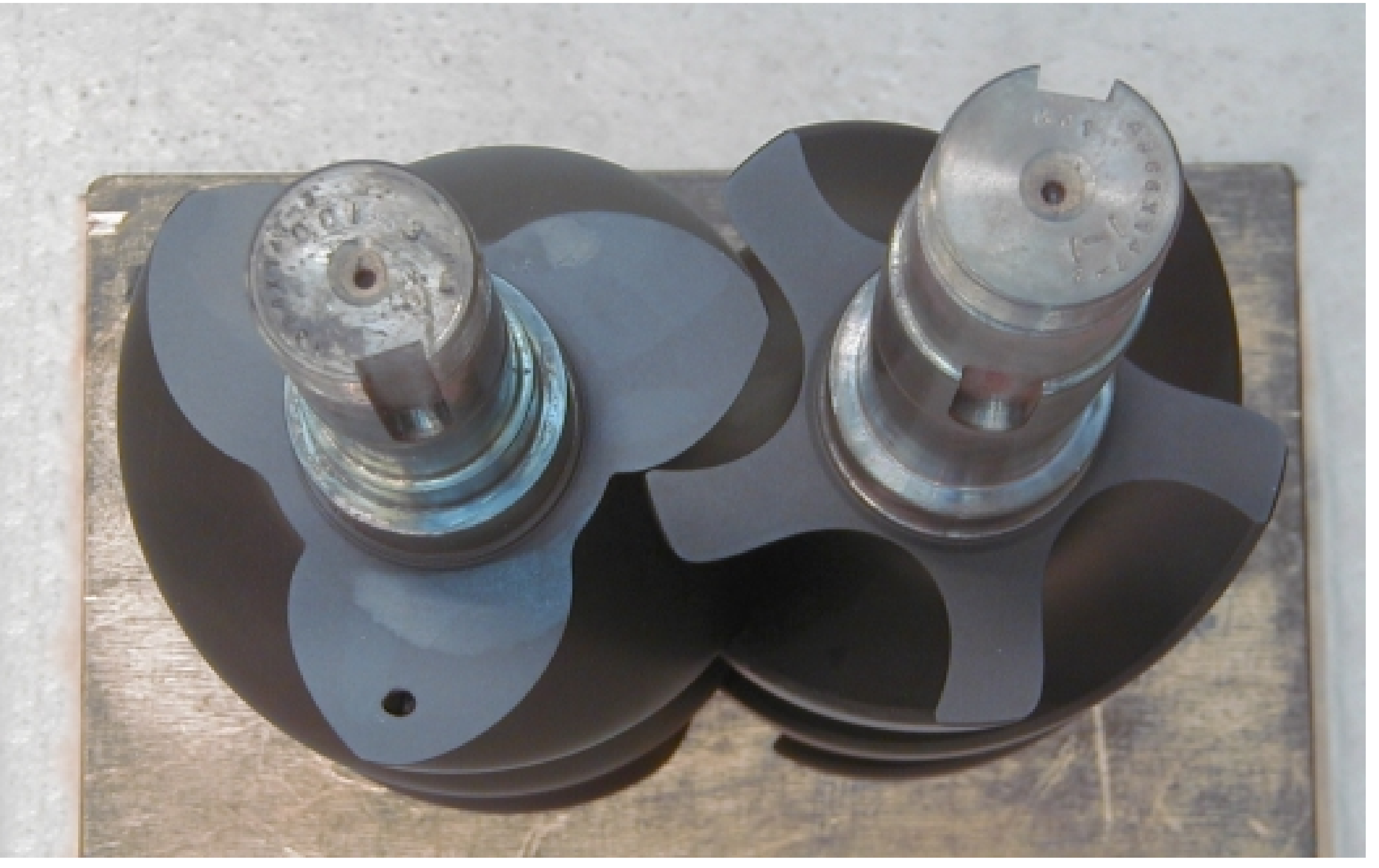
Delivery: 350-700 and 700-1000 m^3/h
Working pressure: 1-2.5 (2.7) bar

Volumetric efficiency 90 % +
Low specific power
Simple, reliable and compact machine

