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Academic excellence for
business and the professions

8th International Conference on Compressors and Refrigeration, Xi'an, China, 20-22/07/2017

KEYNOTE

Modelling of Multiphase Twin Screw Machines

Professor Ahmed Kovacevic
Howden Chair in Engineering Design and Compressor Technology

Centre for Compressor Technology
Department of Mechanical Engineering and Aeronautics

18,000 students - 46% at postgraduate level from more than 150 countries

1894 - Northampton Polytechnic Institute
1966 - University created by Royal Charter
2016 - City joins the University of London

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5 Schools: Business, SMCSE, Arts and Social Sciences, Law, Health Sciences
Graduate School; Research Centres; Interdisciplinary Centres

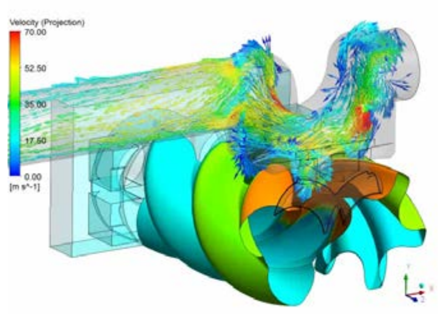
School of Mathematics, Computer Science and Engineering
Department of Mechanical and Aeronautical Engineering
Centre for Compressor Technology

Centre for Compressor Technology

- Established in 1995 to assist UK compressor industry
- Consultancy to over 100 organisations in 30 countries
- Books, Journal and Conference Publications, Patents, Awards
- Spin off and Start-up, licenseing rotor profile and software

Main activities

- Research in Screw Compressors and Expander; rotor profiling; modelling; multiphase flows; CFD, computational and experimental methods
- Design, Testing, Development with industry

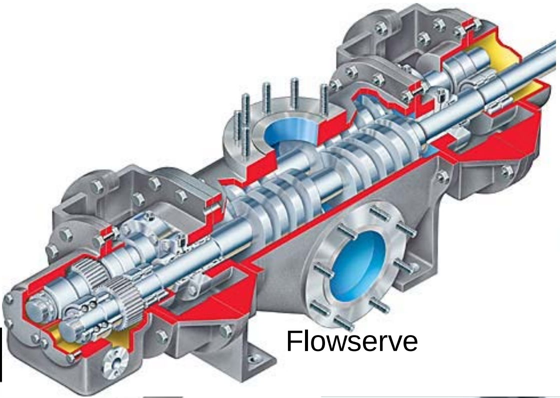


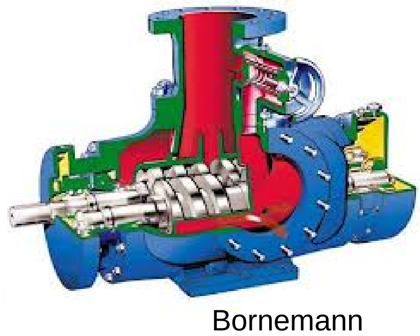
Agenda

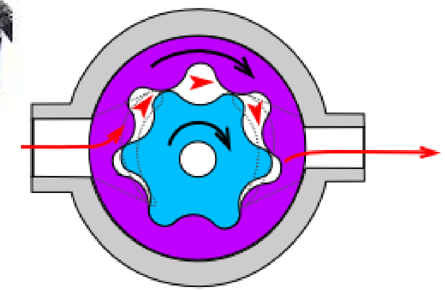
Introduction

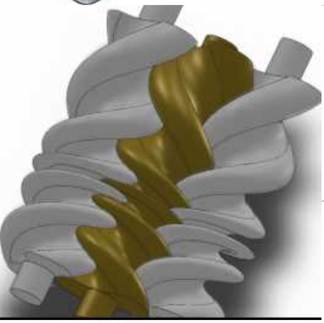
- 3D Modelling of multiphase screw machines
- Recent developments in grid generation for CFD in multiphase screw machines
- Test cases

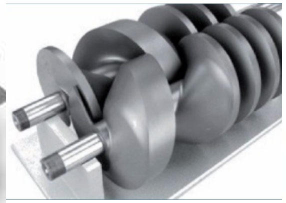
Thank you to:
EPSRC, Trane, UTRC, Goodrich, Howden, Kobelco,
PDM Analysis Ltd, Simerics, Star-CCM+, CFX Berlin, VertRotors,
for support in development of SCORG and CFD in PD screw machines.











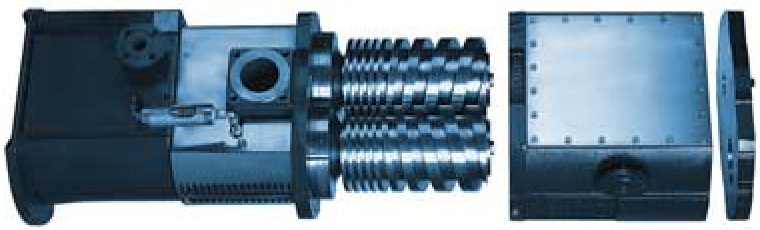
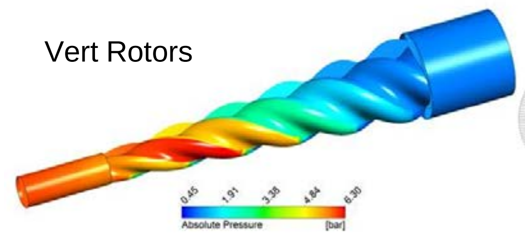
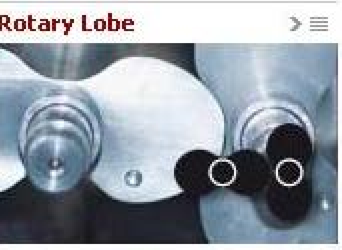
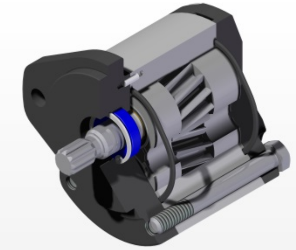
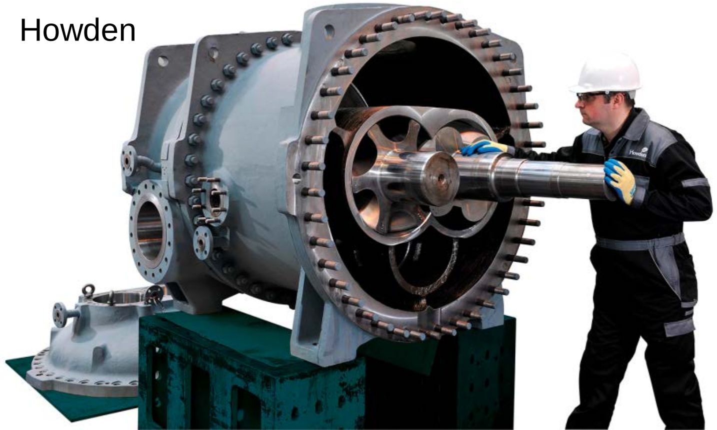


Fig. 2: Rotary screw vacuum pump (courtesy of SIHI).









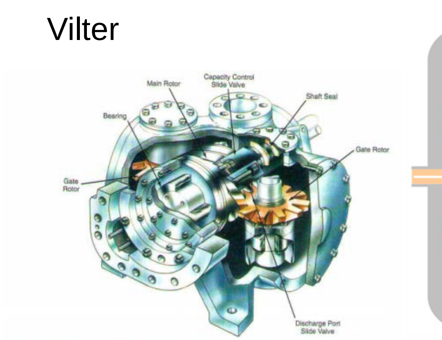
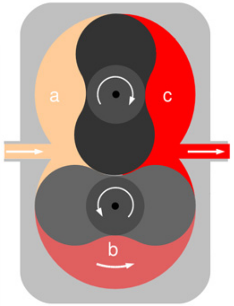
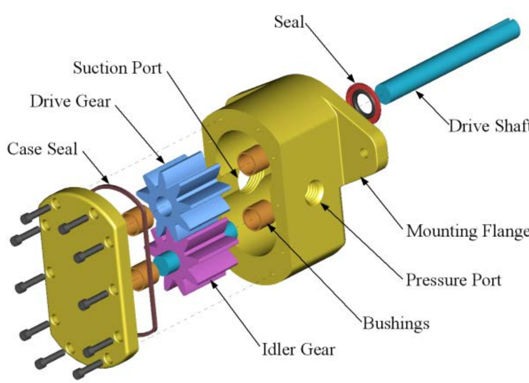
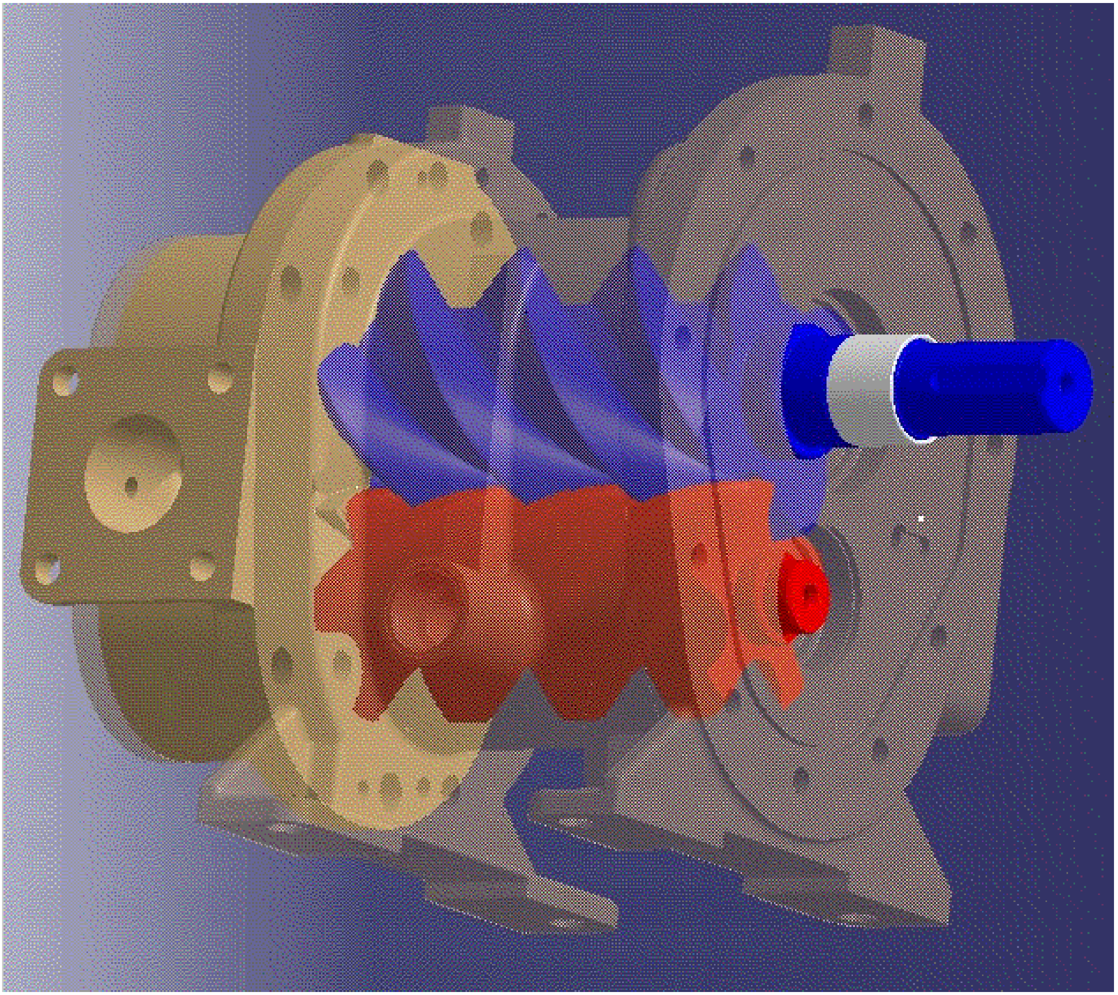


Figure 4-67. Single screw compressor. Note the location of the main rotor in relation to the two gate rotors. (Vilter Manufacturing Corporation)





Screw Compressors Today



83% Oil injected ; 17% Oil free

- Applications:
  Industrial and commercial
  Air compression, Refrigeration,
  Process gasses Oil & Gas,
  Expanders, multiphase
- Dia (35) 50 – 1000 mm
- 0.3 – >1000 m^3/min
- 0.5 kW – 5 MW
- High Efficiency, Reliable

~ 11 million screw compressors produced to date

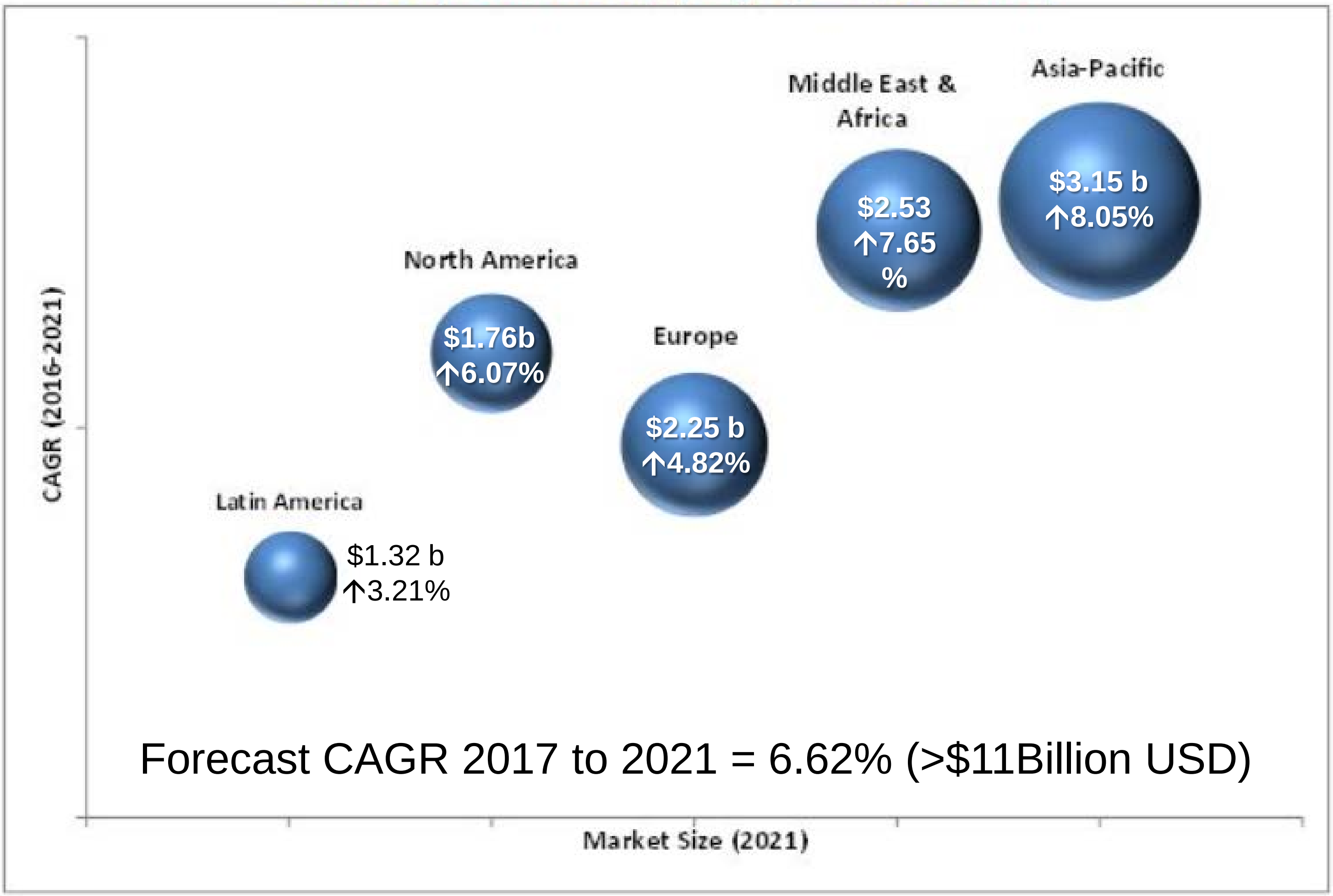
Bear shaft screw compressors commissioned in 2016 (est.):

Refrigeration:	166,000
Oil and Gas:	24,000
Petrochemical:	25,000
Air (est.)	>500,000
Large Packages	~800

80% of new industrial compressors are screw compressors
17% energy produced in developed countries used for compression
25% energy in USA during summer is used for refrigeration and air-conditioning

Global screw compressor sales $7.99 billion in 2016

Screw Compressor Market, by Region, 2021 (USD Billion)



Source: MarketsandMarkets Analysis

The oil-free segment is expected to grow at the highest CAGR from 2016 to 2021

Compressors and energy

- Compressors consume more than 17% energy produced in developed countries. This pollutes the environment with more than 3000 MtCO_2 per year, while energy costs exceed €275 billion per year*.
- The global CO_2 emission will increase by up 28% from 2015 to 2030,
- The latest EU targets for 2020 are to reduce the CO_2 emissions by 20% from the levels recorded in 1990. This requires:
  - 20% of energy produced by renewable sources
  - increase energy efficiency by 20% from the levels recorded in 2007.

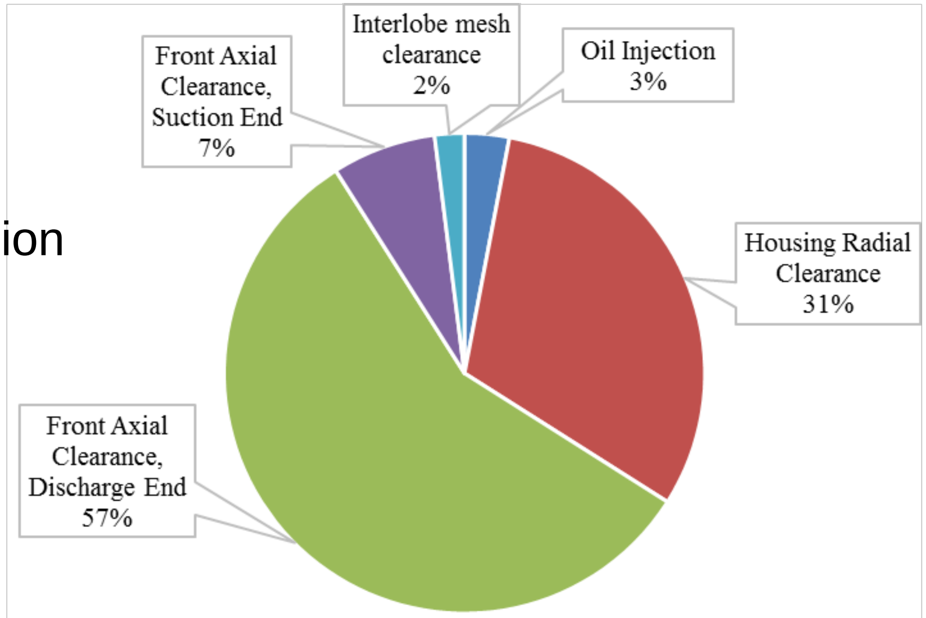
currently these targets may not be achieved despite efforts by both industry and academia.

- Oil injected compressors and other multiphase fluid handling machines have great potential for improvements in efficiency and contributing to reduction in CO_2 emission.

*Source: http://www.iea.org/publications/freepublications/publication/EE_for_ElectricSystems.pdf

Introduction

- Purpose of oil Injection
  - Cooling of the gas during compression
  - Sealing of the leakage gaps
  - Lubrication of rotors in contact



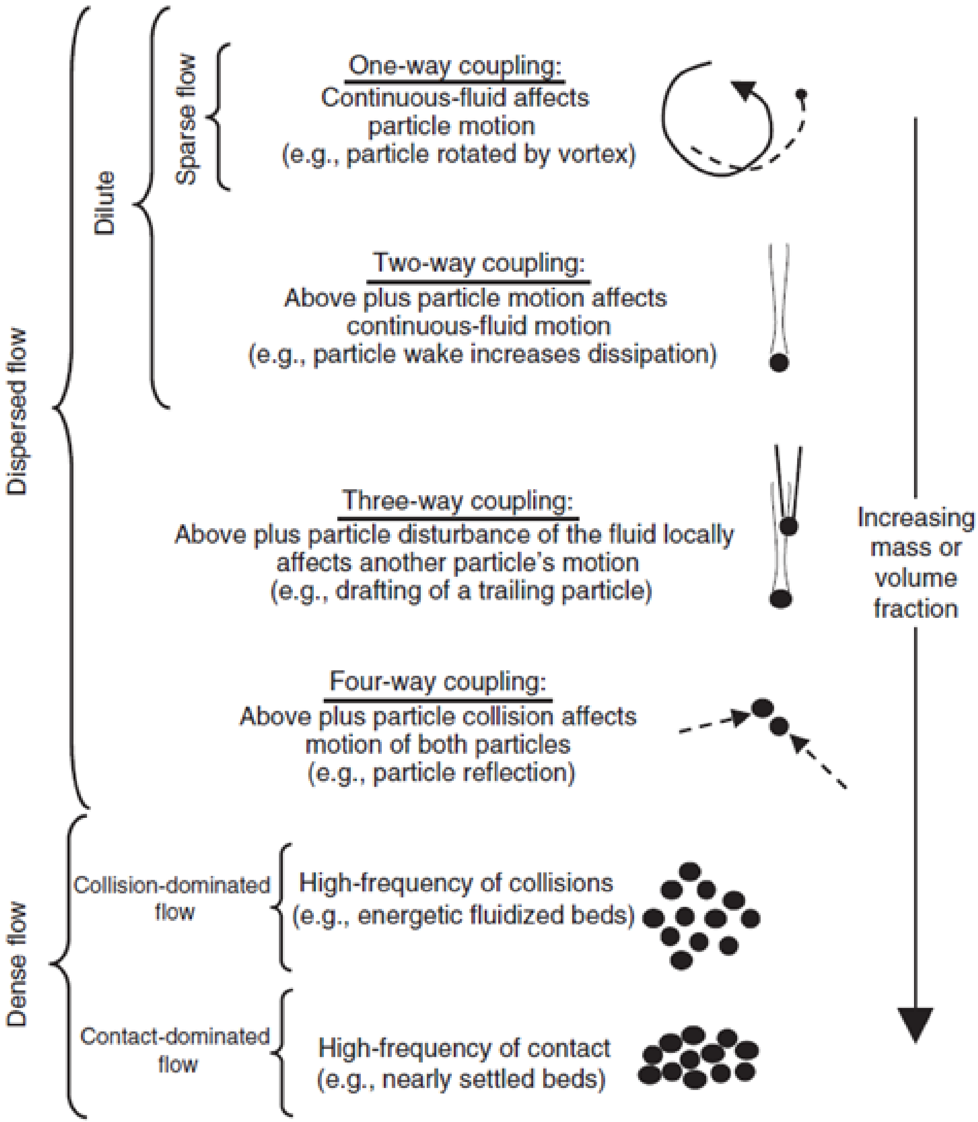
Oil Power Loss Contributions in Leakages

(*Deipenwisch and Kauder*, 1999)

- Factors that affect efficiency:
  - Viscous friction power loss, oil drag and momentum loss
  - Optimum quantity and timing of oil injection
  - Oil injection temperature and residence time inside the compression chamber
  - Spray formation, droplet diameter and spread,
  - Impingement on the rotors and casing effect the usability of injected oil.

