MODELING and SIMULATION based DEVELOPMENT of ROTORDYNAMIC PUMPS

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Source: „SULZER Horizonte“, 1994
This Year‘s CADFEM User Meeting Motto is:

Success of a development project = 10 Simulations + 1 TEST

The purpose of this presentation is to prove the validity of this equation.
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1. Introduction
2. Sulzer AG contributions to CFD as applied in turbomachinery
3. Pump Analysis Workflow in ANSYS-CFX
4. Three CFD-based pump projects
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1. Introduction
A centrifugal pump

*Sulzer Pumps image*
Pump = Hardware + Performance

X * (Design, Build & TEST) (1830 until 1990's)
Pump Users’ requirements (Sales/Market requirements)

- short delivery time
- high efficiency
- overall performance according to specification or to a standard
- simplicity of construction
- reduced and easy maintenance
- durability
Pump performance requirements (T)

• guaranteed **head** rise and **efficiency** level at the BEP (Hoo, ETAoo, Nq) and at off-design points of operation

• **stable characteristics** up to a certain part-load, or over entire range of flow rate

• **Shut-off head**, Ho

• **SHAPES** of Q-H, and Q-ETA curves

• **cavitation free** operation

• **vibration free** operation

• remain within certain size envelope
Pump = Hardware + Performance

X * (Design, Build & TEST)  
(from 1830 till 1990’s)
Pump = Hardware + Performance

Successfully applied in pumps since mid-1990's: CAD & CFD

ABSTRACT
Flow field calculations of the NASA transonic axial compressor Rotor 37 were performed. These were carried out by two commercially available 3D Navier-Stokes solvers: STROPOS and TASCFlow using different turbulence models, i.e. SST-2 and k-ε. Some of the results were submitted to the CFD code assessment exercise organized in 1994 by the Turbulence Committee of the ASME, which is a joint of “t” CFDC code assessments were compared against previously observed experimental data taken in a NASA clean rig.

The objective of these calculations was to test this code in the same way as they were previously used by experienced engineers for standard industrial design tests. Thus, the effect involved in grid generation, flow simulation and postprocessing was subject to the usual limitations in computer resources as well as a diligent observation of cost-effectiveness (reasonable available computing power). Therefore, the flow field was predicted using two different turbulence models: SST-2 and k-ε. Generally, the results of both cases showed good agreement with respect to the measured overall performance characteristics and averaged aerodynamic distribution. It is clear that the SST-2 model is more accurate in predicting the flow field in the intermediate region.

INTRODUCTION
In 1994 the Turbulence Committee of the ASME issued an invitation for CFD calculations on a “shock free” flow field of the STI-100 axial compressor. The code assessors were asked to submit their results with two different turbulence models: SST-2 and k-ε. Generally, the results of both cases showed good agreement with respect to the measured overall performance characteristics and averaged aerodynamic distribution. In particular, the SST-2 model is more accurate in predicting the flow field in the intermediate region.
The industry needs:

- simple to use,
- low labor intensive techniques, which
- guarantee the optimum solution, on
  the first time basis
Sulzer (Pumps, and Innotec) contributions to CFD and its applications (1990-2008)

1. CFD software evaluation re. workflow and validation (CFD results vs. test)
2. Pump analysis for Q-H, Q-ETA predictions – „CFD learning curves“
3. \( H_{CFD}/H_{Test} = f(Nq) \) CFD prediction for pumps / „CFD accuracy diagram“
4. Radial Compressor Impeller internal flowfield (DLR Köln – Eckardt case)
5. Axial Compressor Impeller ASME blind test (NASA Rotor 37)
6. (EPRI project) Radial Pump Stage *)
7. Radial Pump Vaned Diffuser
8. **Mixed Flow Pump (Spiral Volute) – Pump Nq90**
10. Inlet Sump for Vertical Cooling Pumps
11. *) Rotordynamics ( =>Texas A&M, NASA)
12. Cavitation
13. Erosion
14. CFD „best practises“ and application guidelines
CFD vs. Experiment: Q-H, Q-ETA

IMPELLER CHARACTERISTICS: STATIC PRESSURE RISE $\psi$ AND IMPPELLER HYDRAULIC EFFICIENCY
CFD vs. Experiment
Head prediction error estimation