3/5 lobe rotors

Large displacement

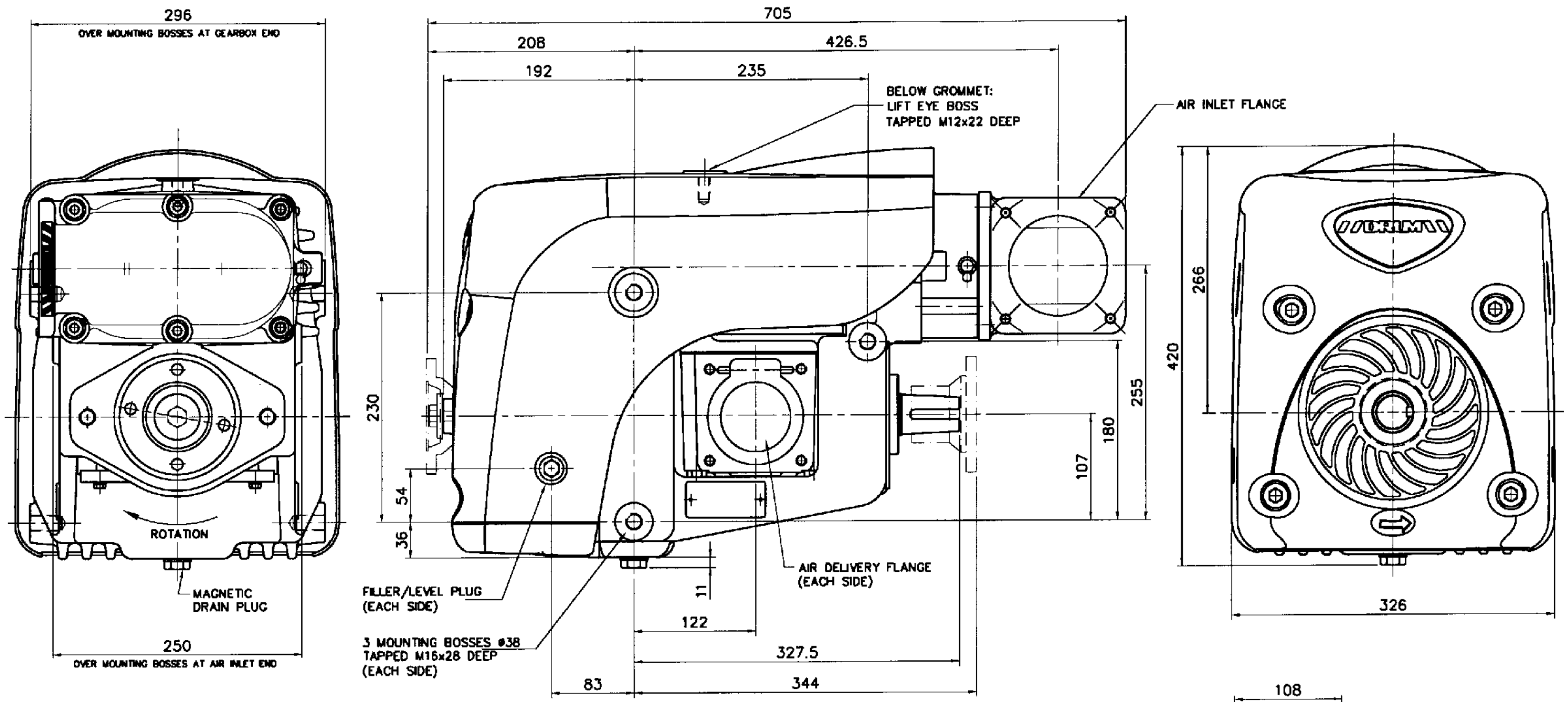
Convenient gear ratio 5/3=1.67



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

XK18 Screw Compressor General Arrangement



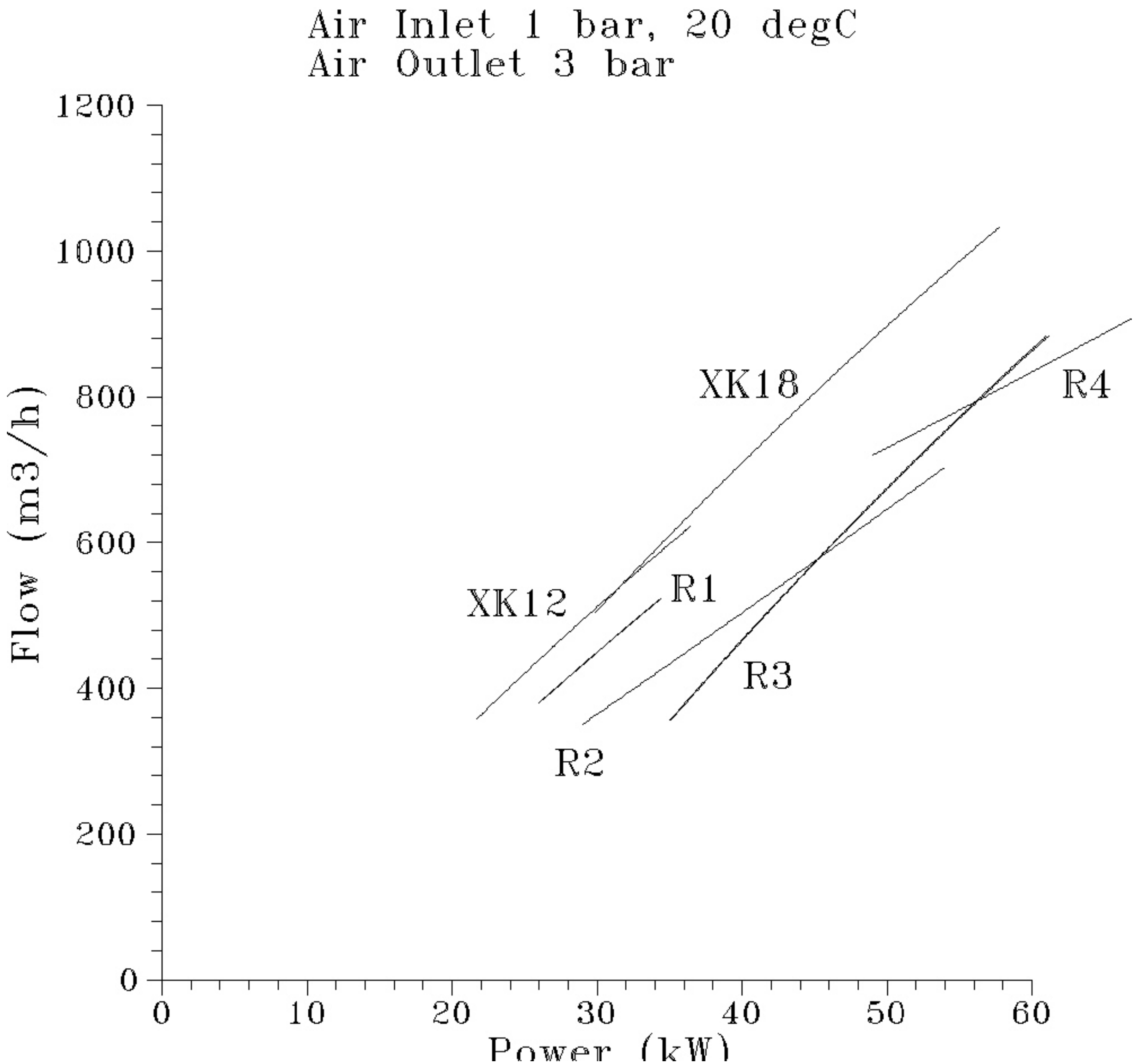
New design, fully customized by the manufacturer