- Challenges:
  - It is almost impossible to gain optical visualization inside the compression chambers.
  - Limitations of currently used experimental and analytical methods

Modelling of Multiphase Flow



Crowe et al. (2006)

- Characterization of multiphase flow regimes based on the various coupling effects between the continuous and the dispersed phases

- Eulerian – Lagrangian
  - dispersed phase is very low and the phase particles are very fine with negligible momentum
  - compressed gas is treated as the continuous phase and oil droplets as particles in the Lagrangian frame
  - one – way, two – way and turbulence couplings possible

- Eulerian – Eulerian
  - condition of heavily oil flooded operation
  - in addition to the oil droplets, oil film on the rotor and housing will occur
  - the pressure field solution is shared by the two phases and the independent momentum equations calculate relative slip and shear between the gas and oil

Mathematical models for calculation of multiphase positive displacement screw machines

1) Analytical methods

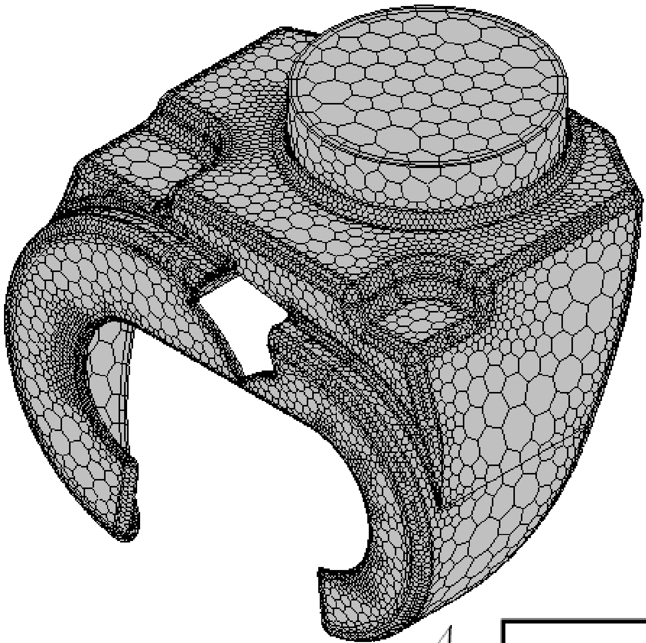
   pV^n=const - analytical calculation of "n"

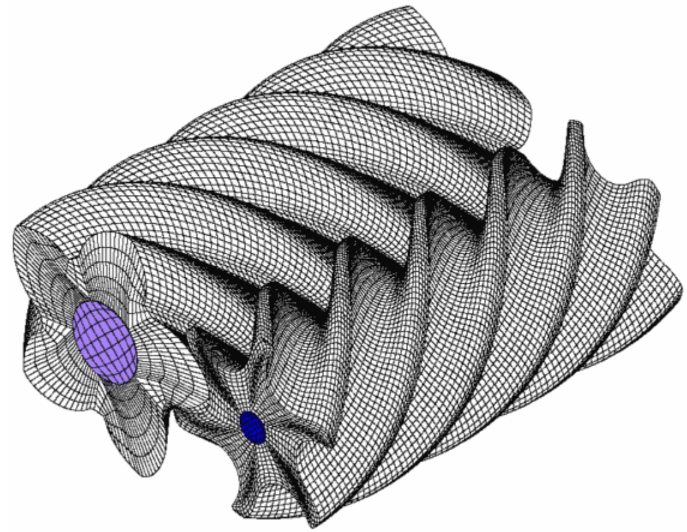
   simple model, very inaccurate

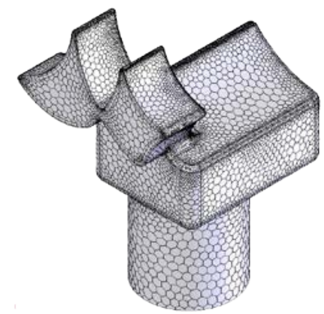
2) Differential methods (based on conservation principles)

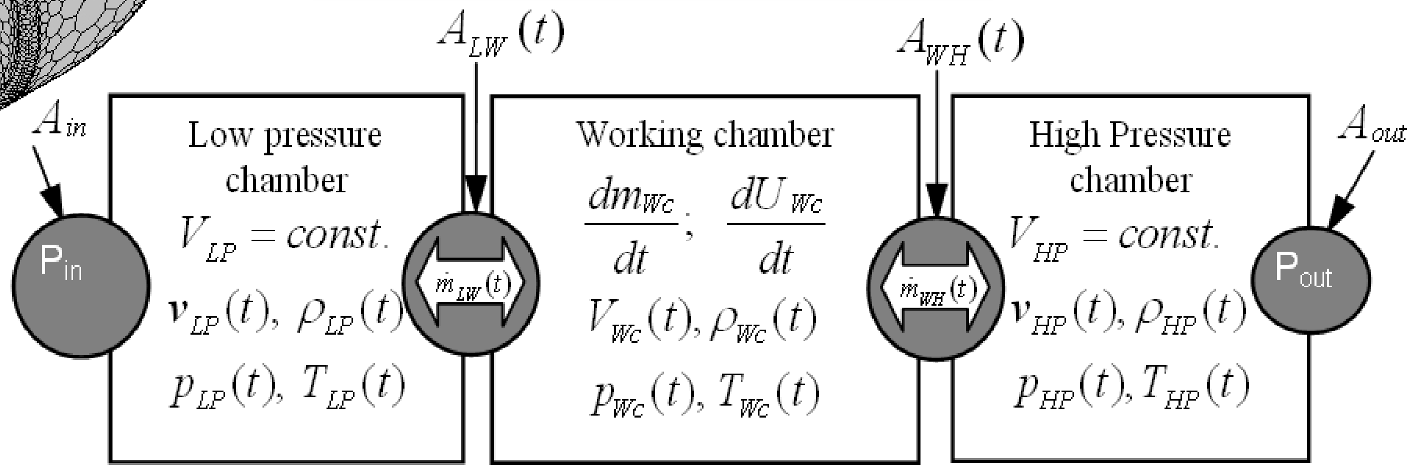
   A. Thermodynamic chamber model – increased complexity, many assumptions, better accuracy
   B. 3D Computational Fluid Dynamics (CFD) model complexity significantly increased, very few assumptions are made
   C. Integrated model – not as complex as 3D but more accurate than Chamber model

General conservation equation – differential methods









$$\frac{d}{dt}\int_V \rho\phi dV + \int_S \rho\phi(\mathbf{v}-\mathbf{v}_b)\cdot d\mathbf{s} = \int_S \Gamma_\phi \operatorname{grad}\phi \cdot d\mathbf{s} + \int_S \mathbf{q}_{\phi S}\cdot ds + \int_V q_{\phi V}\cdot dV$$

Oil Injected Screw Compressor Modelling

■ Thermodynamic Chamber Model

Dilute Oil Flow

Oil droplets are assumed to be spherical of a constant mean Sauter diameter.
Balance of the heats exchanged between the spherical droplets and the gas is used for calculating temperature

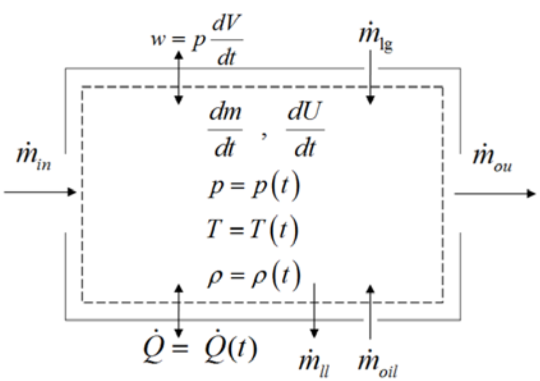
$$\frac{dT_o}{d\theta} = \frac{h_o A_o \left(T_{\text{gas}} - T_o\right)}{\omega m_o c_{\text{oil}}}$$

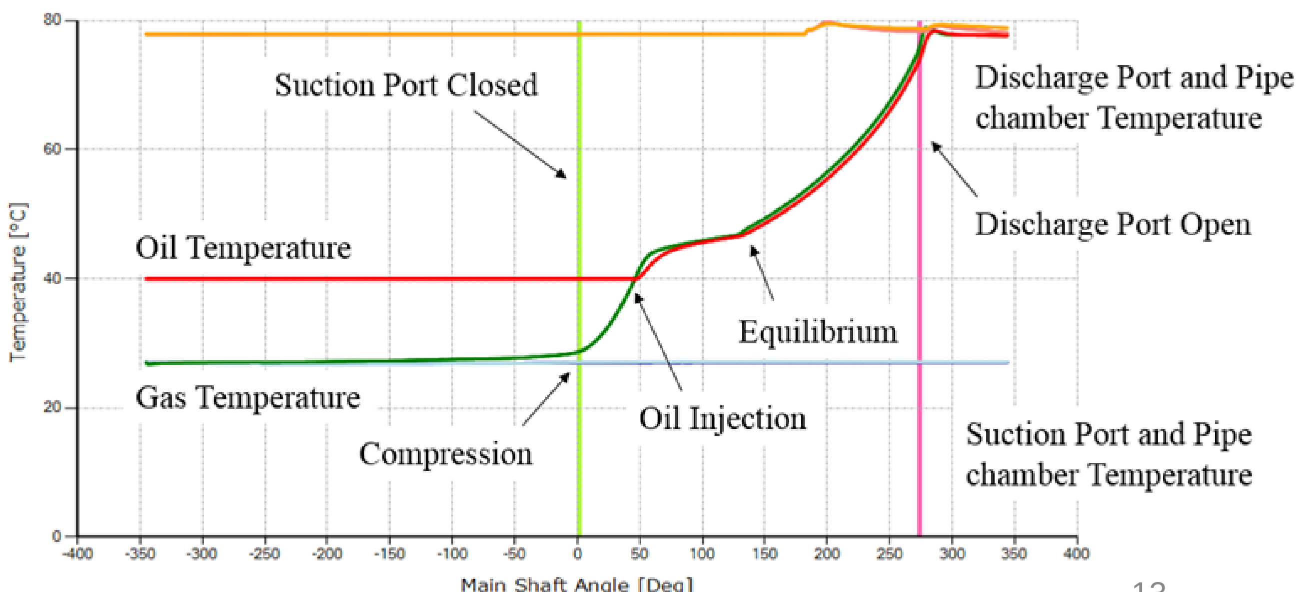
$$T_o = \frac{T_{\text{gas}} - kT_{o,p}}{1+k}$$

$$Nu = 2 + 0.6\,\text{Re}^{0.6}\text{Pr}^{0.33}$$

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

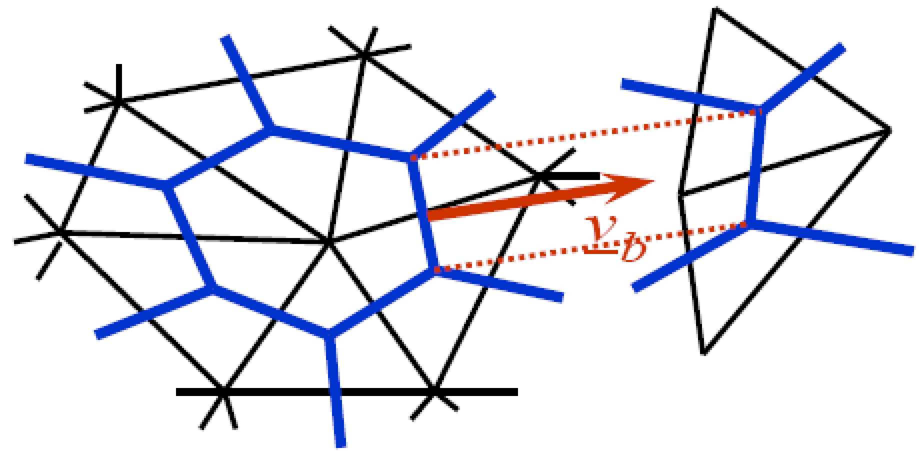
the non-dimensional time constant of the droplet





Mass, Momentum, Energy and Space conservation in 3D CFD

$\underline{v}_b$ is the velocity of the control volume edge



$$\frac{\partial}{\partial t}\int_V \rho \, dV + \int_S \rho \, (\vec{v} - \vec{v}_b).\vec{n} \, dS = 0$$

$$\frac{\partial}{\partial t}\int_V \rho\vec{v}dV + \int_S \rho\vec{v}(\vec{v} - \vec{v}_b).\vec{n}dS = \int_V \rho\vec{f}dV + \int_S \vec{T}dS$$

$$\frac{\partial}{\partial t}\int_V \rho E dV + \int_S \rho E(\vec{v} - \vec{v}_b).\vec{n}dS = \int_V \rho\vec{f}.\vec{v}dV + \int_S \vec{n}.\vec{\sigma}.\vec{v}dS - \int_S \vec{q}.\vec{n}dS$$

$\rho, p, E = cst, \vec{v} = 0$ →

$$\frac{\partial}{\partial t}\int_V dV = \int_S \vec{v}_b.\vec{n} \, dS$$

Space conservation law

Control volumes must exist in time

⇨ Connectivity must be kept fixed during timestep

from J. Vierendeels, Ghent University, Introduction to CFD in PD machines, Short Course, London, Sep 2015

Modelling of Oil Injected Screw Machines

Eulerian treatment of the compressed gas and the injected oil

a) Full Euler-Euler
- Pressure field shared between the phases
- Independent u,v,w - momentum conservation equation for each phase with interphase drag effects
- Mass conservation between phases in case of phase change.
- Independent energy conservation equation with interphase heat transfer
- Homogeneous or Phase specific turbulence model

a) Volume of Fluid suitable for dense stratified flows
- Pressure field shared between the phases
- One additional momentum conservation equation for liquid phase

b) Simplified Euler approach for fluids with no slip conditions
- Pressure field shared between the phases
- Additional concentration equation with special modelling of source terms

Definition of phases → Definition of Interphase Interactions → Solve Conservation PDE's

Solution algorithm

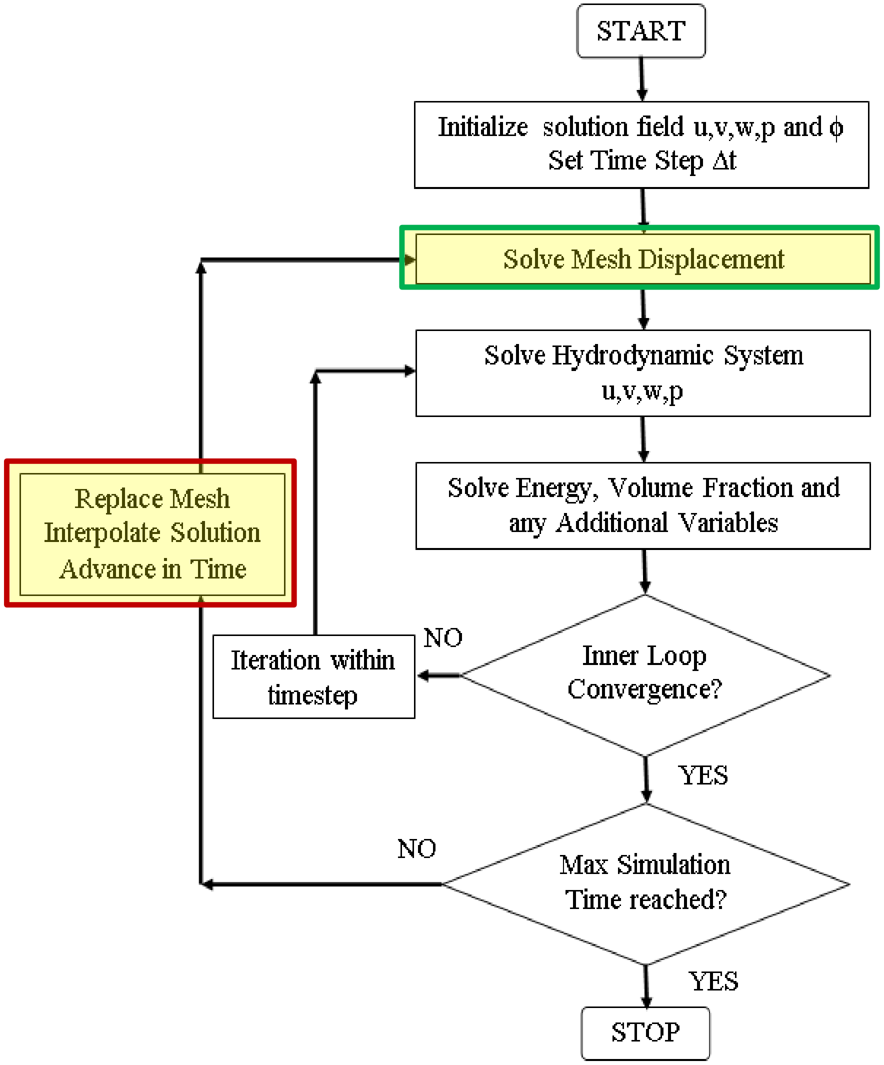
Remeshing

Prediction parameters

- Pressure field,
- Temperature field,
- Velocity field,
- Mass flow rates,
- Leakage volume and Efficiency,
- Power,
- Dynamic losses,
- Design Improvements.