New rotors,

New, improved compressor

New concept, better screw compressor compared with competition

Comparison of Test Results



R1- GHH C80

R2- Drum D9000

R3- Mouvex Typhoon

R4- GHH CS-1000

Screw Compressor Family for Oil-Flooded Operation

Design aims:

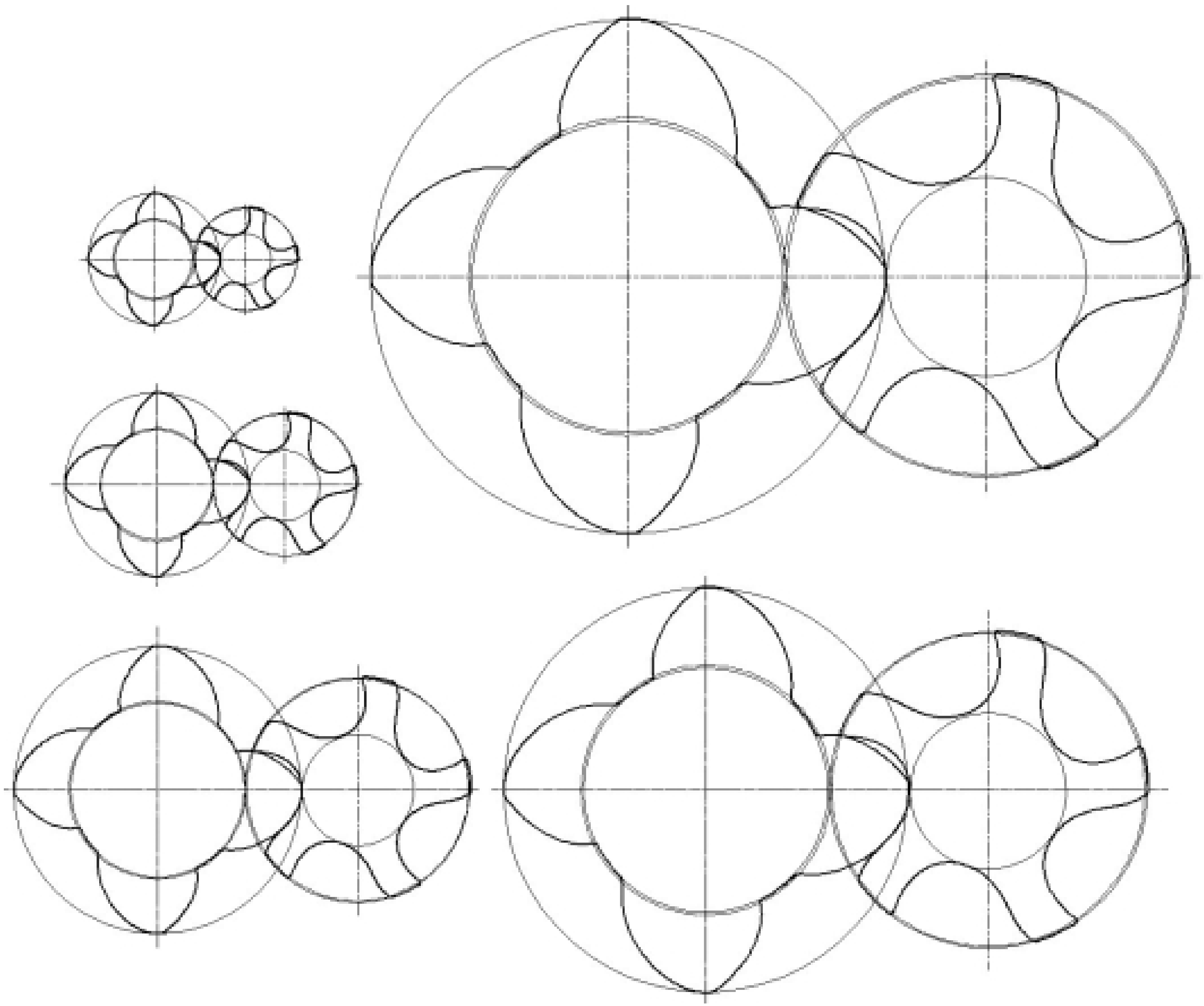
Delivery: 0.6-60 m^3/min
Working pressure: 5-13 bar

Volumetric efficiency 90 % +
Low specific power
Simple, reliable and compact machine

4/5 Rotors

Reasonable efficiency for
moderate pressure ratios, at least the same as for 4/6 rotors