Key Elements

- 3D Transient,
- Turbulence,
- Grid Generation,
- Moving / Deforming Boundaries,
- Compressible Fluids,
- Oil Injection



Prerequisite for reliable 3D CFD

Geometrical Inputs
Boundary Distribution Inputs
Meshing Inputs

↓

Generation of Rotor Profiles and Rack as the Parting line

↓

Intersection with Outer Circles to determine CUSP points and ‘O’ Grid outer boundary

↓

Boundary Discretisation

↓

Adaptation and Mapping of ‘O’ Grid inner and outer boundaries
Check Regularity

↓

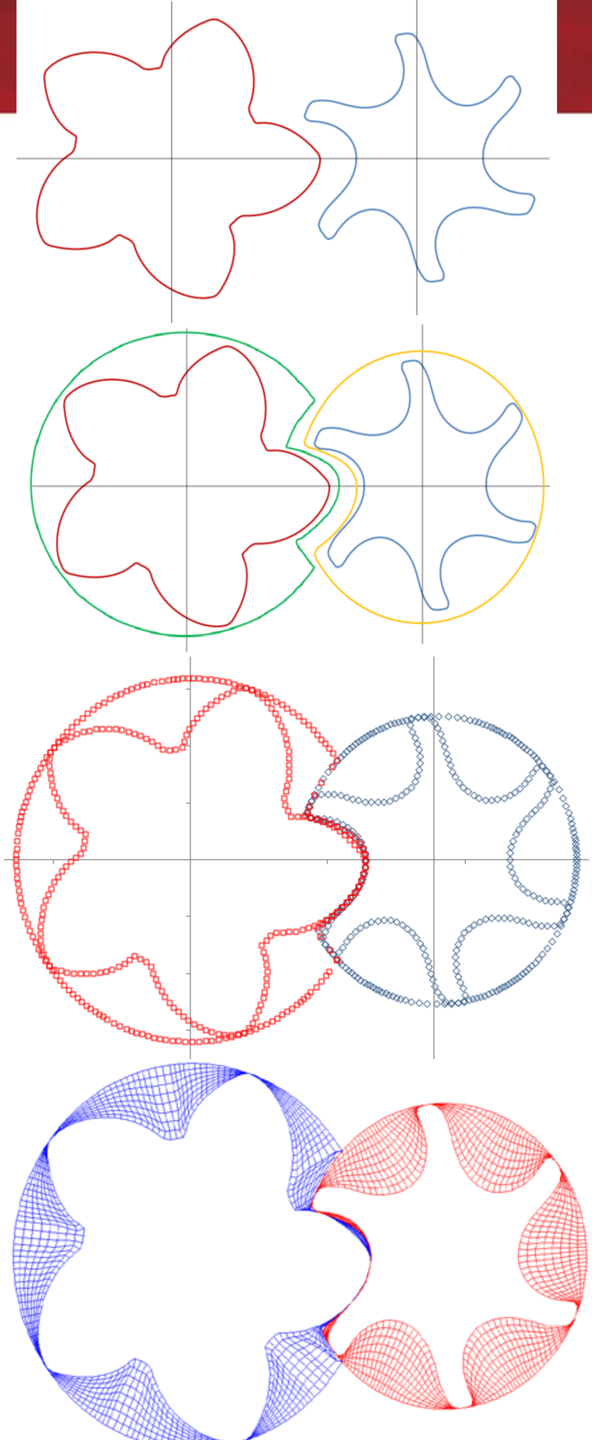
Transfinite Interpolation for Interior Node Distribution

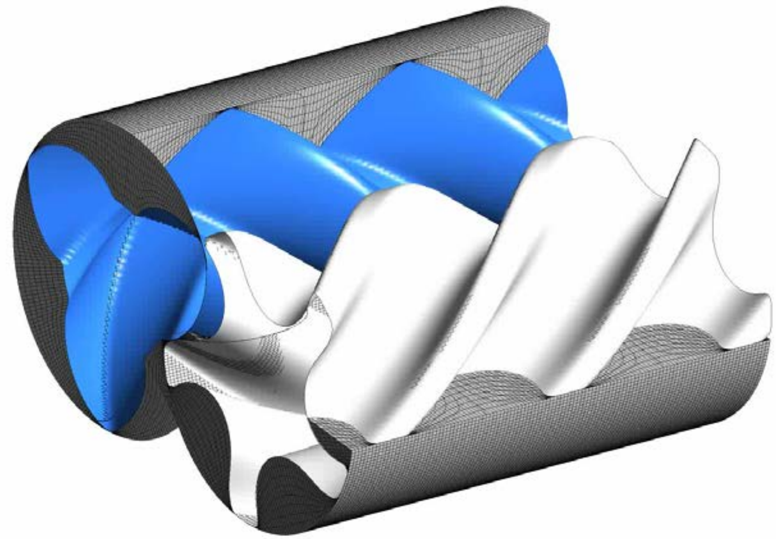
↓

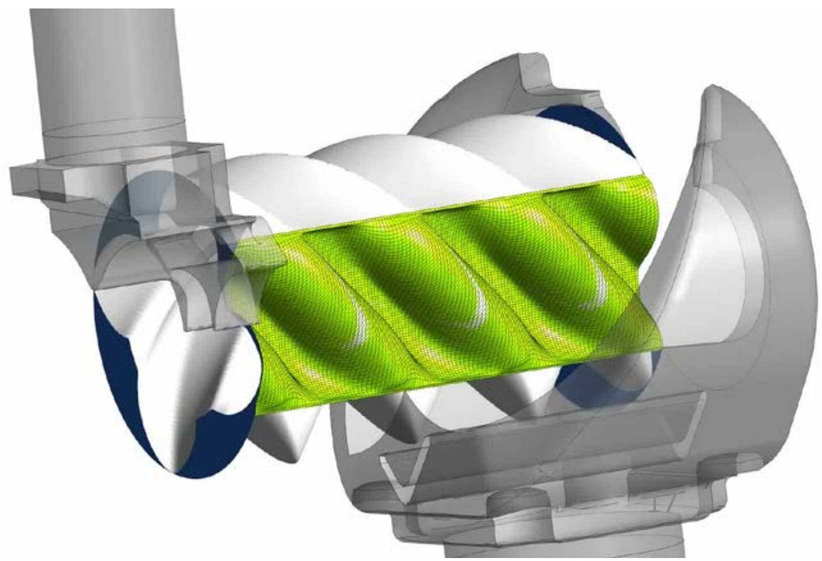
Grid Orthogonalisation and Smoothing

↓

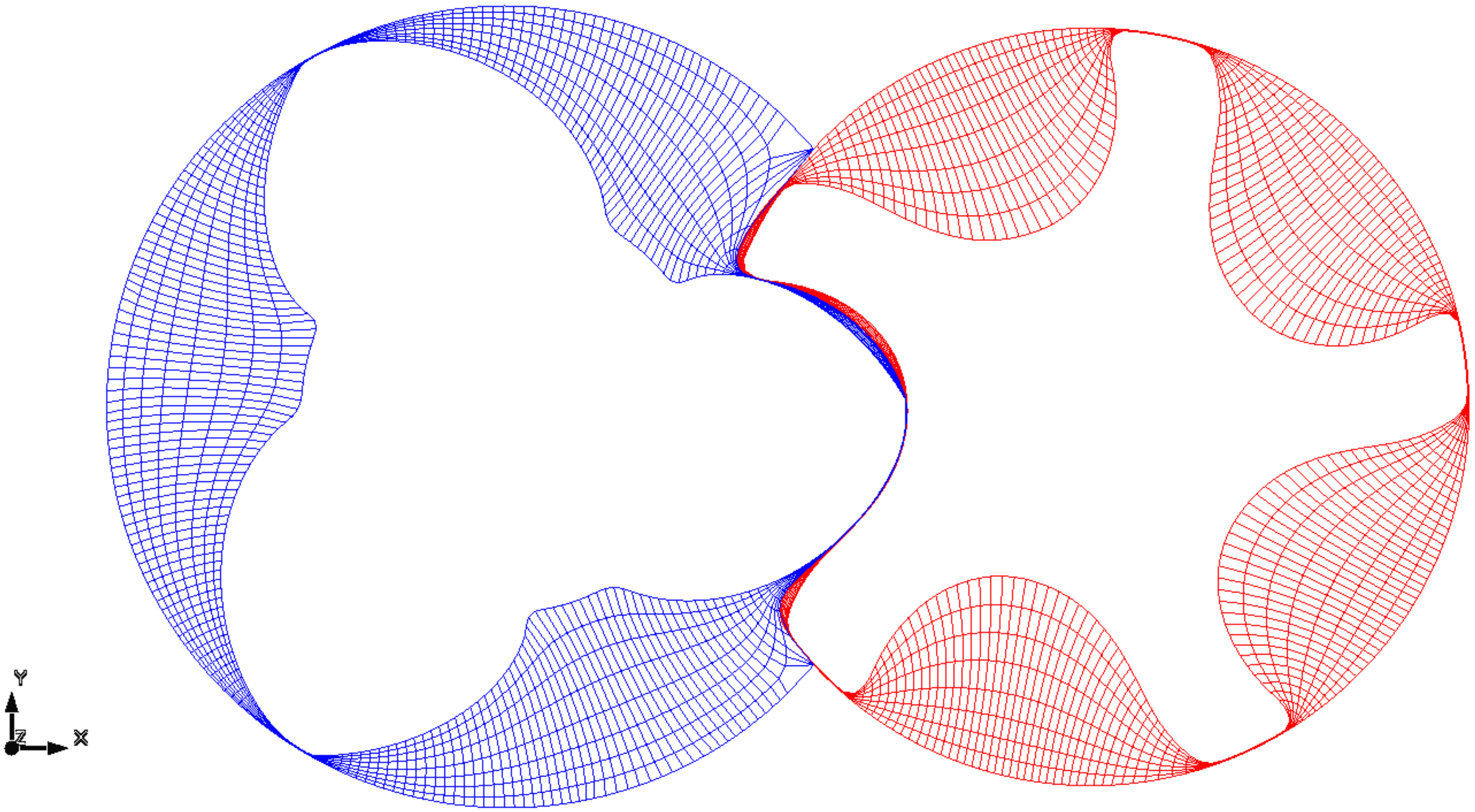
Write Vertex, Cell connectivity and Domain Boundary Data







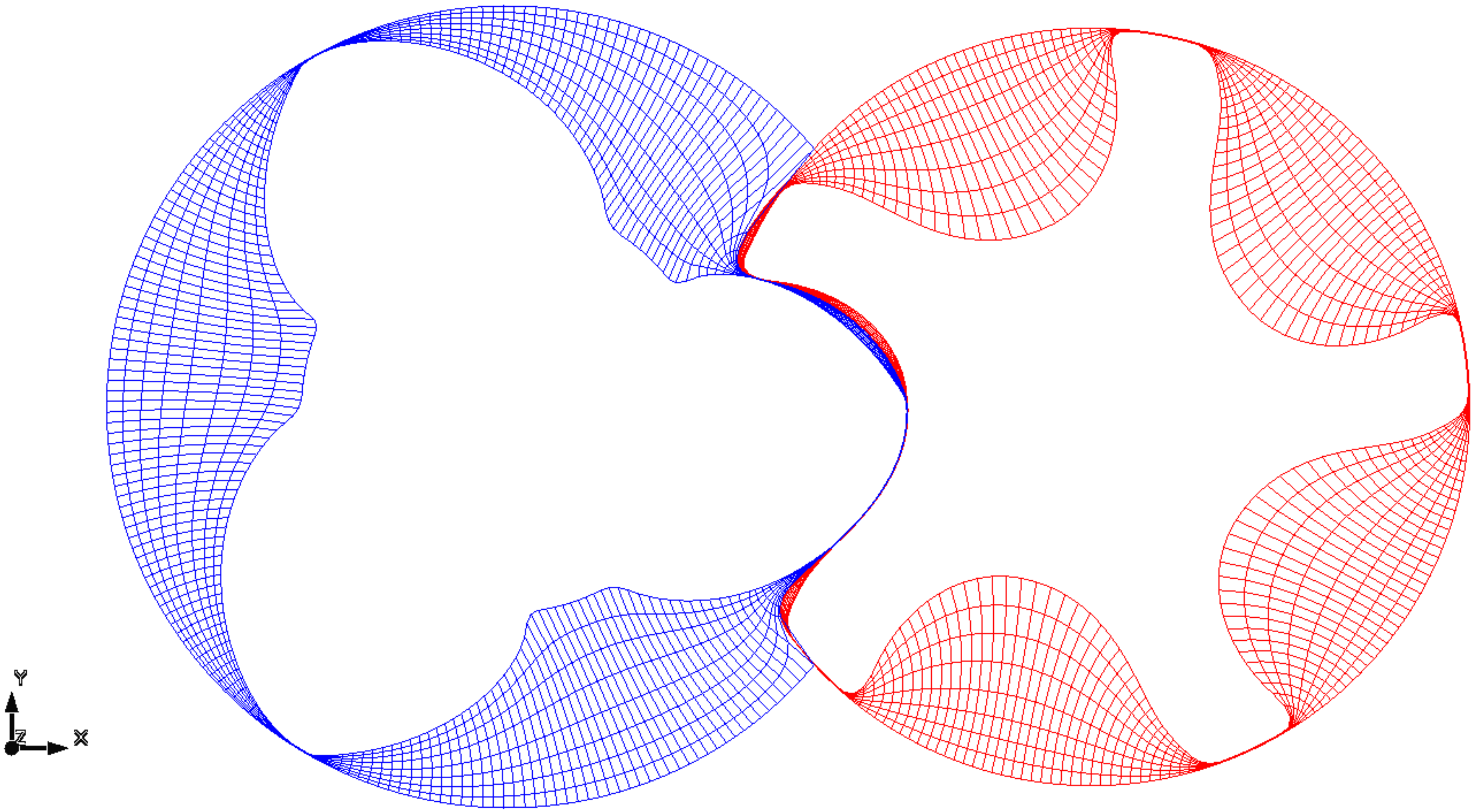
Rotor to Casing Non-Conformal



SCORG

Analytical grid generation with differential smoothing
Sliding and stretching interface between rotor subdomains

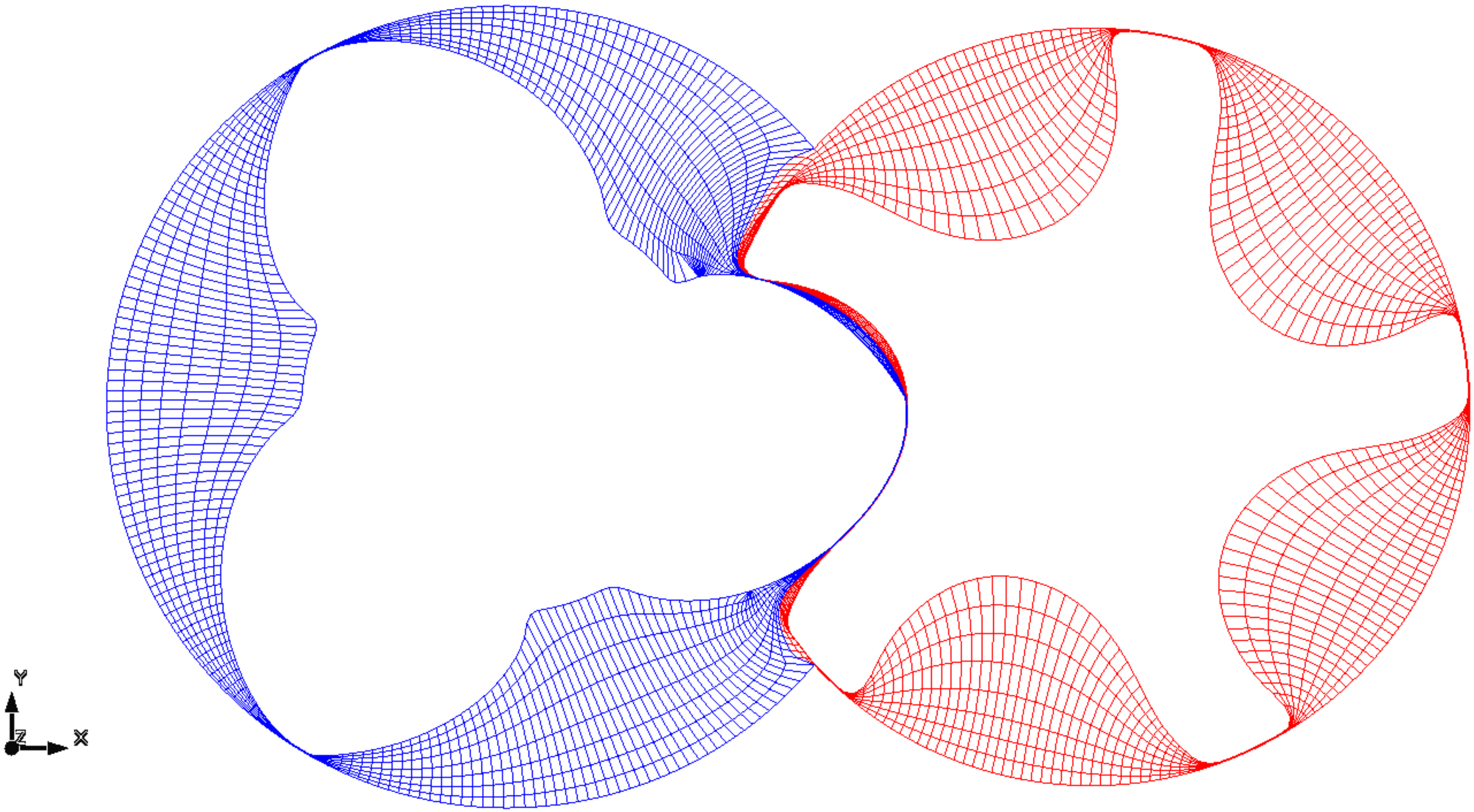
Casing to Rotor Non-Conformal



SCORG

Analytical grid generation with differential smoothing
Stretching non-conformal interface
between rotor subdomains

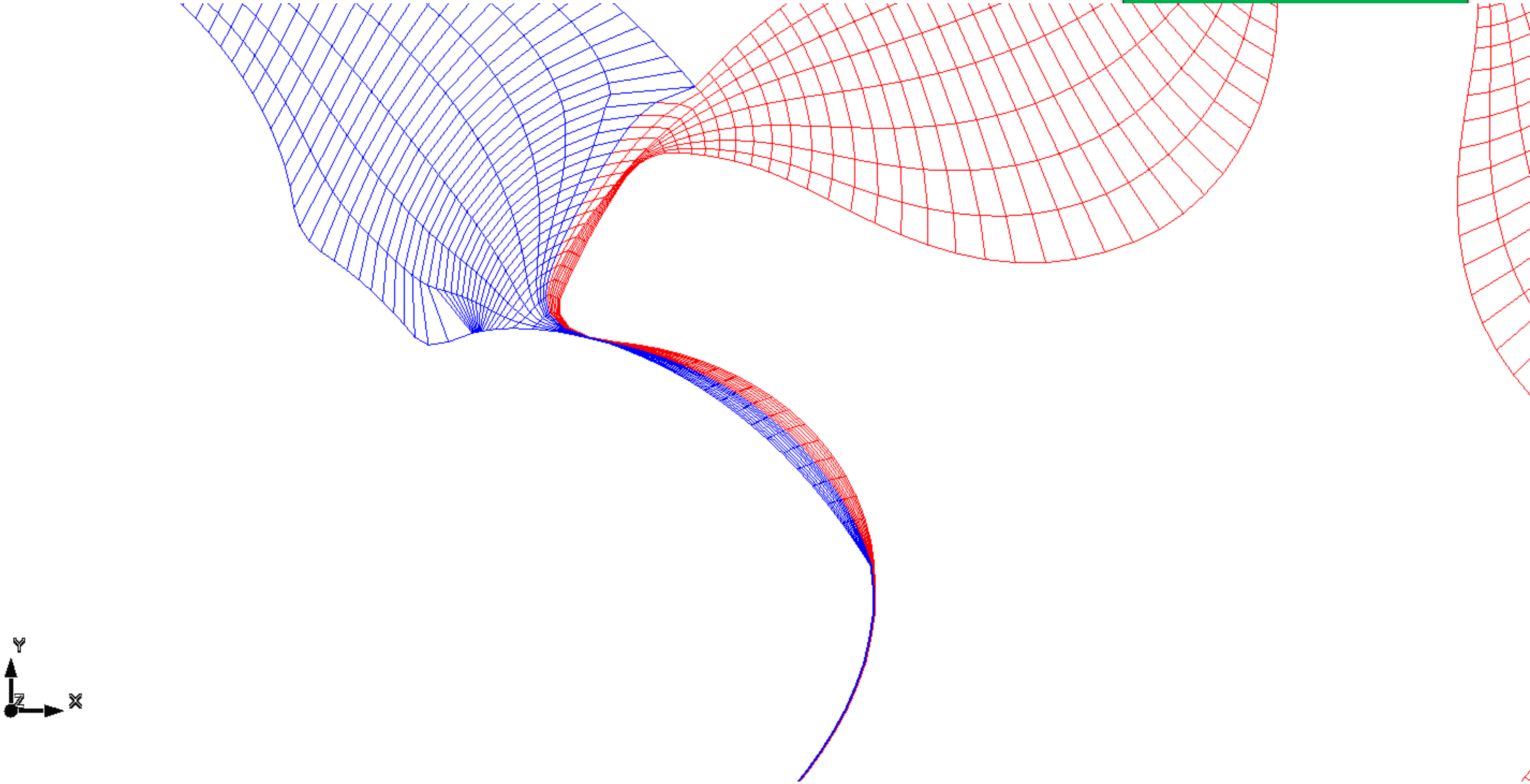
Casing to Rotor Conformal



SCORG

Analytical grid generation with differential smoothing
Single domain of two rotors with no interface

Casing to Rotor Conformal



SCORG

Analytical grid generation with differential smoothing
Single domain of two rotors with no interface

Application of different grid types

Rotor to Casing Non-Conformal

- Two rotor domains with sliding interface
- Machines with multiple gate rotors
- Rotors with straight and helical lobes
- Accurate mapping of rotor profile
- Grid adaptation possible
- Suitable for single phase calculation

Casing to Rotor Non-Conformal

- Two rotor domains with stretching interface
- Machines with multiple gate rotors
- Rotors with straight and helical lobes
- Suitable for large wrap angles (vacuum pumps)
- Suitable for VOF multiphase calculations

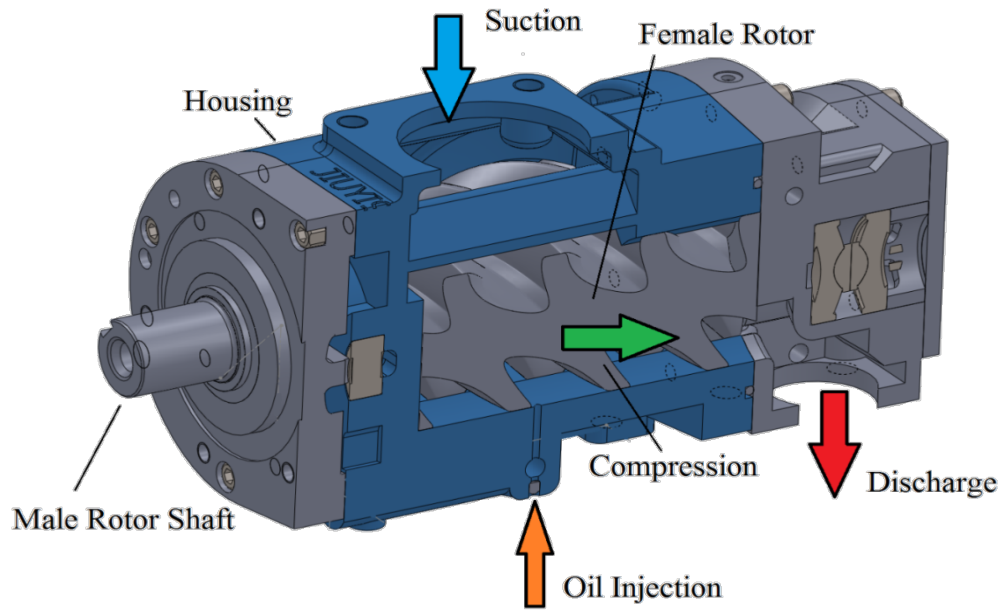
Casing to Rotor Conformal

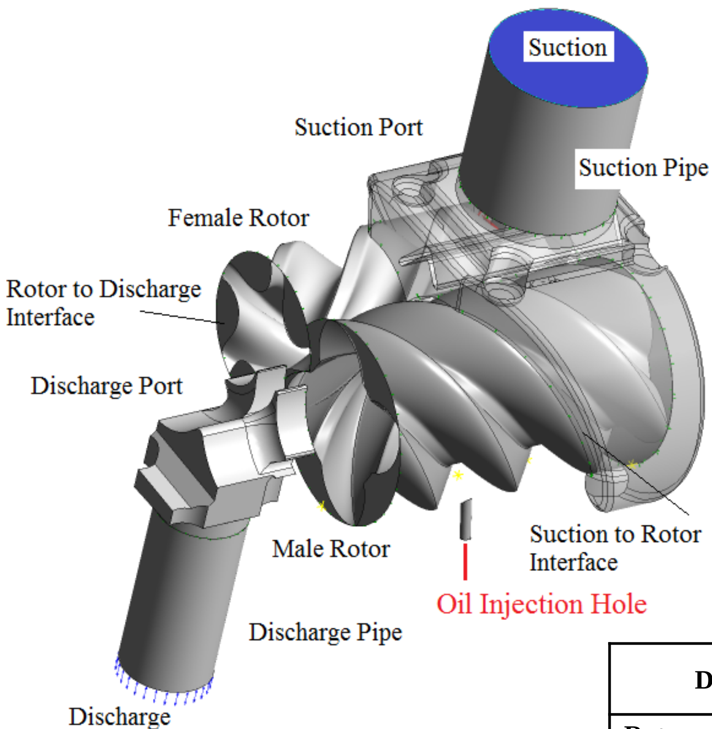
- Single Domain for both rotors – no interface
- Rotors with straight and helical lobes
- Most suitable for multiphase flows
- Stable for Euler-Euler multiphase calculation
- Any commercial CFD solver

SCORG

Oil Injected – SCORG & Ansys CFX

- 4/5 "N" rotor profile
- Centre Distance, 93.00mm
- Main Rotor OD, 105.28 mm
- L/D Ratio, 1.55
- Wrap Angle, 306.6º
- Built in V_i, 4.8





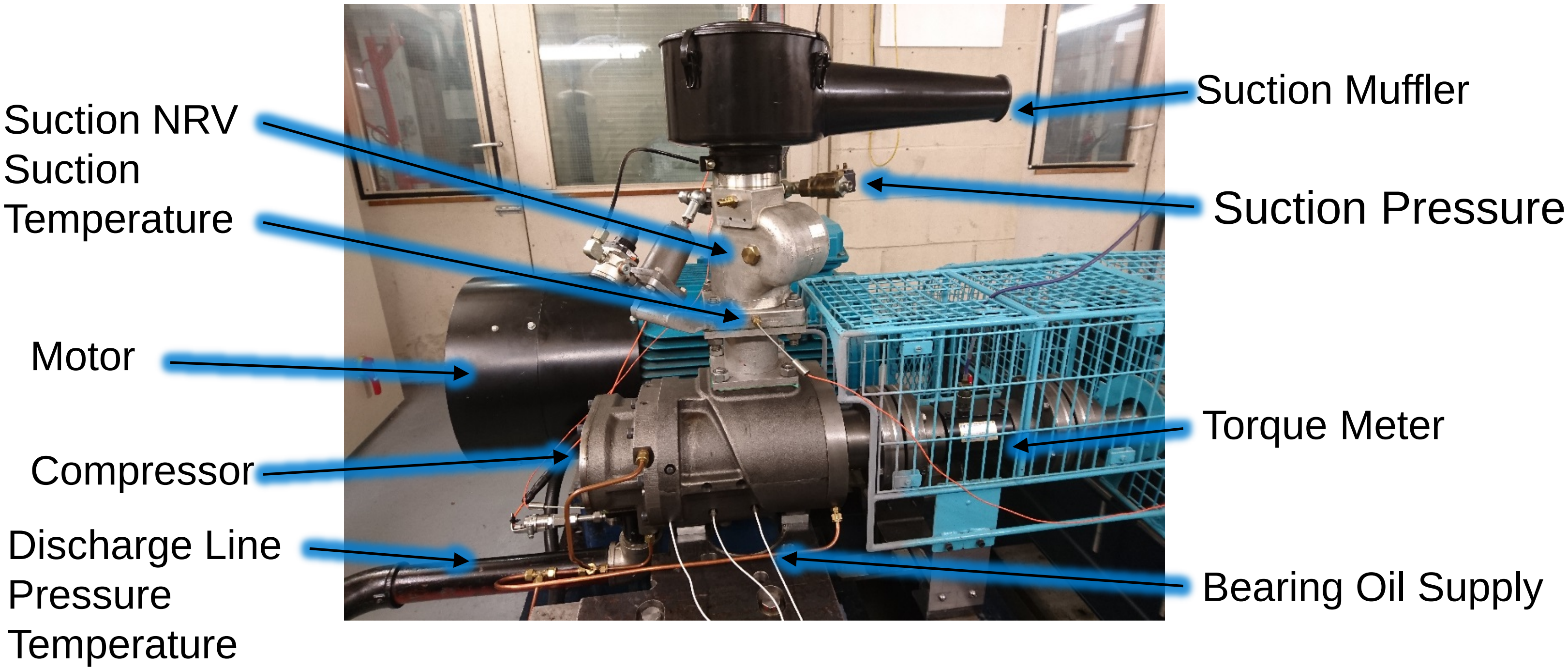
Clearances,

Interlobe **50 μm**
Radial **50 μm**
End Axial **50 μm**

- Eulerian – Eulerian two phase model
- Phase I – Air Ideal Gas
- Phase II – Constant property Oil
- First order discretisation
- CFX Solver

Domain	Cell Structure	Node Count	Cell Count	Orthogonality Angle (Min)	Expansion Factor	Aspect Ratio
Rotor	Hexahedral	468677	406368	7.4	646	488
Suction Port	Tetra + Hex	119058	203255	30.2	279	9
Discharge Port	Tetra + Hex	98521	253095	19.6	53	28
Oil Injection Port	Hexahedral	28340	25144	55.7	4	4

Measurements



City, University of London Test Rig

Case	Speed (rpm)	Suction Pressure (bar)	Suction Gas Temperature (K)	Discharge Pressure (bar)	Oil injection Pressure (bar)	Oil Injection Temperature (K)
1	3000	1.00	298.0	6.0	5.5	323.0
2	3000	1.00	298.0	8.0	7.5	323.0
3	6000	1.00	298.0	8.0	7.5	323.0

Oil Injected Compressor – Mesh type comparison

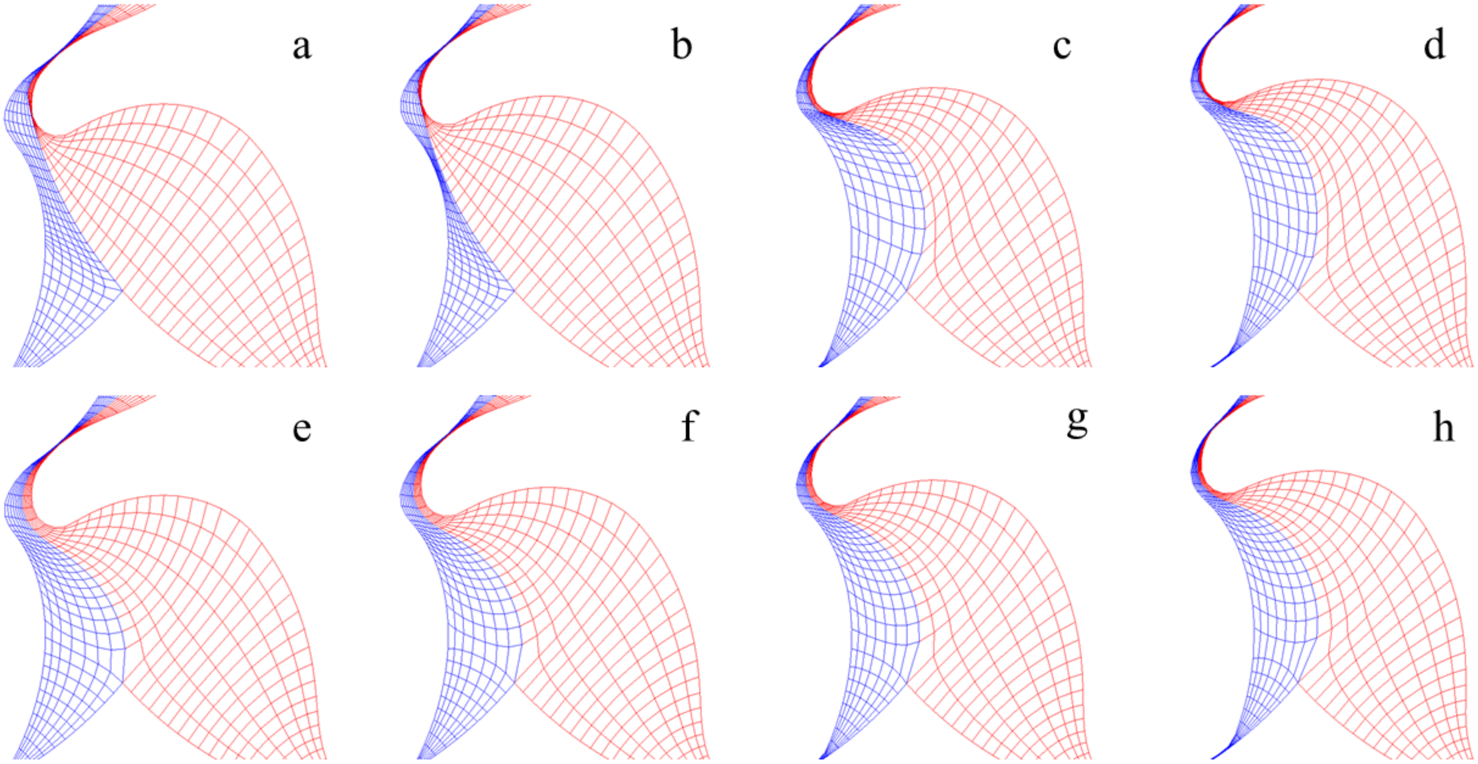
Grid transition in cross section

Case 1
Single
Domain
Algebraic

Case 2
Single
Domain
Differential

And

Case 3
Two Domain
Differential



Oil Injected Compressor – Mesh type comparison

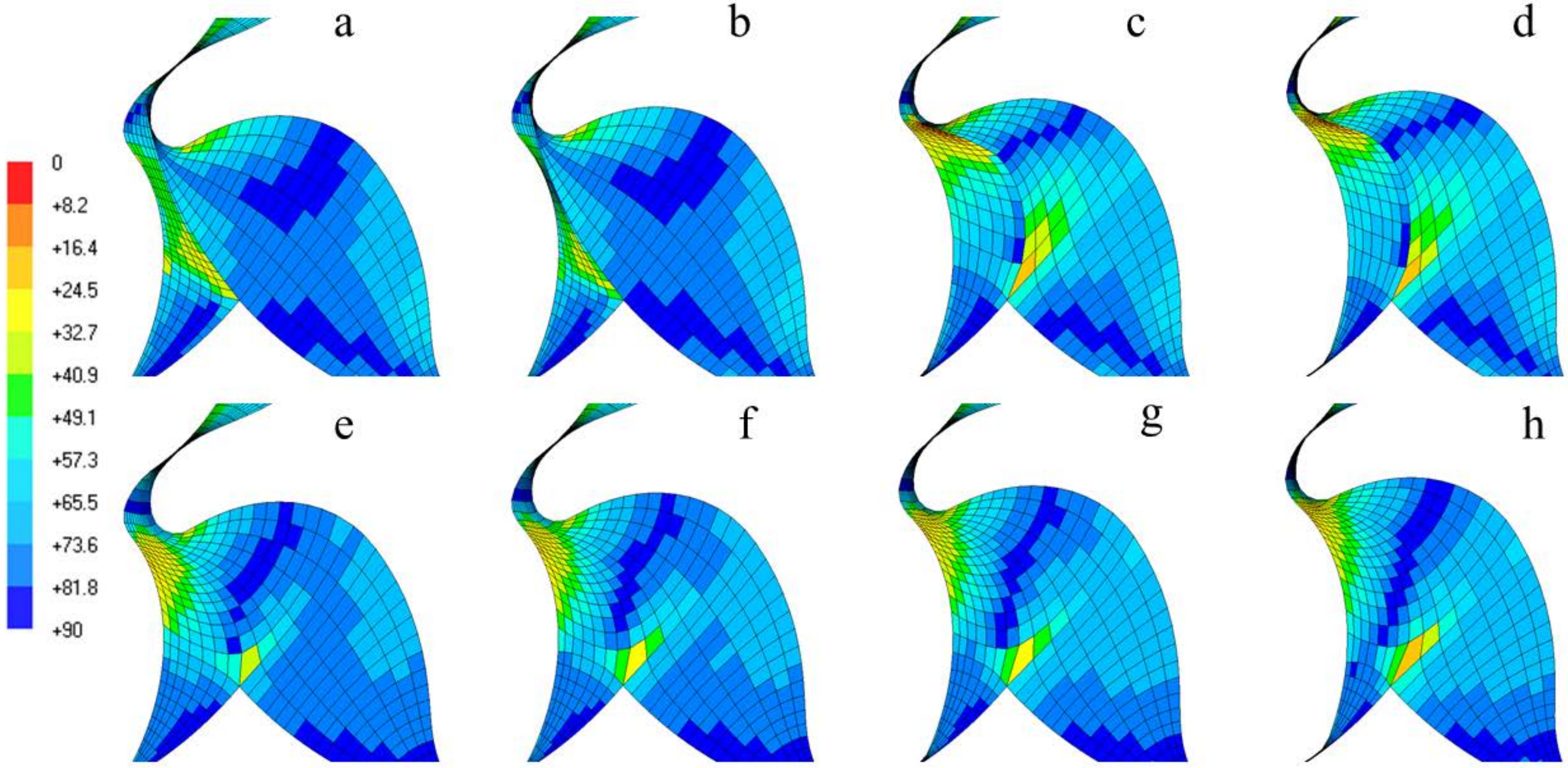
Grid transition in a cross section – mesh orthogonality

Case 1
Single
Domain
Algebraic

Case 2
Single
Domain
Differential

And

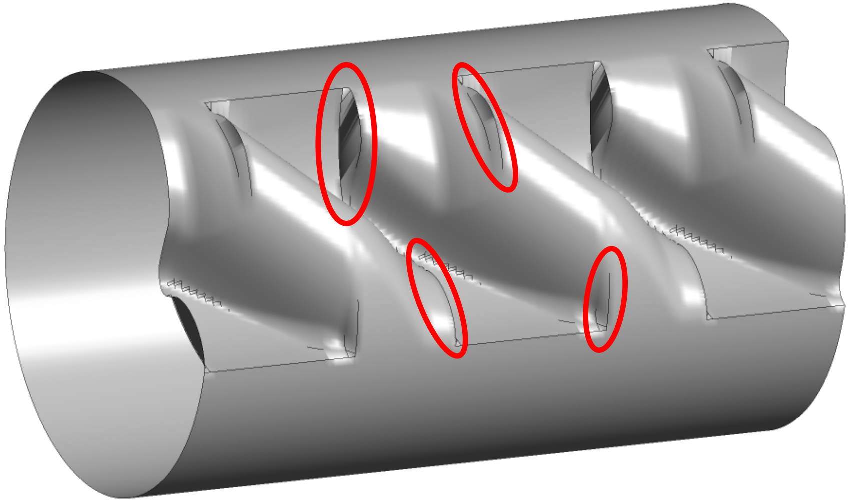
Case 3
Two Domain
Differential



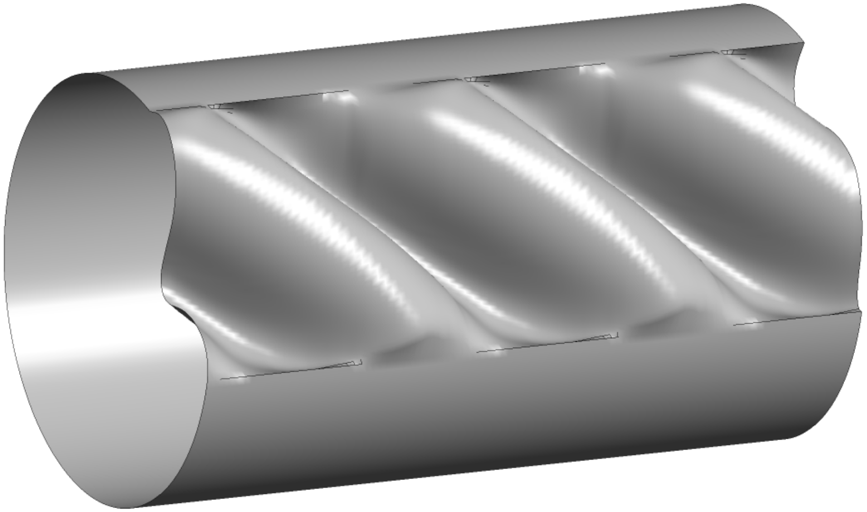
Oil Injected Compressor – Mesh type comparison

Grid transition of the connecting plane between two domains

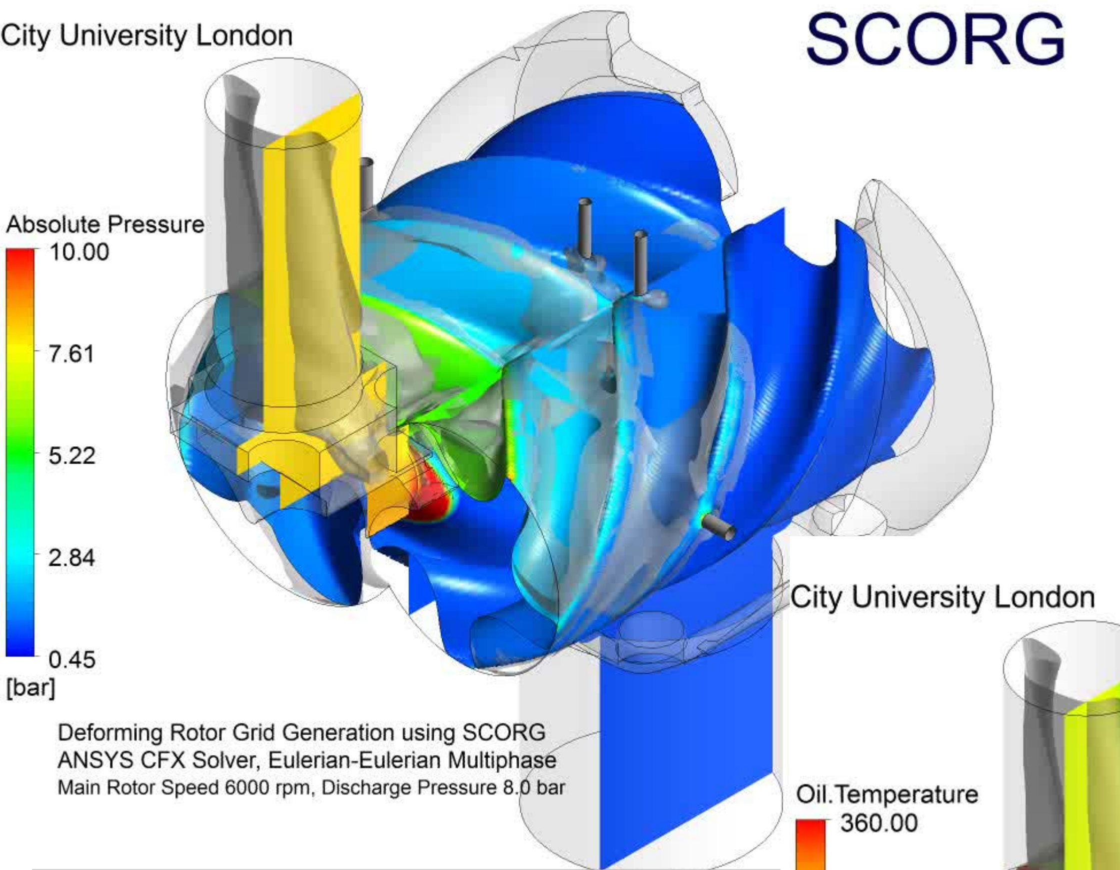
Case 1 and 3

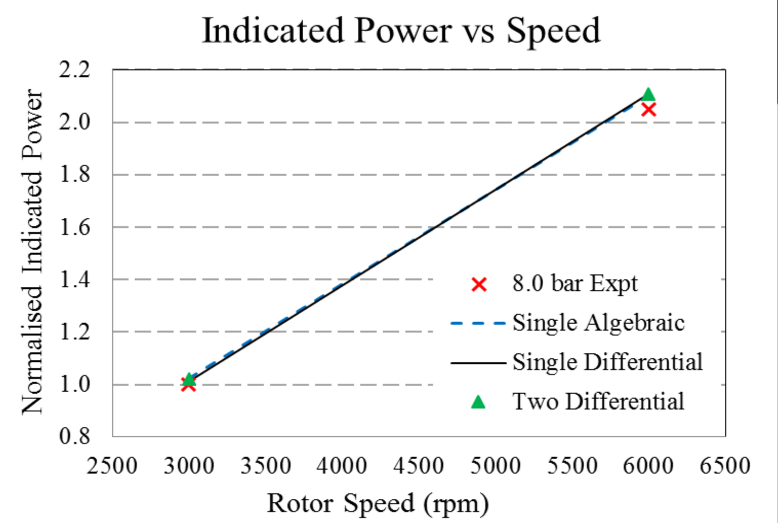


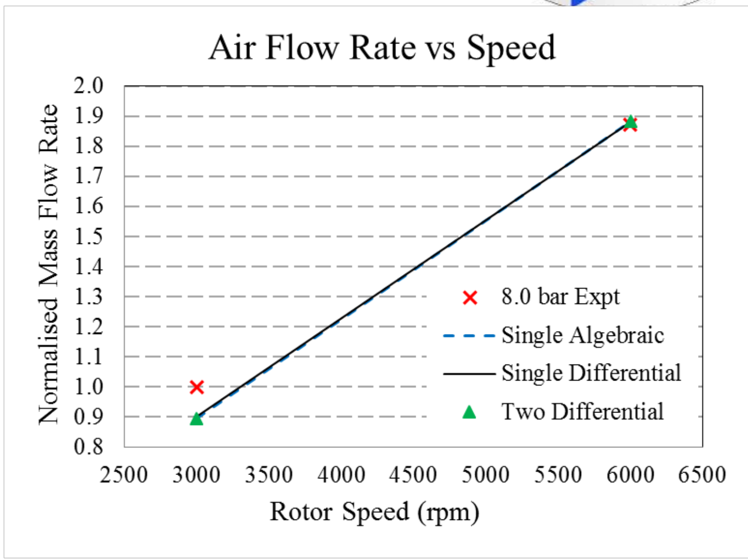
Case 1 Algebraic grid without interface smoothing