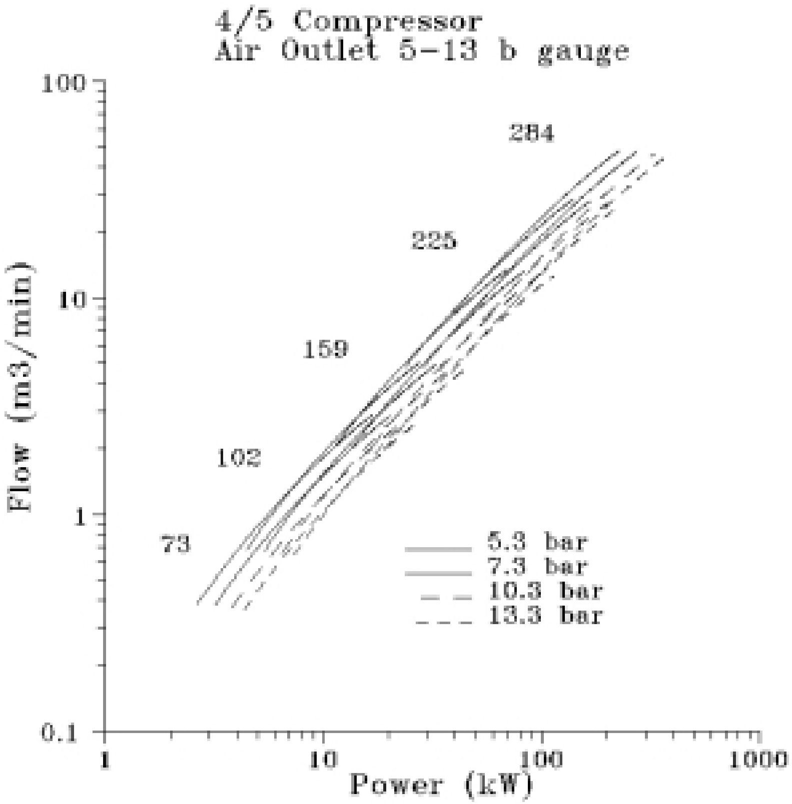
Improved capacity and efficiency
Lower power consumption
Reduced manufacturing cost

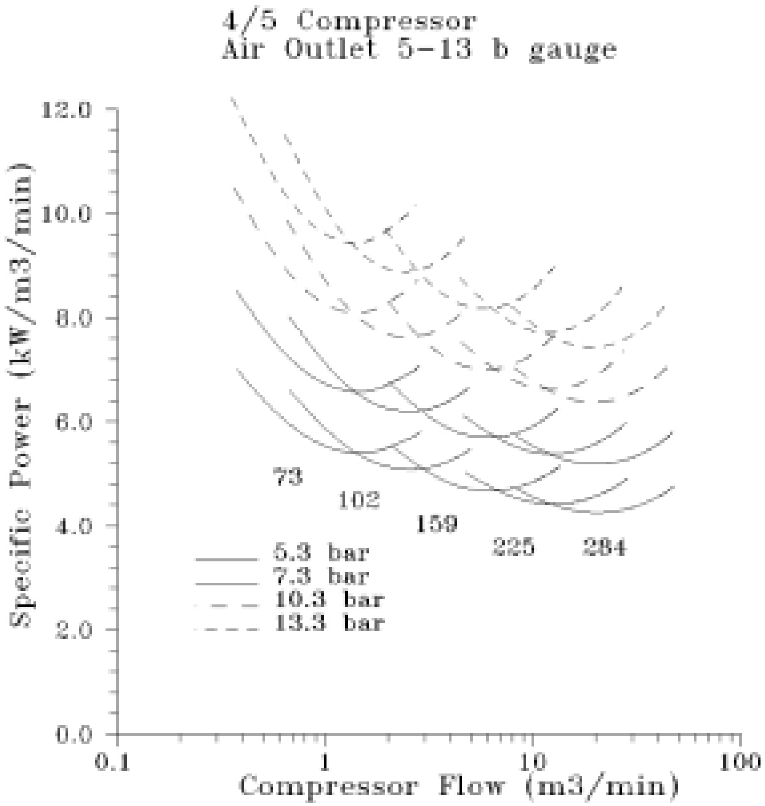


Five compressors, rotor diameters:

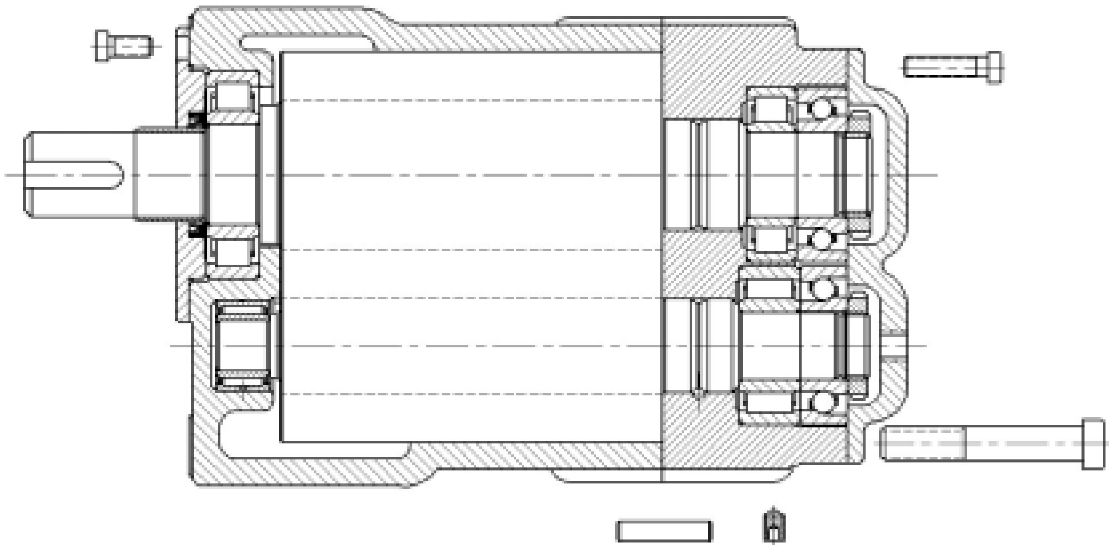
74, 102, 159, 225 and 285 mm, L/D 1.55

Performance of the Compressor Family





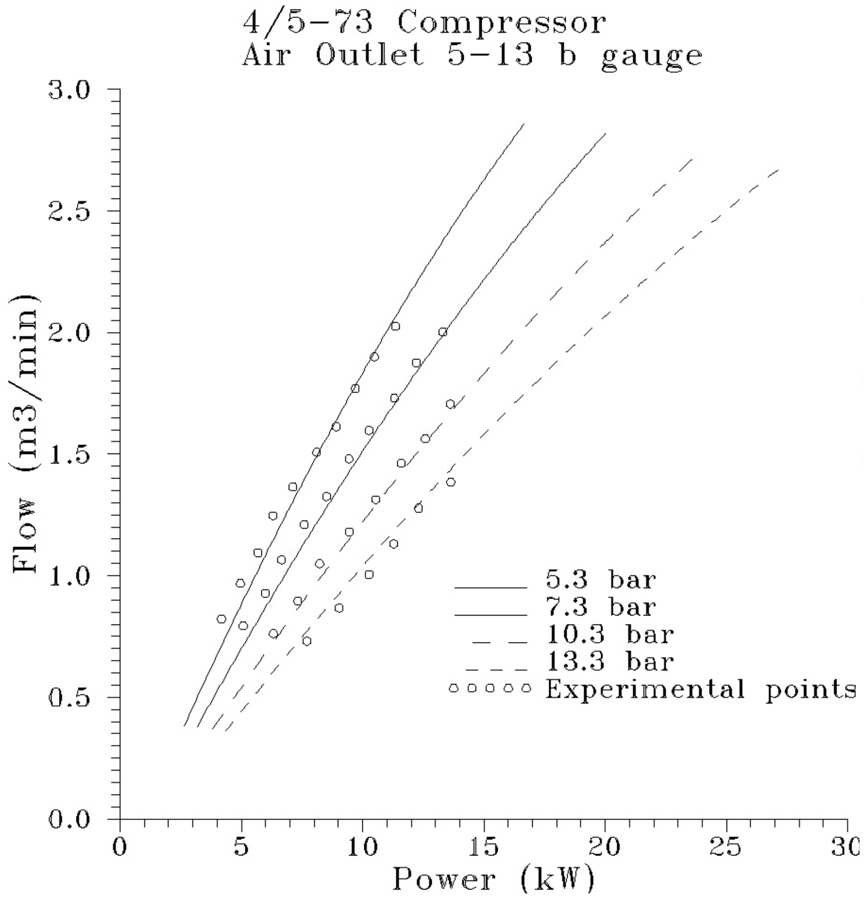
Proven design, fully customized to the the manufacturer’s needs