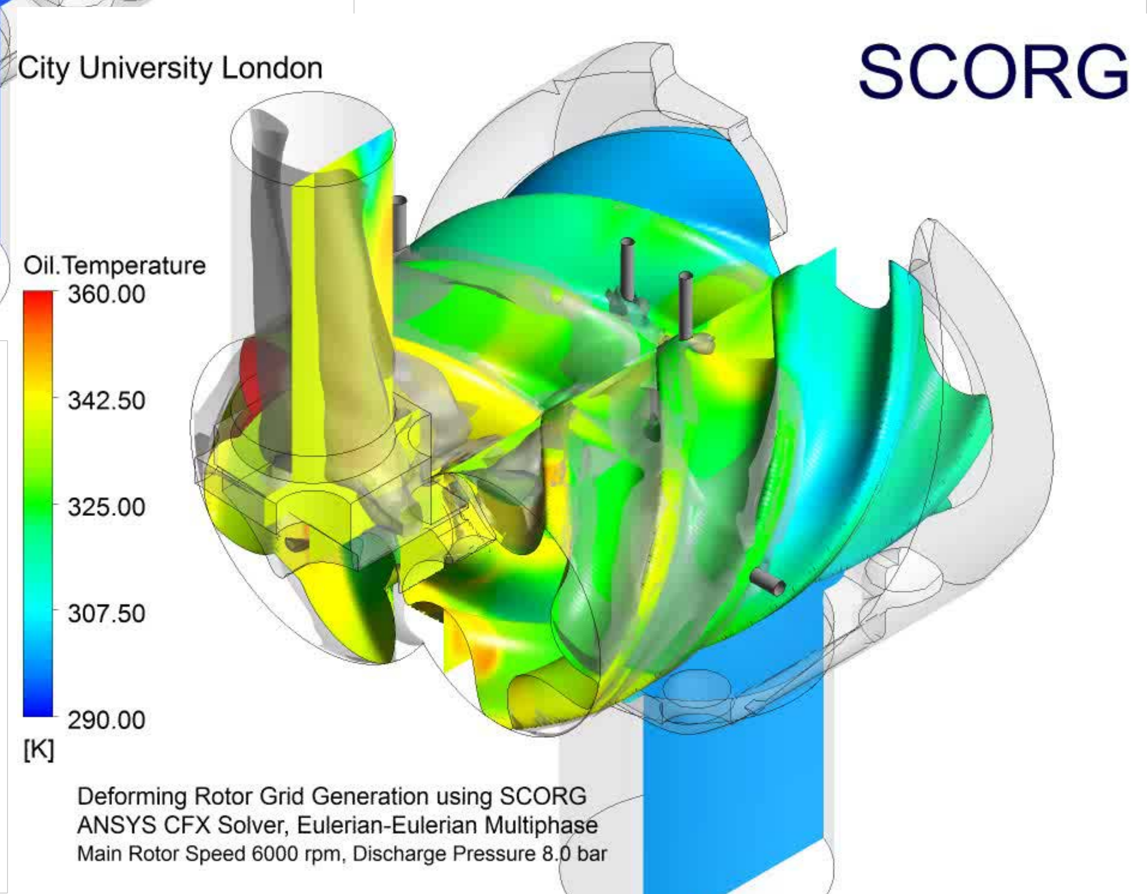


Case 3 Algebraic grid with differential interface smoothing

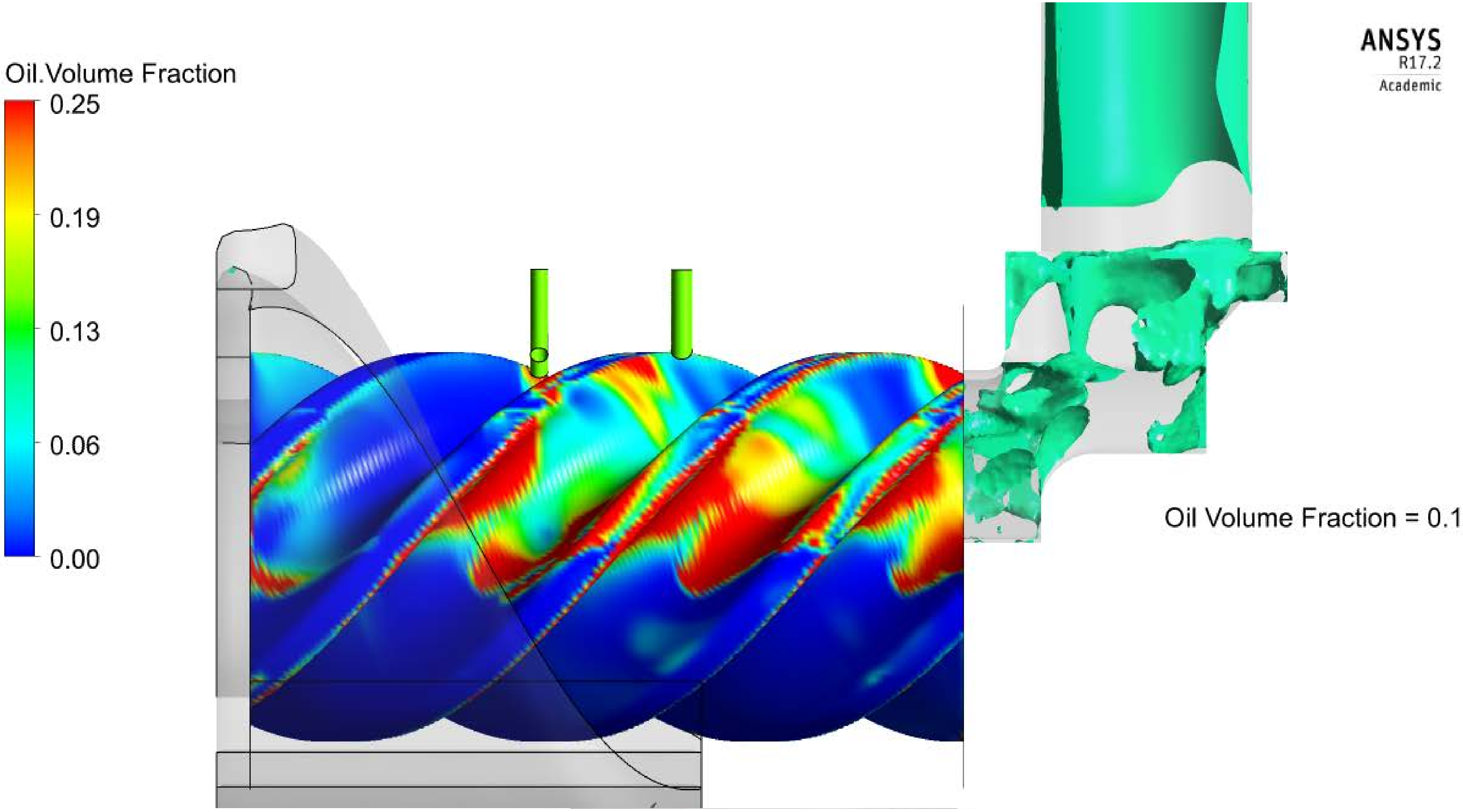

City University London
SCORG
Absolute Pressure
10.00
7.61
5.22
2.84
0.45
[bar]
Deforming Rotor Grid Generation using SCORG
ANSYS CFX Solver, Eulerian-Eulerian Multiphase
Main Rotor Speed 6000 rpm, Discharge Pressure 8.0 bar


Indicated Power vs Speed
Normalised Indicated Power
0.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2500
3000
3500
4000
4500
5000
5500
6000
6500
Rotor Speed (rpm)
8.0 bar Expt
Single Algebraic
Single Differential
Two Differential


Air Flow Rate vs Speed
Normalised Mass Flow Rate
0.8
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2.0
2500
3000
3500
4000
4500
5000
5500
6000
6500
Rotor Speed (rpm)
8.0 bar Expt
Single Algebraic
Single Differential
Two Differential


City University London
SCORG
Oil.Temperature
360.00
342.50
325.00
307.50
290.00
[K]
Deforming Rotor Grid Generation using SCORG
ANSYS CFX Solver, Eulerian-Eulerian Multiphase
Main Rotor Speed 6000 rpm, Discharge Pressure 8.0 bar

Results – Oil distribution



Oil Injected Compressor – Mesh type comparison

Solver Residual Levels

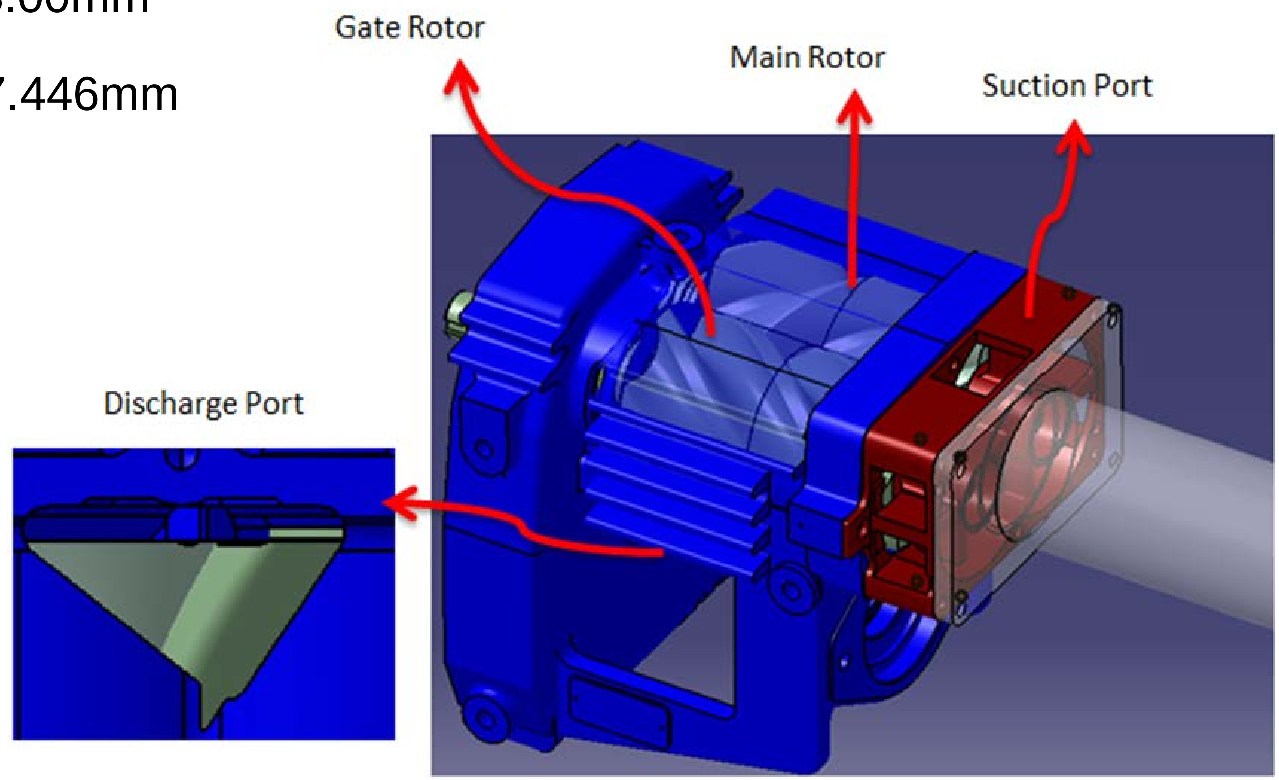
	RMS Residual at identical time step									
rpm	3000					6000				
	Momentum	Energy	Turbulent kinetic energy	Volume fraction	Courant Number	Momentum	Energy	Turbulent kinetic energy	Volume fraction	Courant Number
Case 1	6.8E-04	3.4E-03	6.1E-04	2.1E-04	9.60	1.8E-03	1.7E-03	1.0E-03	2.9E-04	6.33
Case 2	5.8E-04	1.8E-03	1.0E-03	2.1E-04	9.20	5.8E-04	1.1E-03	9.0E-04	2.5E-04	5.69
Case 3	8.2E-04	1.9E-03	1.2E-03	2.3E-04	8.50	6.9E-04	1.2E-03	1.3E-03	2.6E-04	5.55

- Data is collected at the final co-efficient loop iteration of converged time step corresponding to identical rotor position
- RMS residuals with the Differential Grid of Case 2 are better in comparison to Algebraic Grid in Case 1
- Lower Courant numbers in Cases 2 and 3 at both rotor speeds indicate better stability of the solver

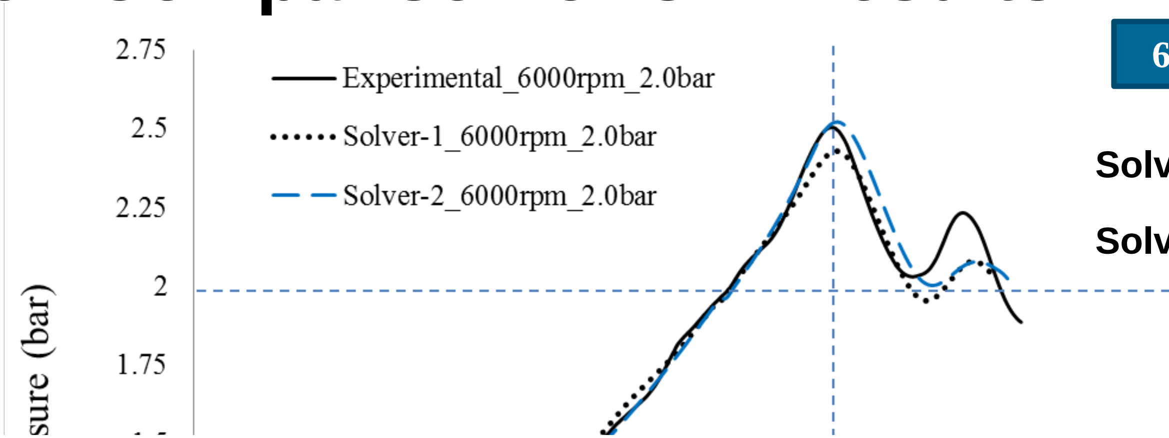
Oil free air compressor (with injection)

Two Domains Algebraic Mesh

- Drum XK18 3/5 ‘N’ Profile
- Centre Distance, 93.00mm
- Main Rotor OD, 127.446mm
- L/D Ratio, 1.6
- Wrap Angle, 280º
- Built in V_i, 1.8
- Clearances,
  - Interlobe **180 µm**
  - Radial **180µm**
  - End Axial **180µm**



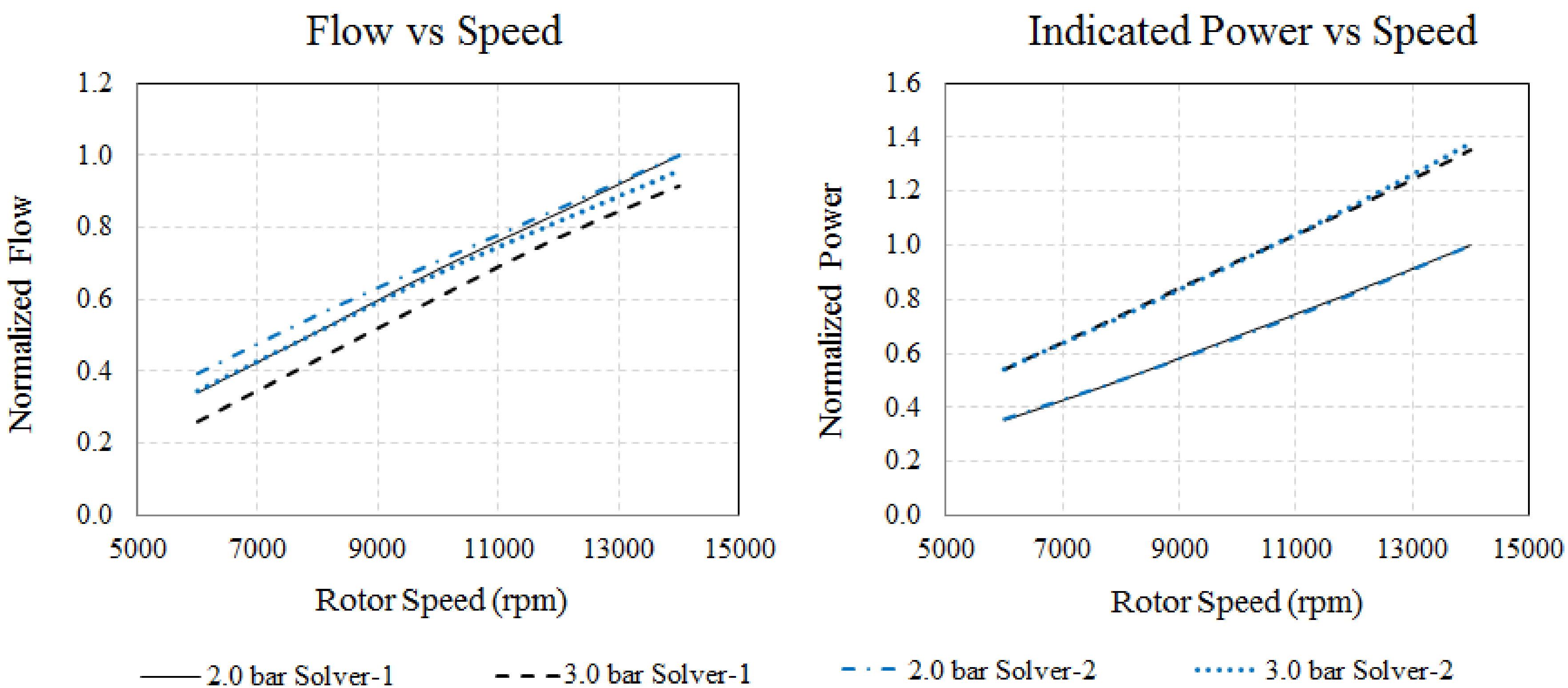
Oil free - Comparison of CFD results



6000 rpm, 2.0 bar

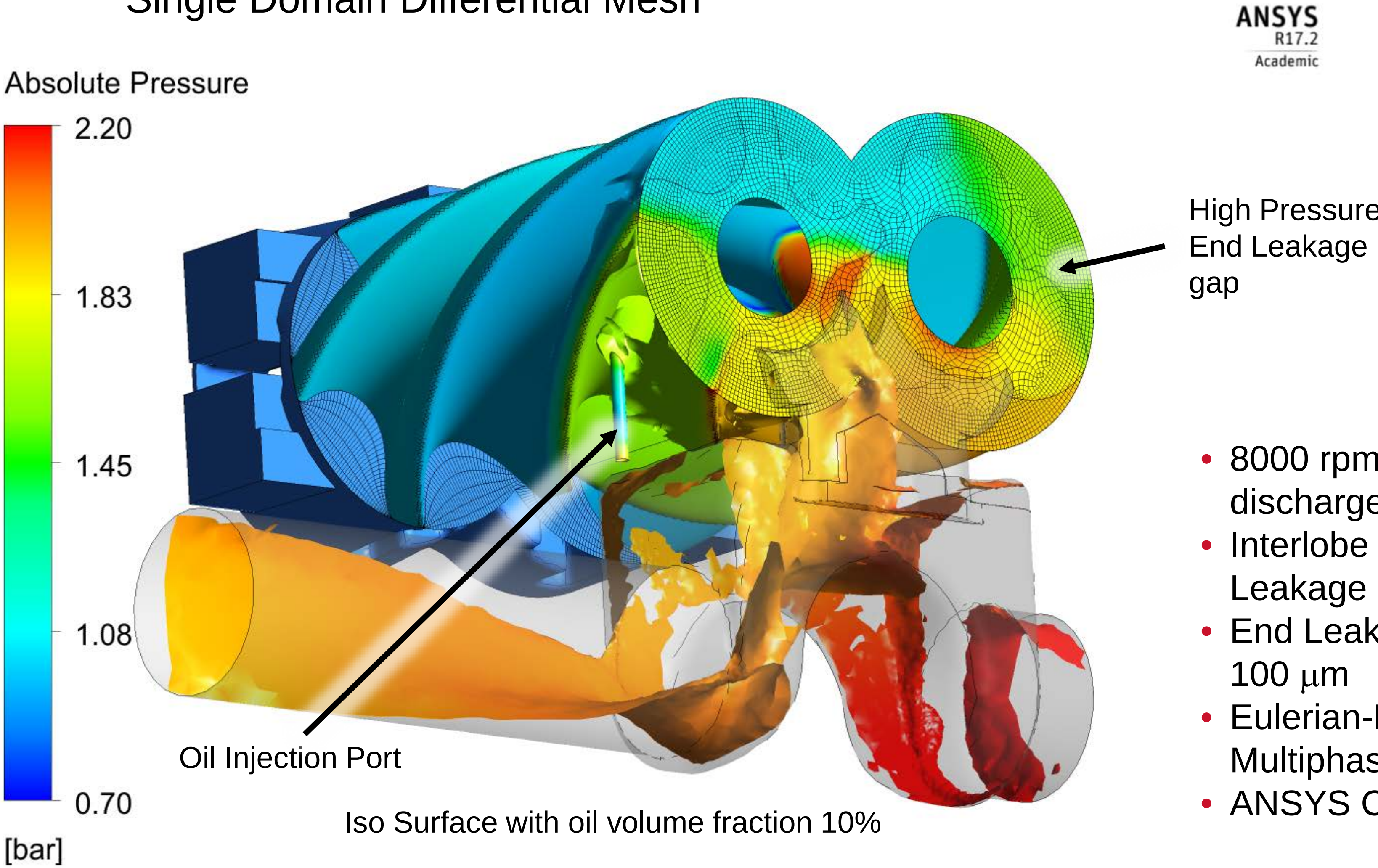
Solver-1 – Ansys CFX

Solver-2 – Simerics PD



DRUM 3/5 Compressor –Oil Injection with the end leakage gap

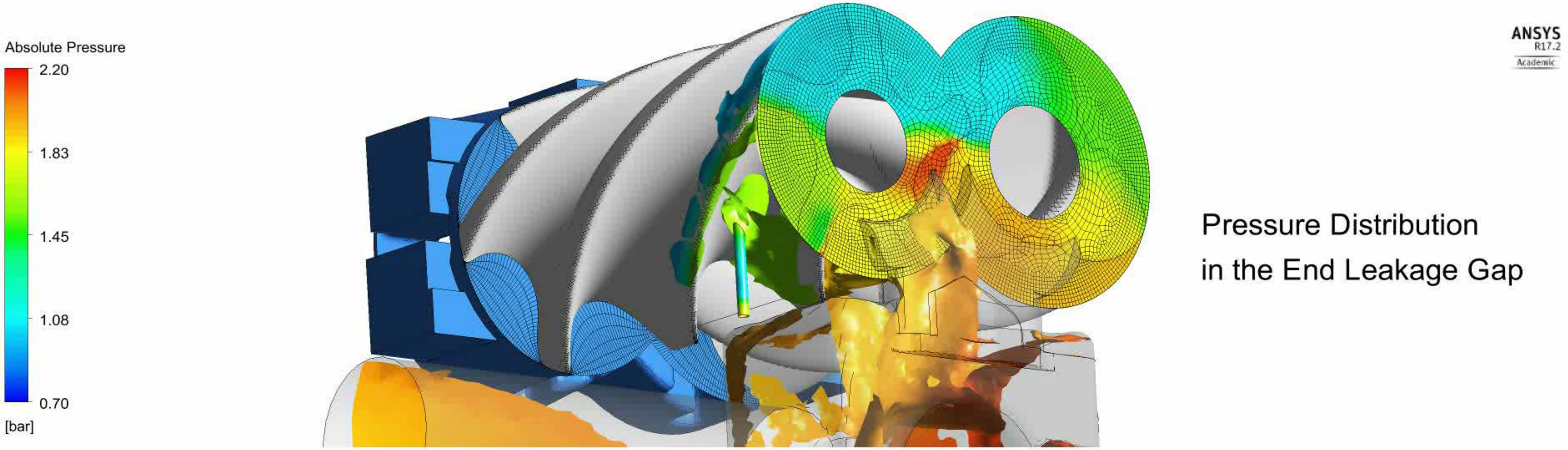
Single Domain Differential Mesh



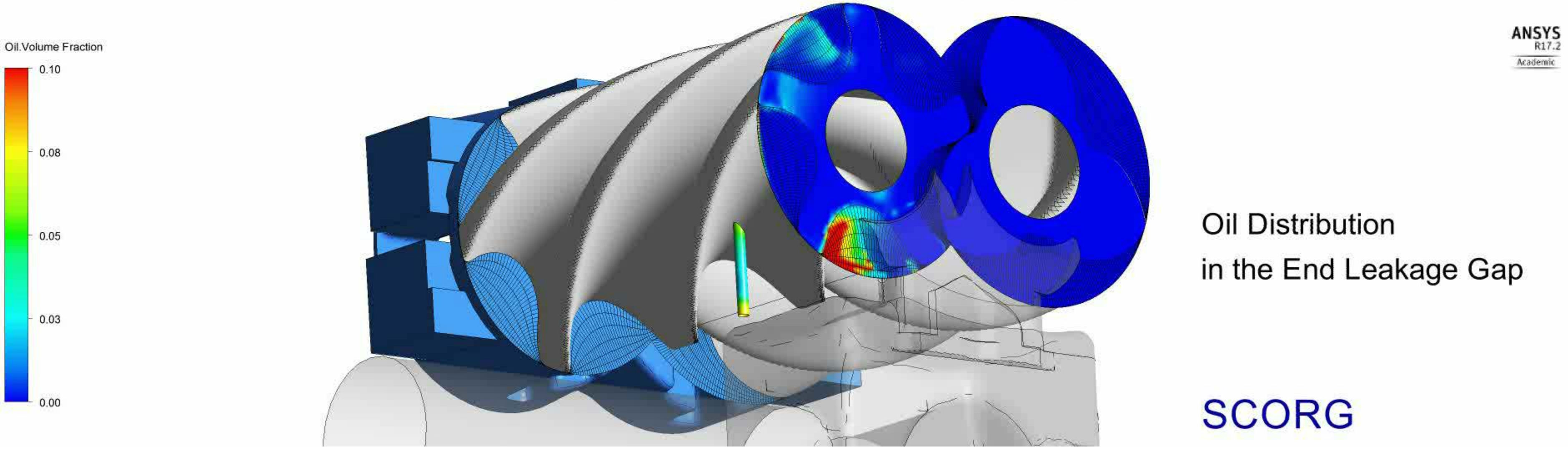
Iso Surface with oil volume fraction 10%

- 8000 rpm, 2.0 bar discharge pressure
- Interlobe and Radial Leakage gaps 60 μm
- End Leakage gap 100 μm
- Eulerian-Eulerian Multiphase Model
- ANSYS CFX Solver

ANSYS CFX – Post processing results



Pressure Distribution
in the End Leakage Gap

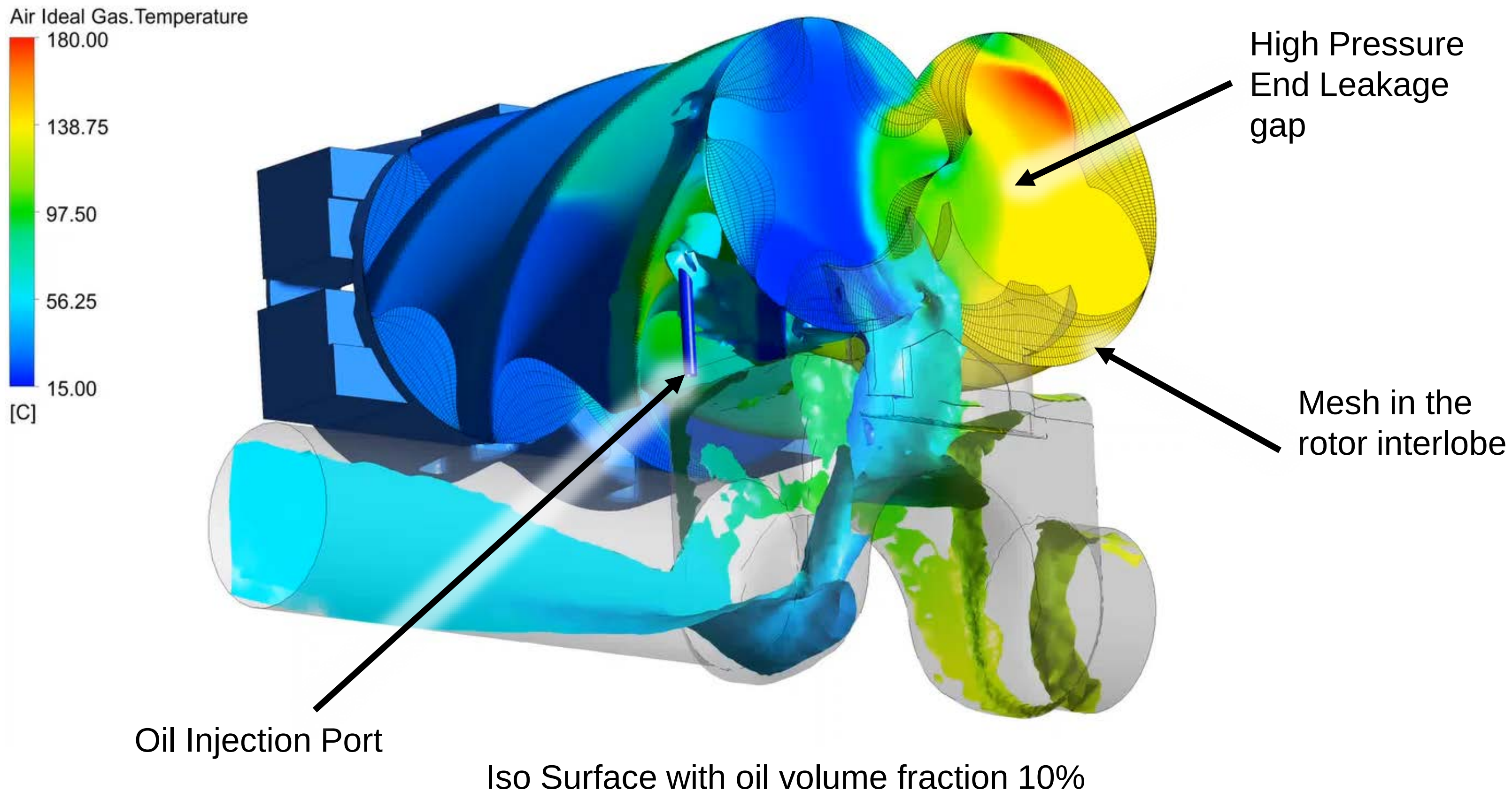


Oil Distribution
in the End Leakage Gap

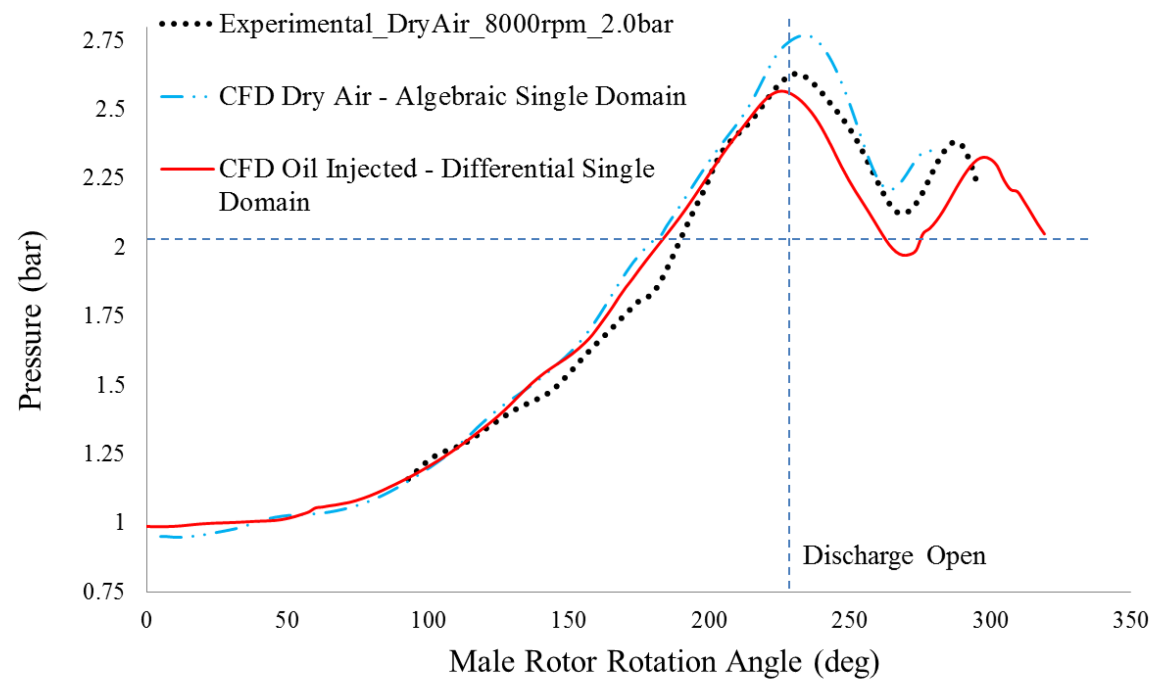
SCORG

DRUM 3/5 Compressor –Oil Injection with the end leakage gap

Temperature distribution



DRUM 3/5 Compressor –Oil Injection with the end leakage gap

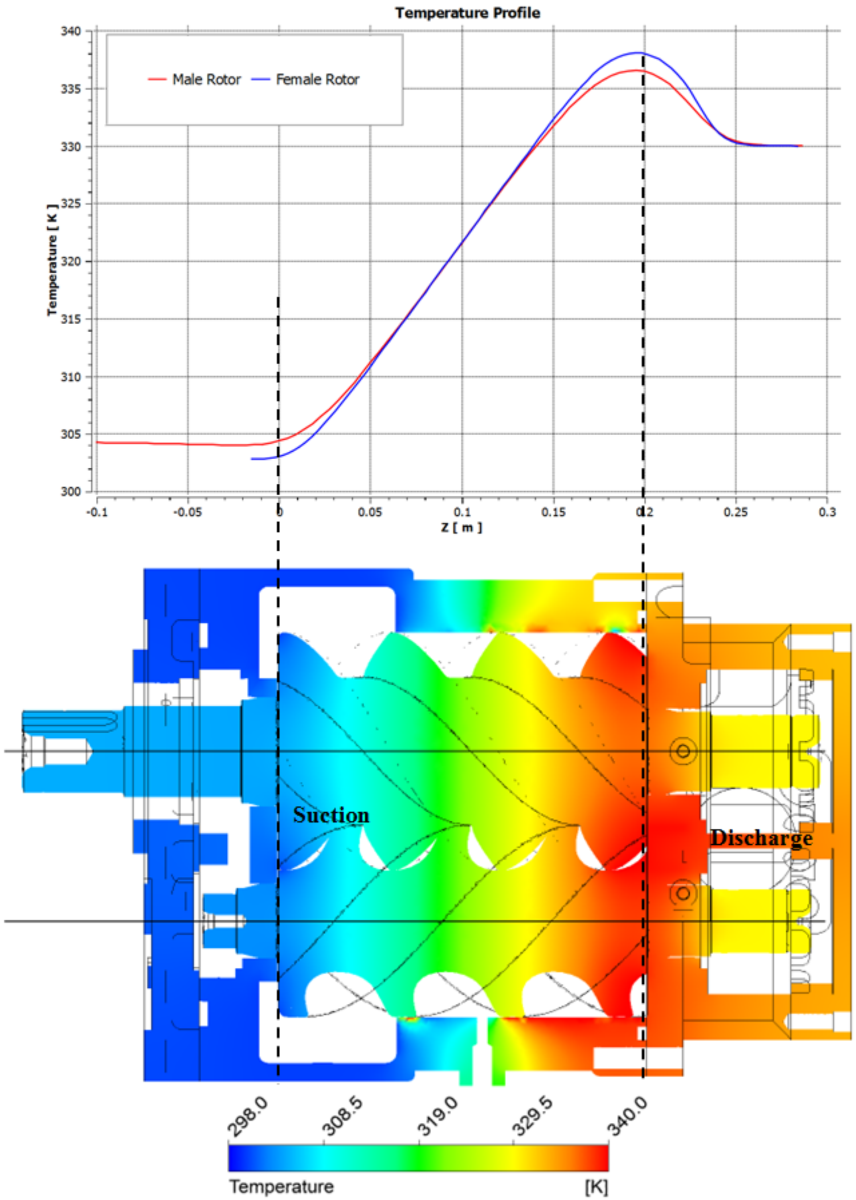


Case	Air flow rate (m3/min)	Volumetric Efficiency (%)	Indicated Power (kW)	Specific Power (kW/m3/min)	Discharge Temperature (°C)
Measurement – Dry Air	9.81	70.46	22.023	2.25	133.76
CFD - Dry Air	9.64	69.25	21.846	2.27	129.11
CFD – Oil Injected	10.53	75.63	21.078	2.00	58.29

Conjugate heat transfer using SCORG and Ansys-CFX

Comparison of temperature profile on the rotor center lines

- Temperature in the internal surface calculated from transient CFD analysis mapped into the model.
- Plot temperature along the centre line of the two rotors
- From gas temperature 300K at suction end to gas temperature 340K at discharge, uniform rise is noticed along the rotors.
- Reduction I temperature on shaft ends due cooling by oil
- Uniform temperature in rotor cross sections



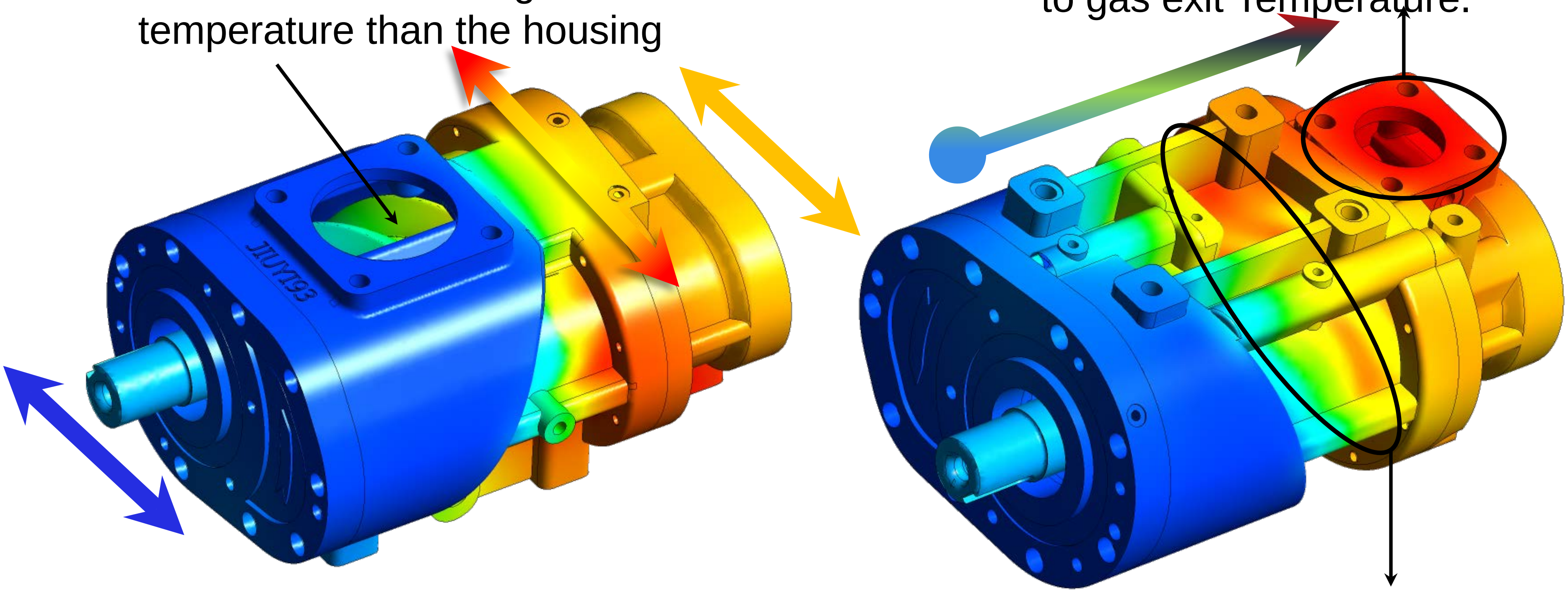
SCORG

ANSYS CFX - Post

Temperature distribution in the Casing Body – Exterior

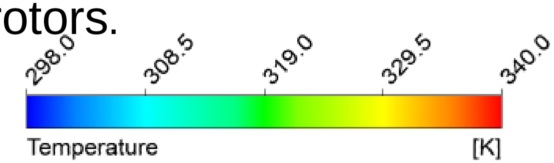
- Rotors inside have higher temperature than the housing

- Highest housing temperature at discharge port subjected to gas exit Temperature.



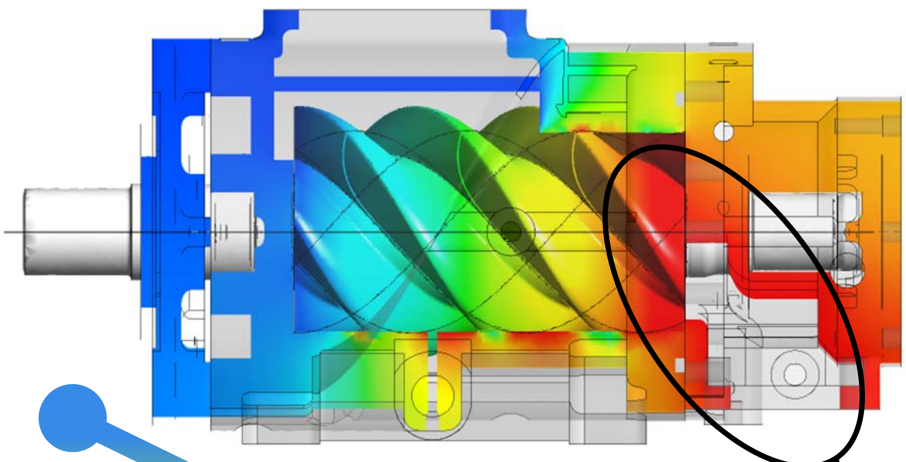
- More uniform temperature distribution across the width than along the length.
- Housing centre under high thermal gradients.