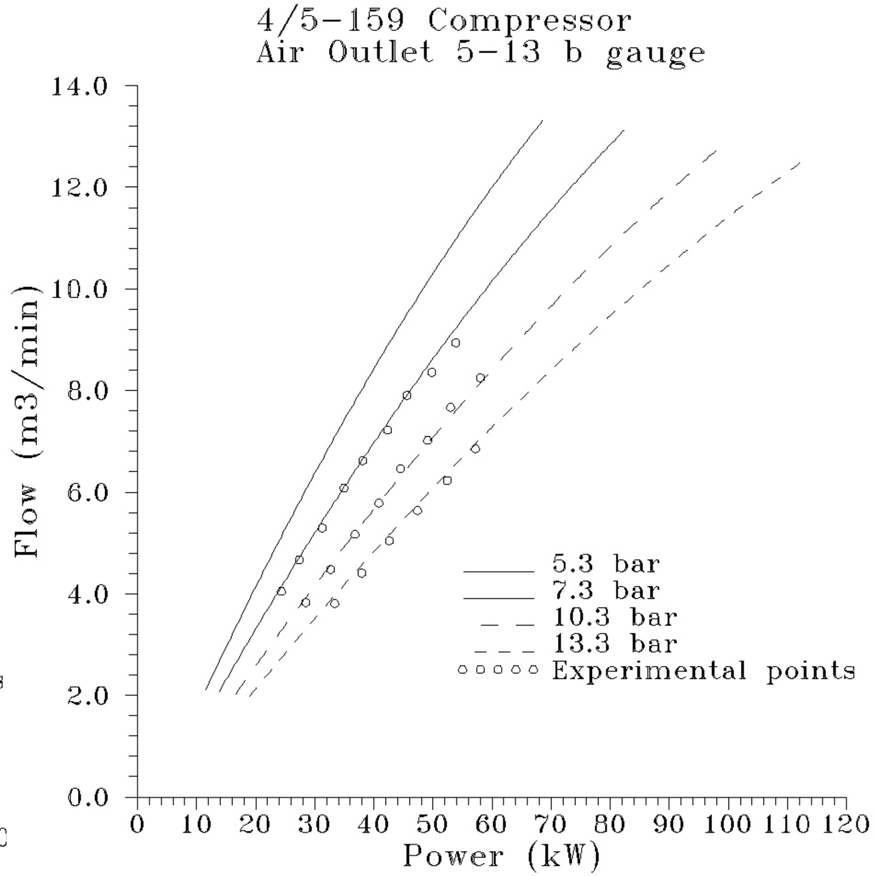


New rotors,

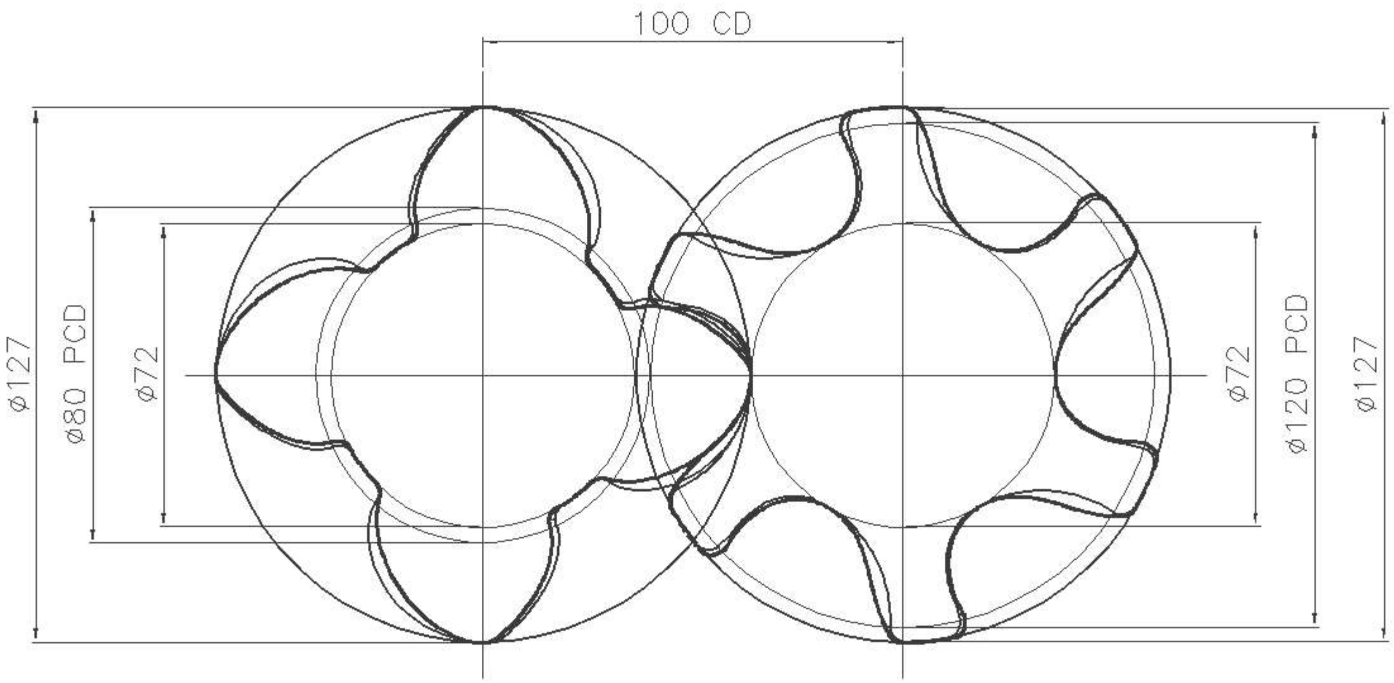
New, improved compressor

Test Results, Compressors 73 and 159 mm





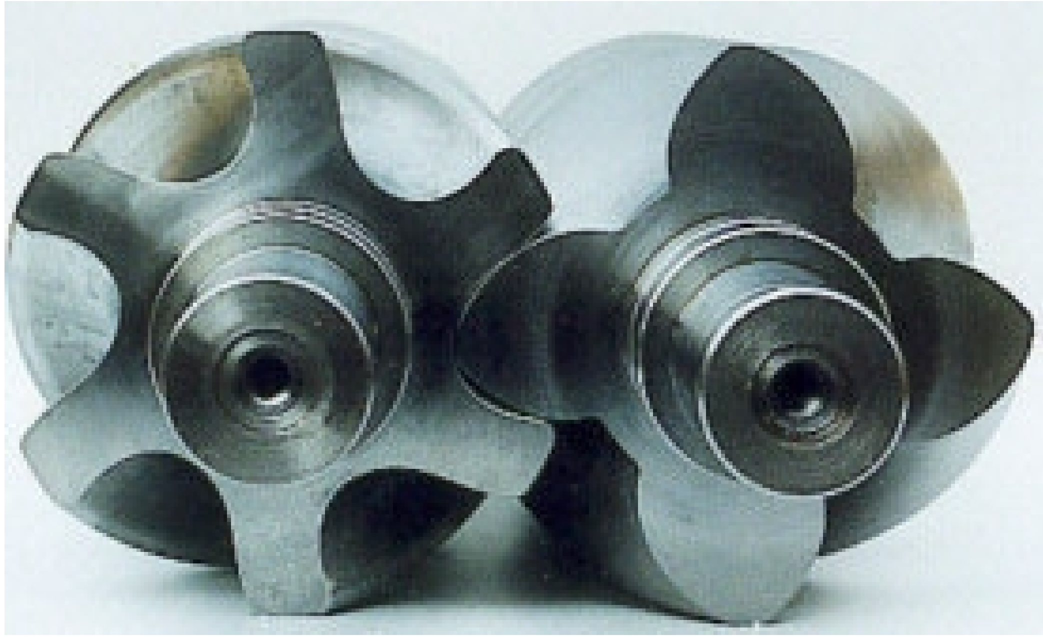
Retrofit ‘N’ Rotors



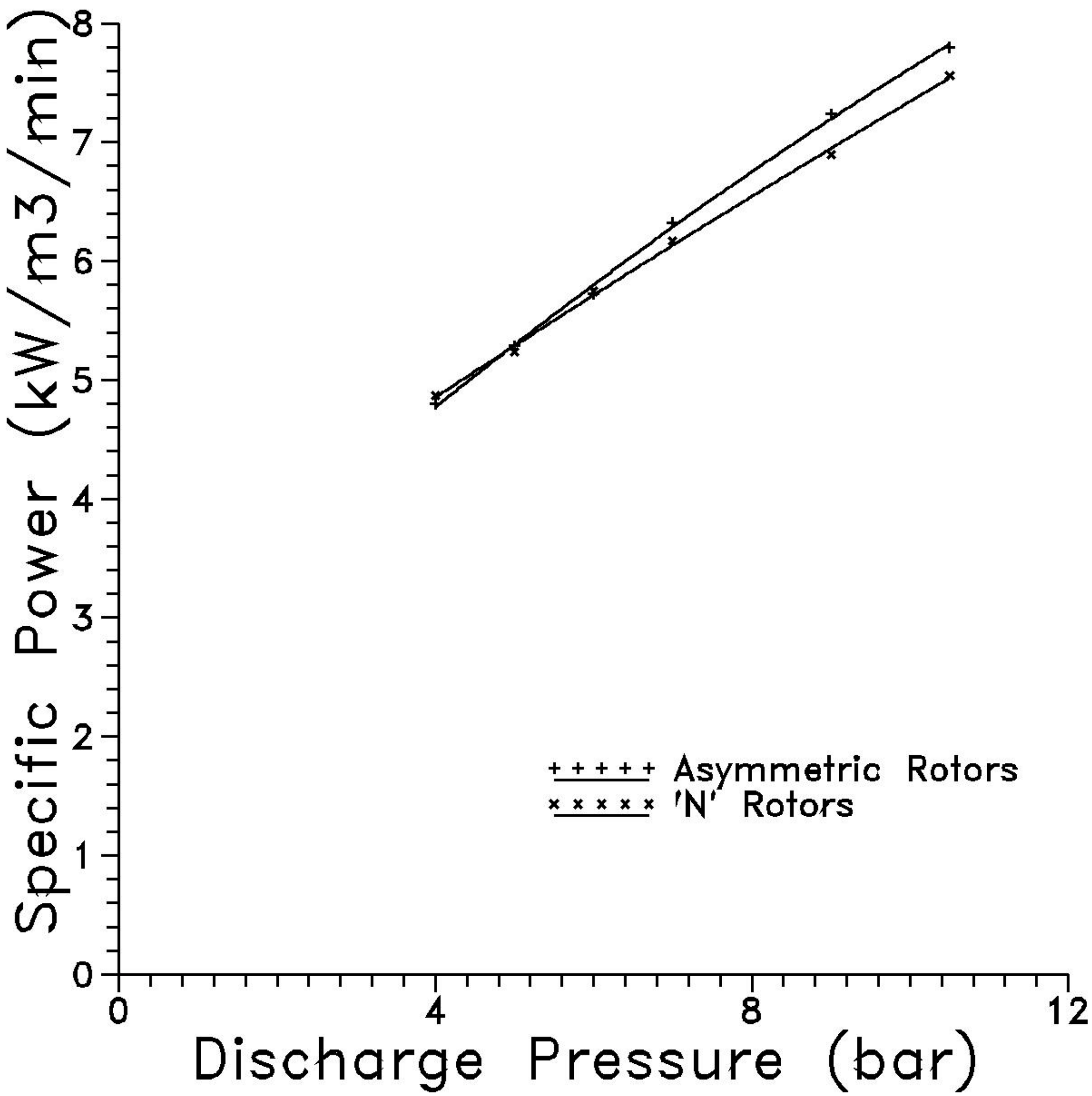
5.4 % more displacement
6.5 % higher delivery
4 % better volumetric efficiency
2.5 % better adiabatic efficiency
75 % less torque on the gate rotor

DESIGN EXAMPLES

Asymmetric Rotors, the most common screw compressor rotors



'N' Rotor retrofit for more efficient screw compressors



Specific Power (kW/m3/min)
Discharge Pressure (bar)
0
1
2
3
4
5
6
7
8
0
4
8
12
+++++ Asymmetric Rotors
××××× 'N' Rotors