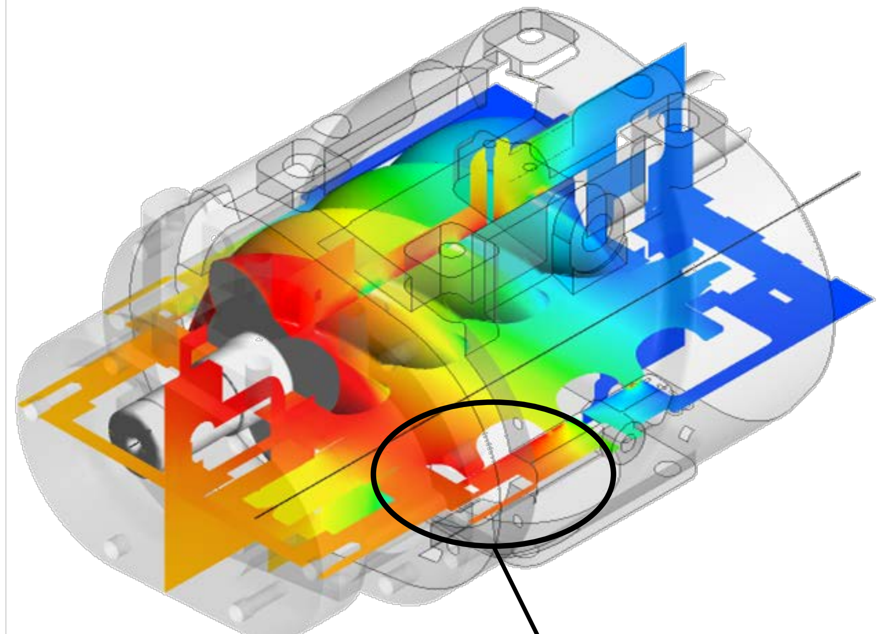
- Local temperature variation suggests that temperature distribution is not uniform regardless it is uniform on the rotors.



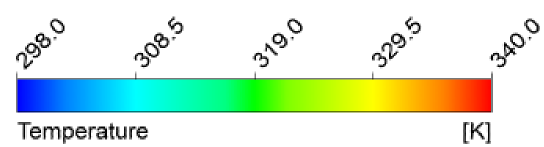
ANSYS CFX - Post

Temperature distribution in the Casing Body – Interior

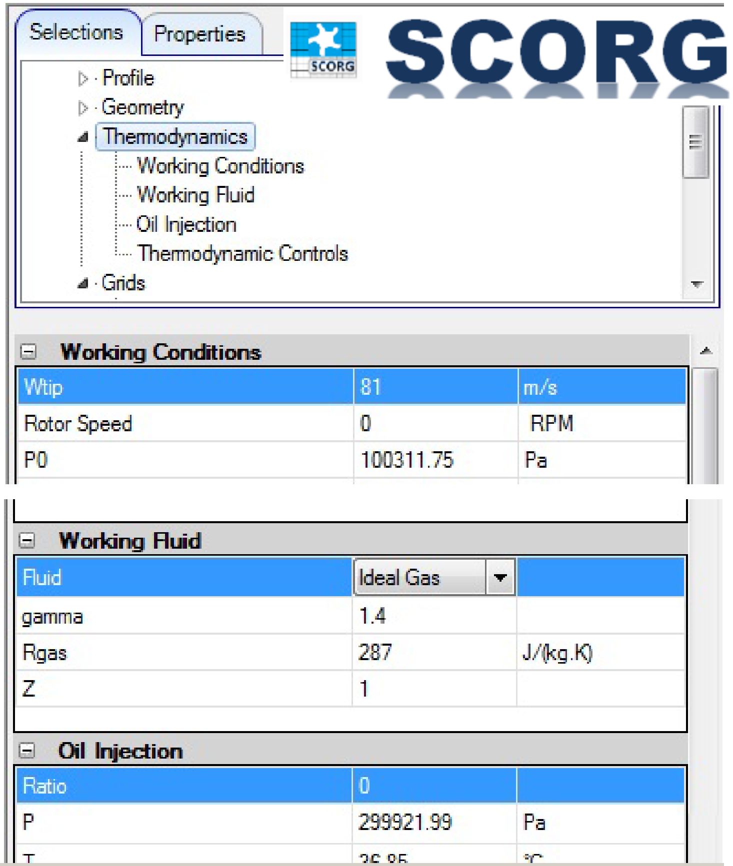




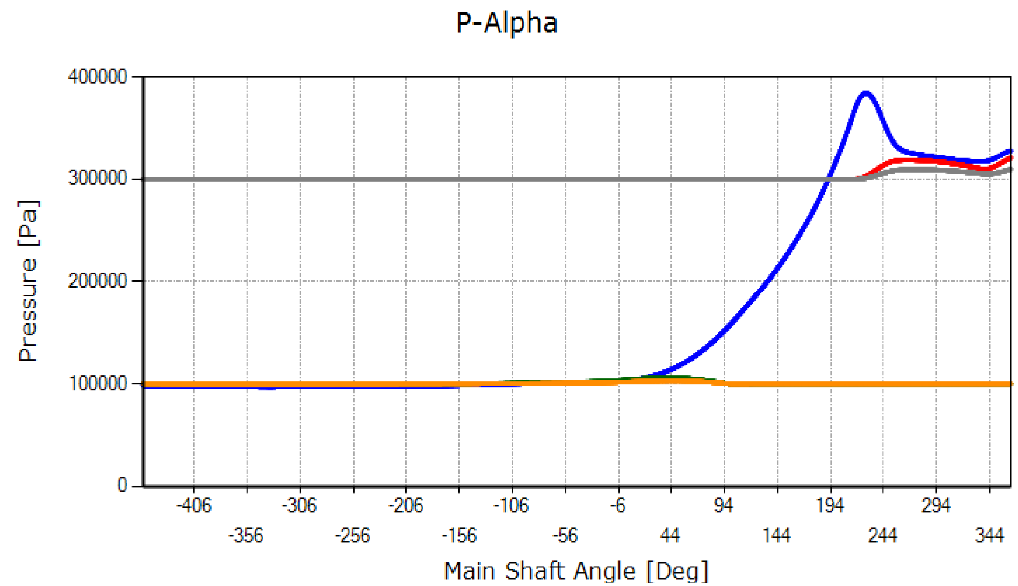
- In the central axial plane, temperature gradient is diagonal from top left suction bottom right discharge end
- High temperature zone with high gradients at discharge end conducting directly to the colder suction side
- Housing temperature is higher at the female rotor side than at the male side because of lower conduction.

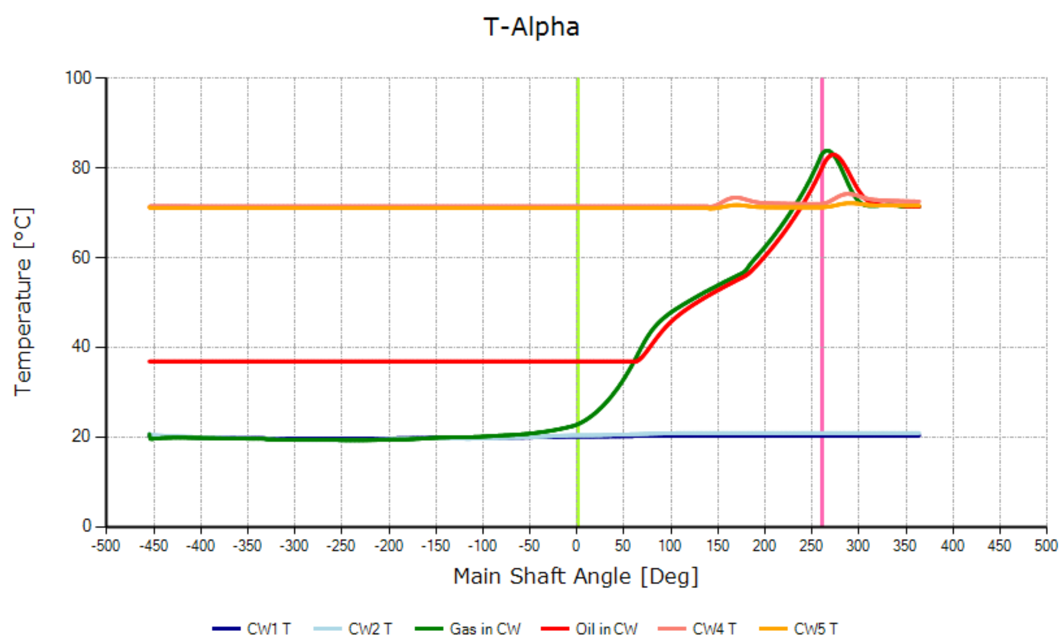


Conjugate heat transfer using SCORG thermodynamics

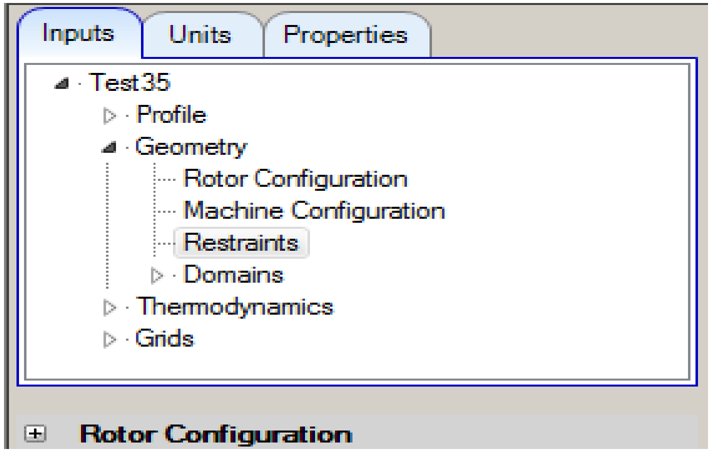


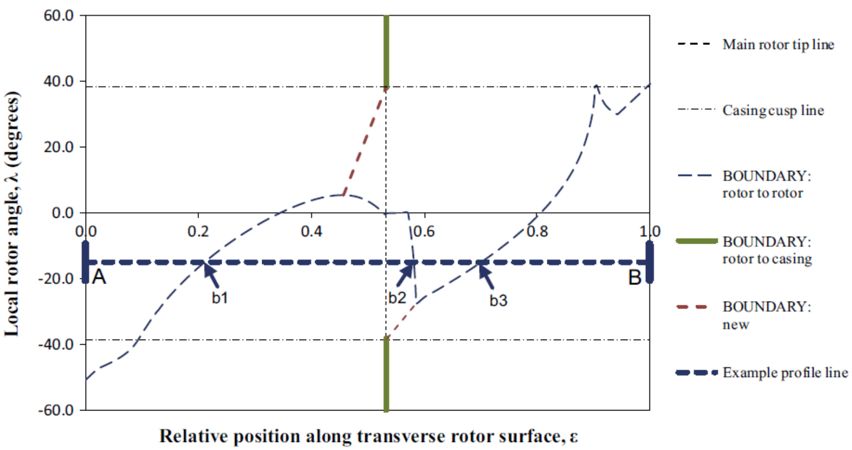

```
 RPM    Flow[m3/min]    Pow[kW]    P1[b]    P2[b]
7596.2    8.5751       39.87        1        3


Indicated Power     [kW]:32.77453
Shaft Seal Power    [kW]:2.27885
Bearing Power       [kW]:3.2544
Oil Drag Power      [kW]:1.56609
Total Shaft Power   [kW]:39.87387

Time elapsed:   00:00:00.7769604
```

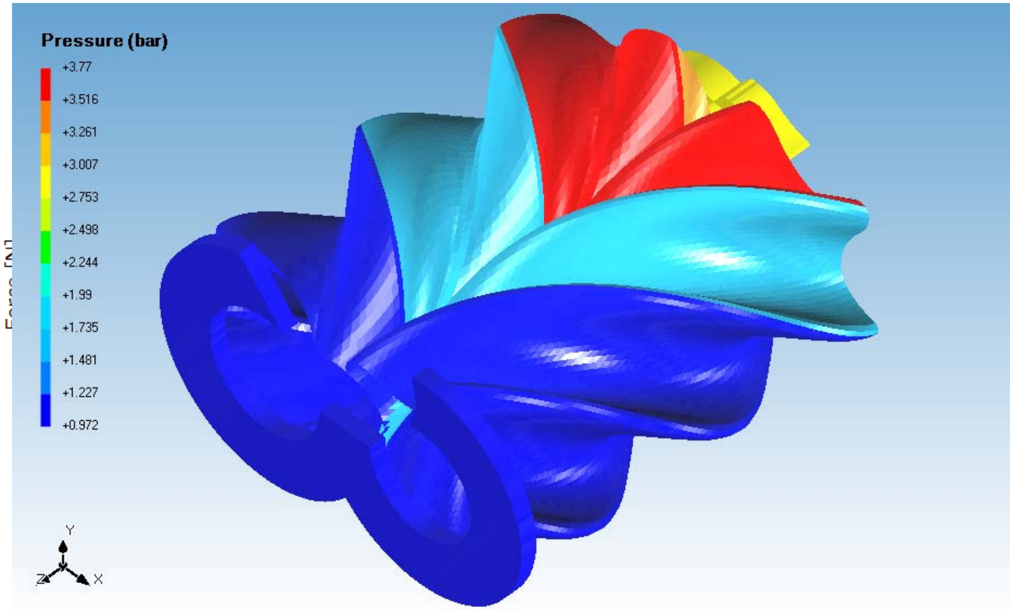




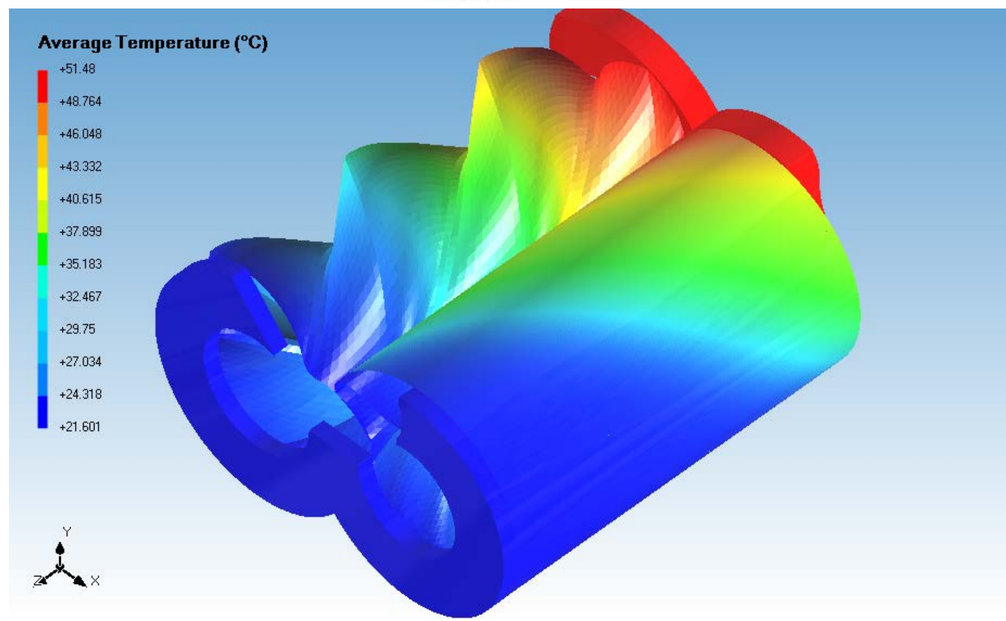
Conjugate heat transfer using SCORG post-processing



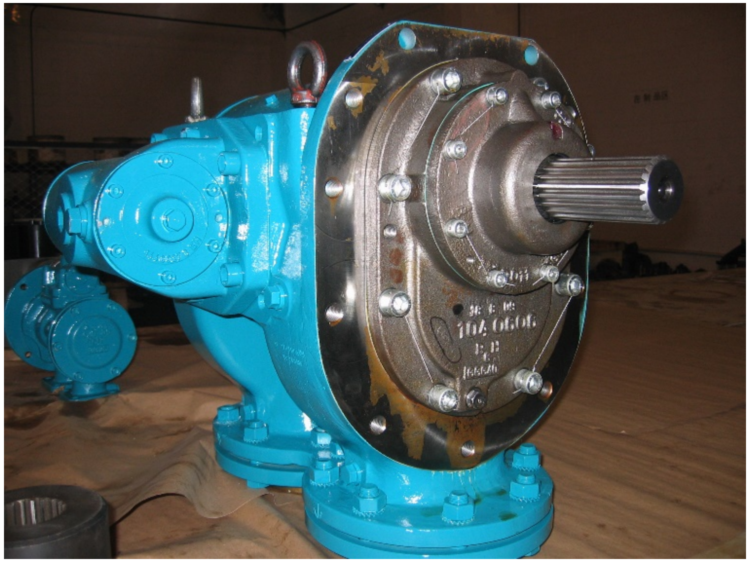


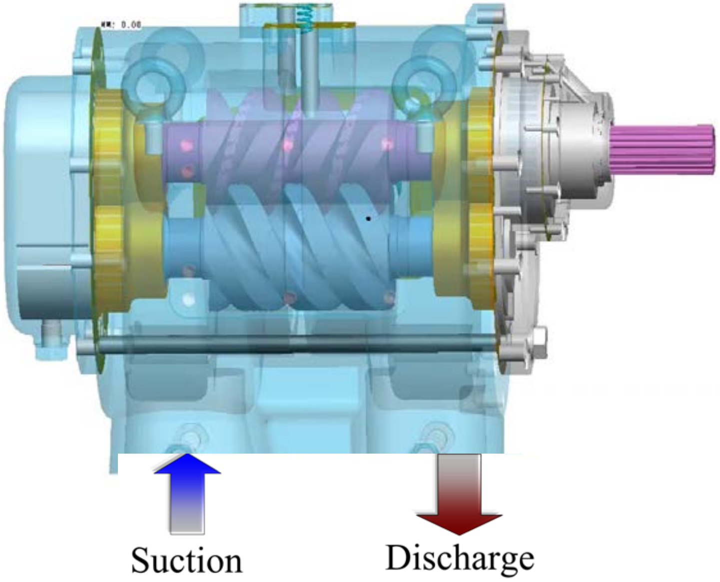


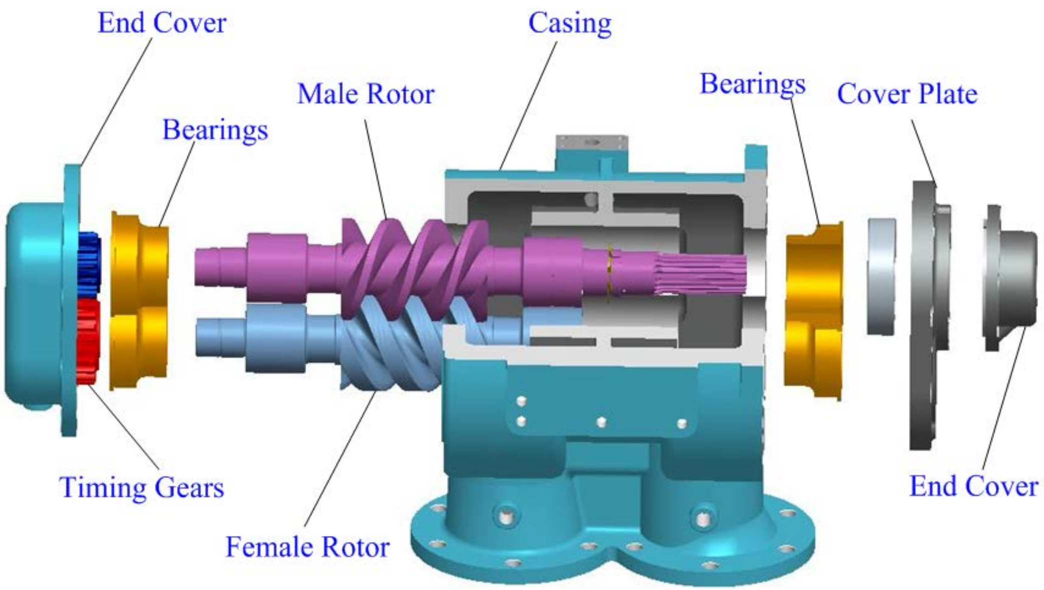

```
=== PROGRAM GEOM ===
  No of points on the profile:        400
SUBROUTINE      Cycle_map
  Mapping to casing bore, main side...
  Mapping to casing bore, gate side...
  Mapping to main rotor surface...
  Mapping to gate rotor surface...
END SUBROUTINE Cycle_map
```