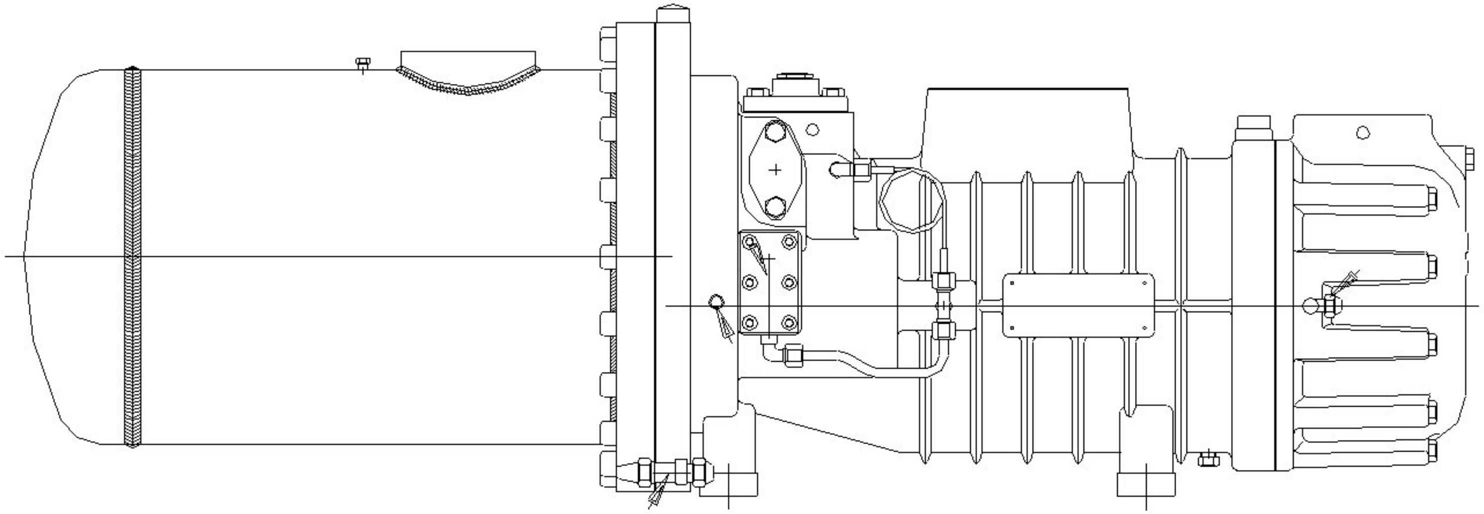
Design of a semihermetic compressor for air-conditioning and refrigeration based on ‘N’ rotors

- 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

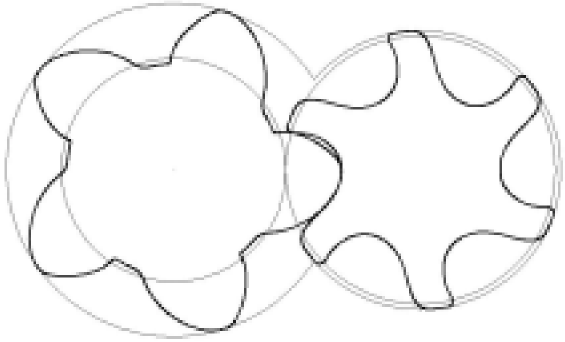
5/6-102 mm L/D=1.55 compressor

- Theoretical displacement 0.72 l/rev
- Gear-box 1500-10000 rpm
- Delivery 0.920-5.80 m^3/min
- R-22
- Air condition 5/40 ^oC 60-380 kW
- COP 4.20 at 3000 rpm
- Refrigeration -5/30 ^oC 33-210 kW
- COP 3.95 at 3000 rpm

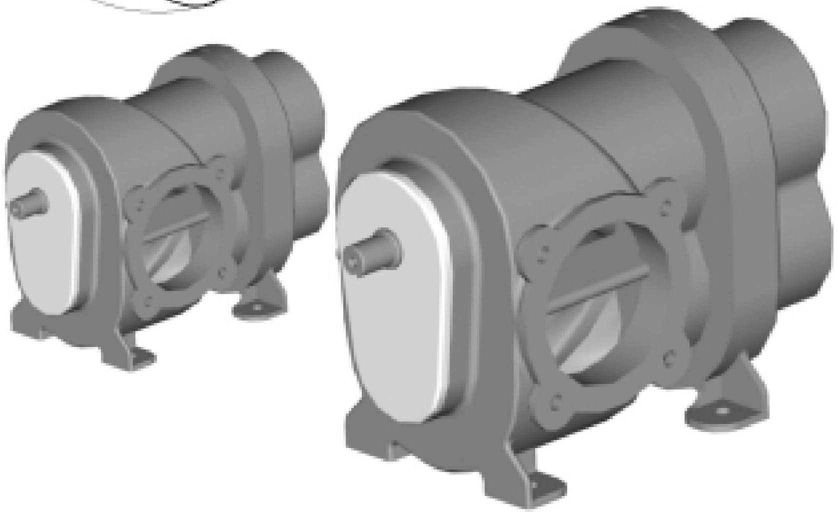
Refrigeration Application



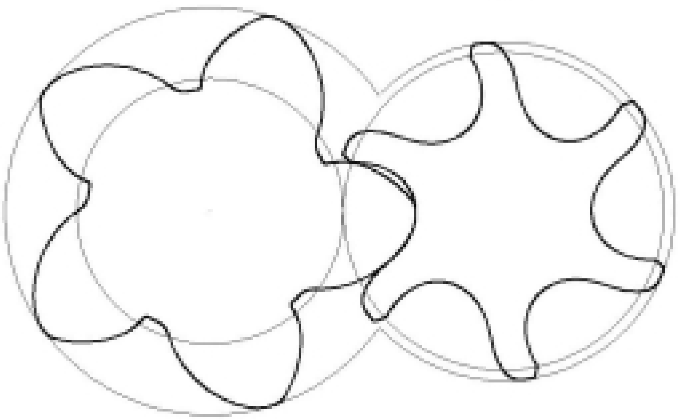
Application of Design Optimization: A Family of Two-Stage Compressors



2nd



1st



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.

EPILOGUE

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.