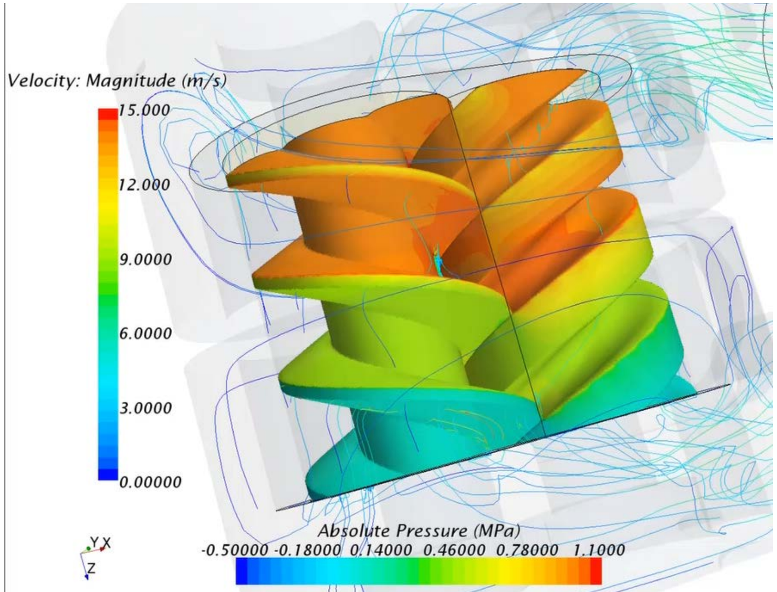


Liquid screw pump

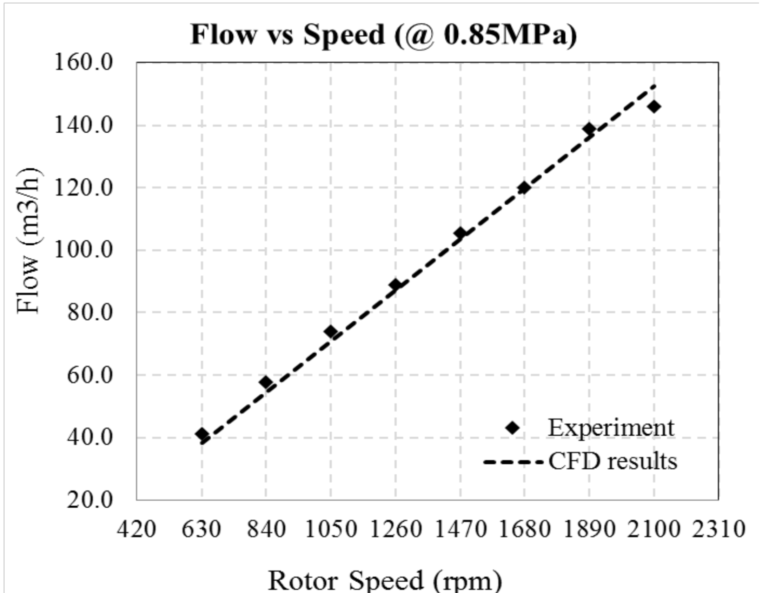


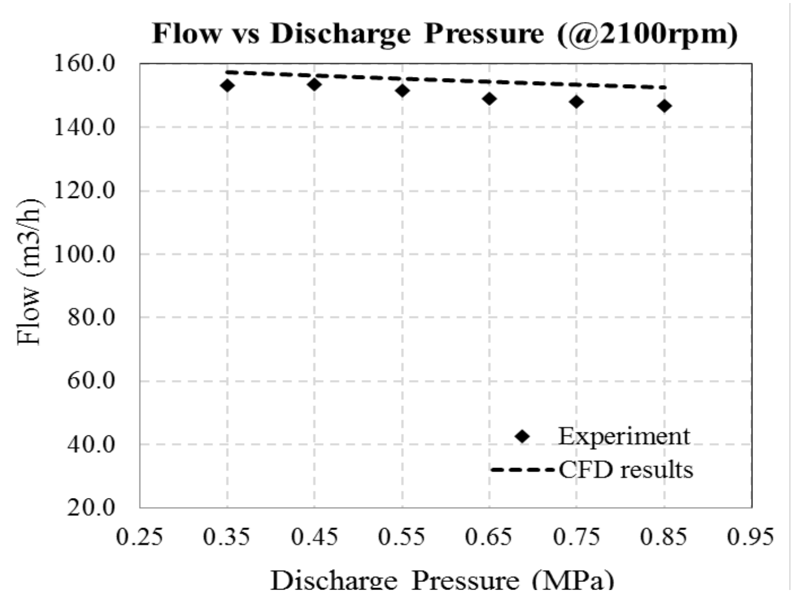


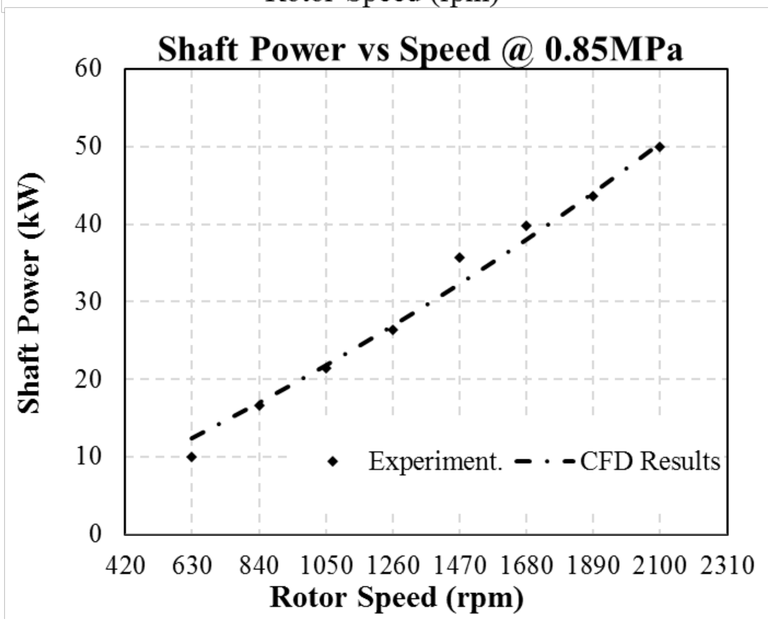


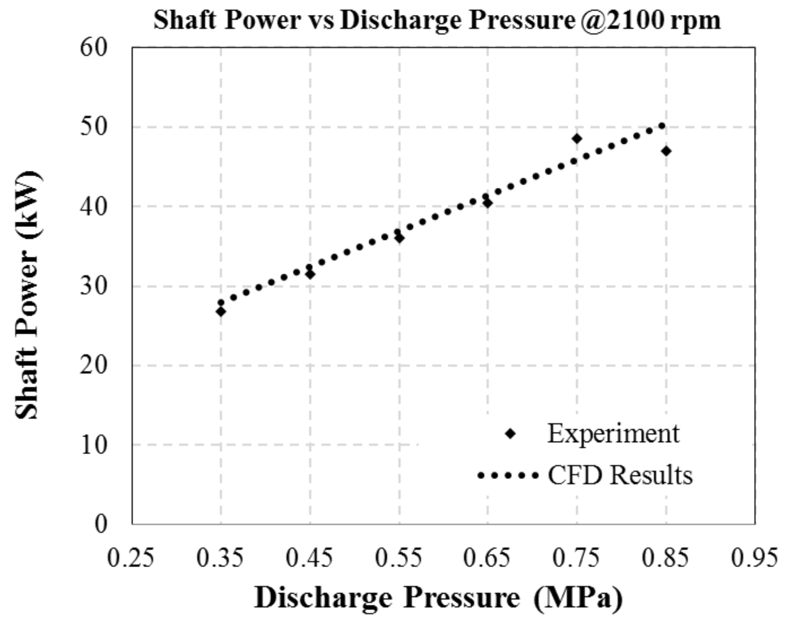


Liquid screw pump - validation





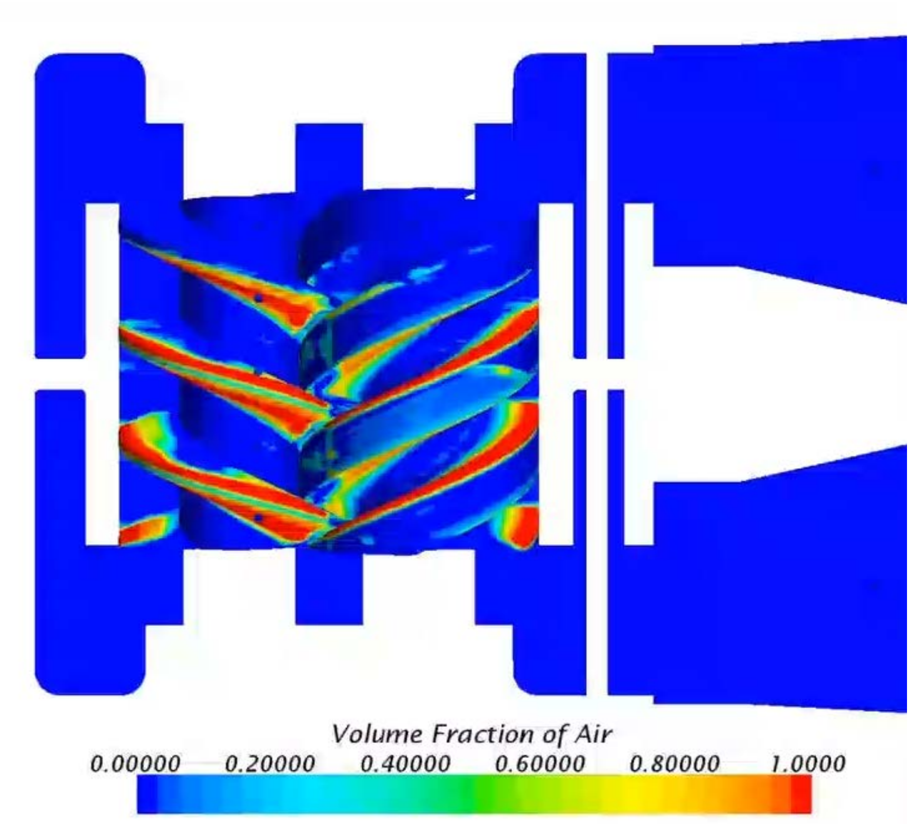




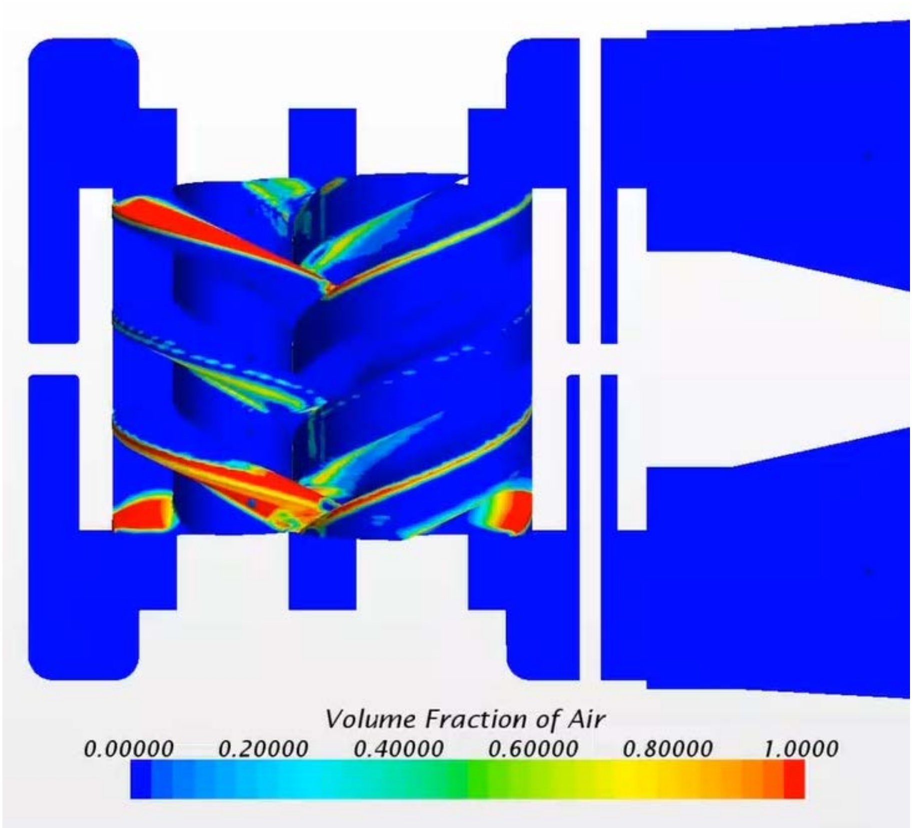
Liquid Screw Pump – Cavitation @ different speeds

'A' type rotor with CD40 lubricating oil

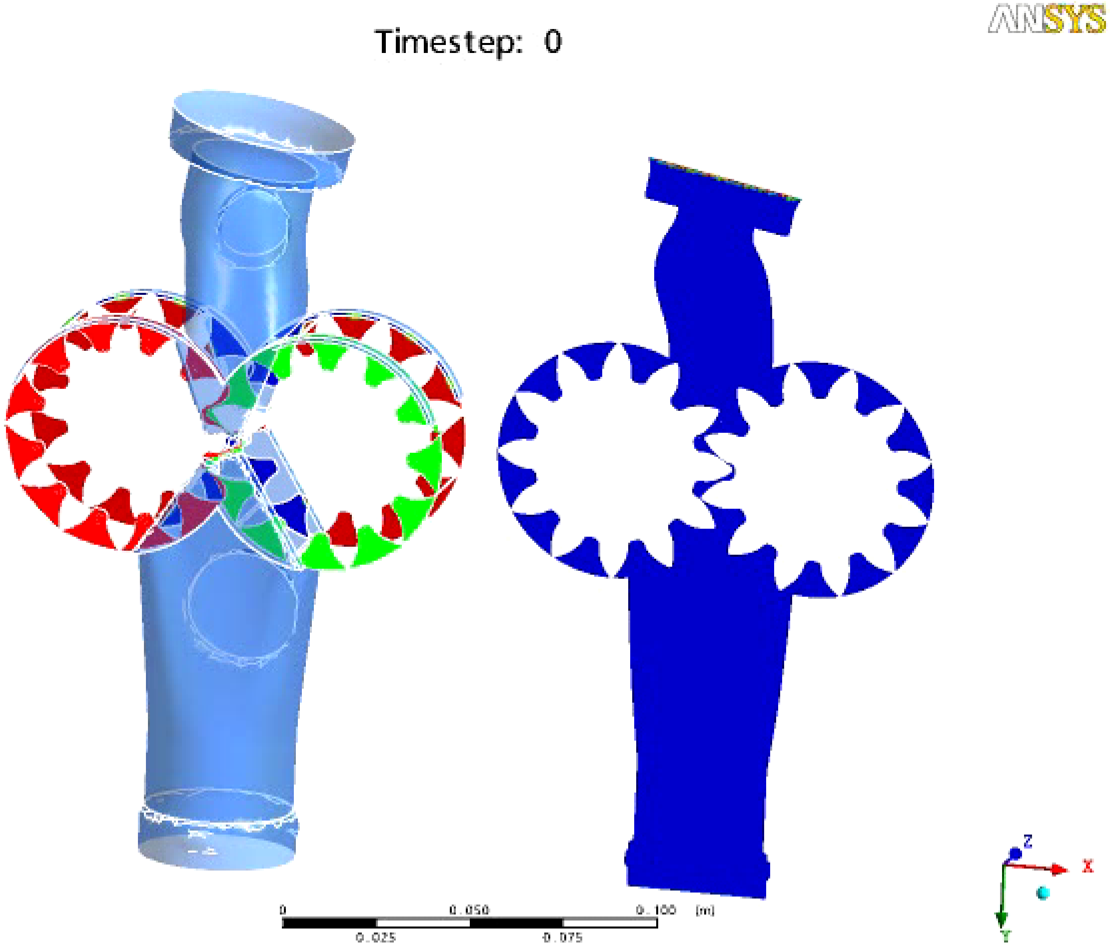
0.85 MPa Discharge Pressure, 630 rpm

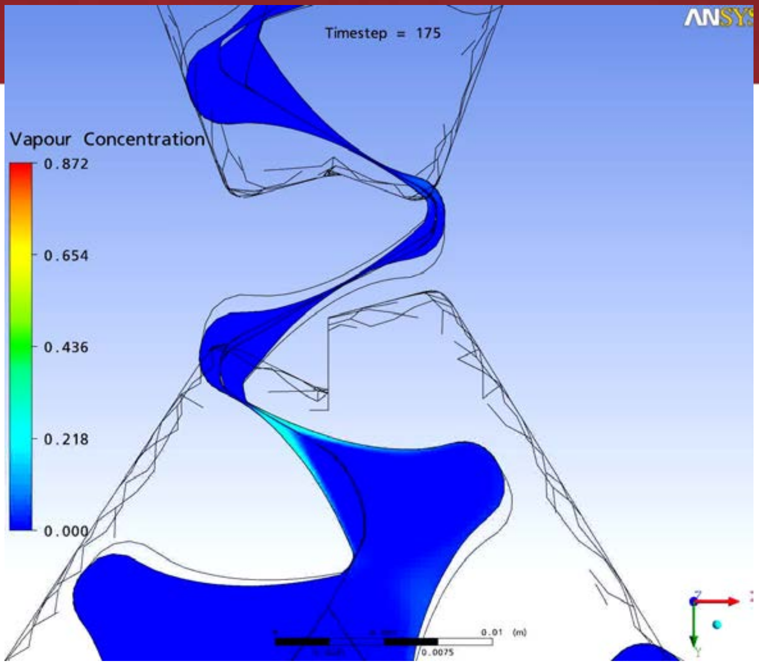


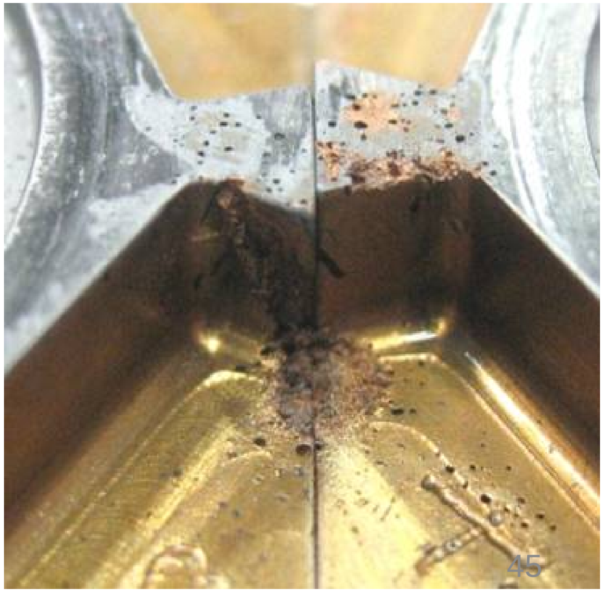
0.85 MPa Discharge Pressure, 2100 rpm



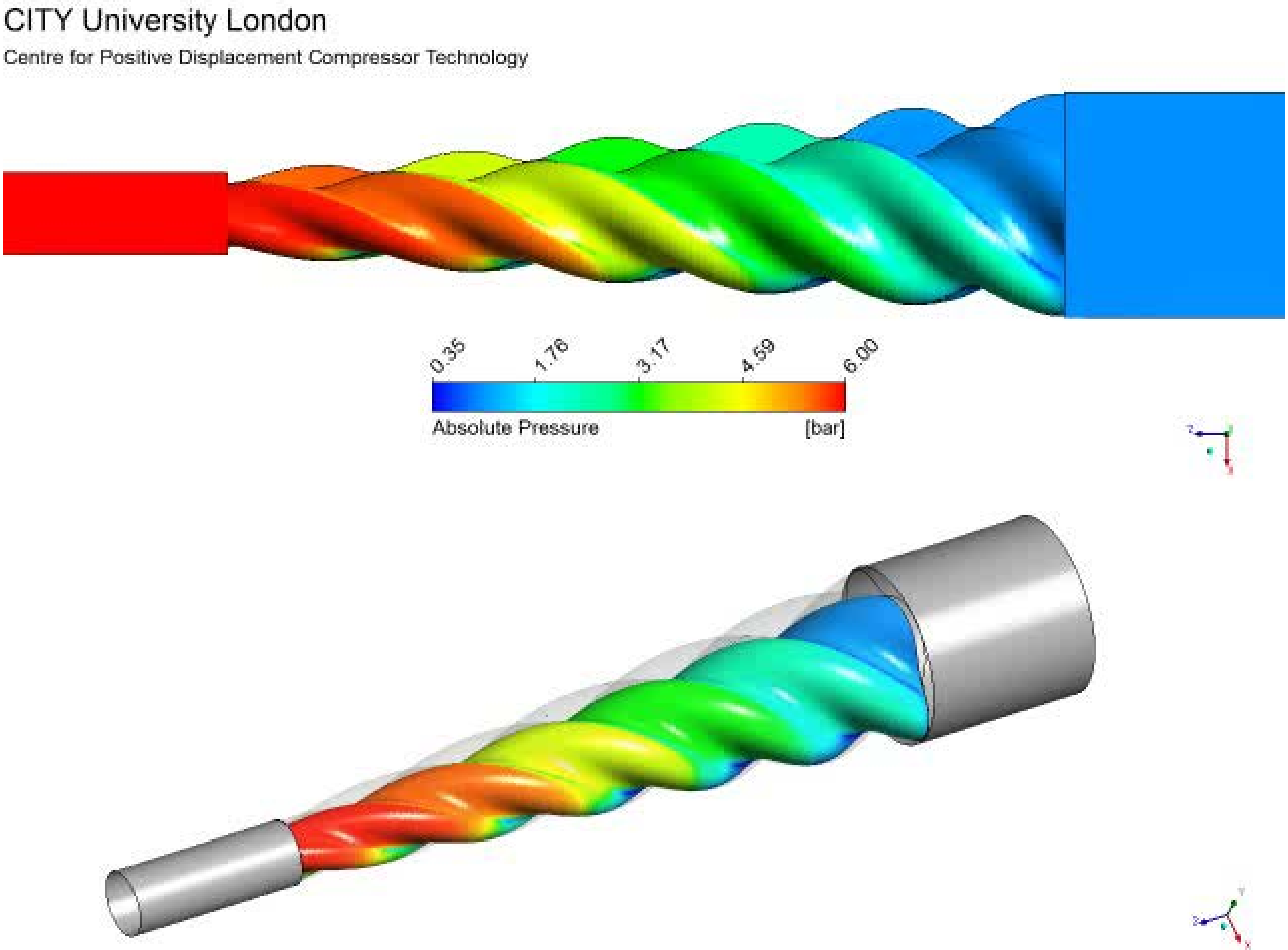
Gear fuel pump cavitation







Internally geared Screw Compressors



Conclusions

- CFD in screw machines is today readily available to be used fro modelling of *multiphase screw machines.*
- The key element required for successful CFD of these machines is *availability a good numerical mesh.*
- *SCORG™* is unique grid generator which allows fast and reliable multiphase CFD with Pumplinx, Ansys-CFX, Star-CCM+ and Fluent
- *Integrated with chamber modelling, SCORG™* enables full accurate and reliable analysis and improved performance of screw machines which contributes to reduction of carbon footprint.

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2. Use of CFD tools to predict variation in gap size.
3. Stability and accuracy of Multiphase flow calculations in 3D simulations of compressors.

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