$n_q$ is a pump design factor; $n_q \leq 10$ for purely RADIAL flow impellers, $n_q \geq 200$ for AXIAL flow impellers
CFD vs. Experiment – internal flows in compressors (DLR-Köln Eckardt‘s impeller)
Sulzer Pumps Radial Flow BFP stage for CFD Analysis and Research

FIGURE 3: COMPUTATIONAL DOMAIN FOR THE RADIAL PUMP DIFFUSER SIMULATIONS

FIGURE 4a: COMPUTATIONAL GRIDS FOR RADIAL PUMP IMPELLER AND DIFFUSER

FIGURE 4b: H-GRID (68750 NODES) FOR ONE PARTITION OF THE RADIAL PUMP DIFFUSER
CFD vs. Experiment: Diffuser flow and the static pressure recovery $C_p$
CFD vs. Experiment using LDA
CFD vs. Experiment: flow velocities at impeller inlet and outlet

Figure 5: Comparison of velocity profiles at the inlet

Figure 6: Comparison of an exit velocity profile (TASCflow H / Grid 50% flow)
Texas A&M research re. the secondary flow path on baseline Sulzer Pumps‘ „EGGER impeller“
Sulzer-ETHZ Research Pump Nq90

Fig. 1: Test impeller and geometric specifications

Median view pressure side
PSH  PSS
median view suction side

Front view section side

Front view pressure side

Fig. 2: Pressure transducer positions in impeller passage

\[
\frac{\Delta p_{\text{tot}}}{\rho_2 (R_2 \omega)^2} = \int_{S_1} \left( p + \frac{1}{2} \rho \omega^2 \right) c_x \, dS
\]

Fine grid computations
Coarse grid computations
Probe measurements

Fig. 17: Pump characteristics: comparison of integrated experimental and numerical data at the runner outlet and of the global pump head
Sulzer-ETHZ Research Pump Nq90
(grids generated for TASCflow, 1996)

Fig. 3: Detailed view of the main part with the runner starting at N = 51 and ending at N = 76

Fig. 4: Surface grid on the hub with the "pinch" at the leading edge

Fig. 5: Surface grid on the hub with the "pinch" at the trailing edge
Sulzer Pumps Mixed-Flow STAGE
CFD performance prediction over entire flow range
Sulzer Pumps CFD Research on:
A. Shut-off Head (centrifugal pumps)
B. Multiphase Pumps

CFD Model and Results for low flow conditions of radial flow pump

EPFL images

Multiphase Pump CFD Analysis
Industrial CAD/CFD/FEA based development of pumps

Figure 1: CAx-system for development of pump impellers

CFD Modeling and Simulation Engine

impeller preliminary design

impeller final design solution
3. Pump Analysis Workflow in ANSYS-CFX

3.1 Sulzer – ETH Zürich Research Pump Nq90, revisited, using ANSYS-CFX

3.2 a double suction pump analysis in ANSYS-CFX
3.1 Sulzer-ETHZ Research Pump Nq90, revisited using ANSYS-CFX
3.1 Sulzer-ETHZ Research Pump Nq90, revisited using ANSYS-CFX

**CFX – preprocessing**
Turbo Tool largely facilitates the CFD model build for any bladed component => e.g. Boundary Surfaces and Conditions assigned automatically

Grid generated in **TurboGrid** using ATM
3.1 Sulzer-ETHZ Research Pump Nq90, revisited using ANSYS-CFX

**CFX predicted Q-H instability at the flow rate it was measured**

**CFX correctly predicted Hoo and H at several flow rates**

Structured 250000 grid generated in TurboGrid using ATM, only small adjustments were necessary

K-eps Turbulence Model, converged solution below 120 iterations

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Fig. 17: Pump characteristics: comparison of integrated experimental and numerical data at the runner outlet and of the global pump head
3.1 Sulzer-ETHZ Research Pump Nq90, revisited using ANSYS-CFX
3.2 A double suction pump analysis using ANSYS-CFX
A double suction pump analysis in ANSYS-CFX
A double suction pump analysis in ANSYS-CFX
4. Three CFD – based pump projects

- SULZER Tehachapi pump
  (performance upgrade by HITACHI Power, 2003-4)

- ANDRITZ (Ritz Pumpen/CTD)

- BITTER GmbH
SULZER – Tehachapi
4 stage pump
CTD: Submersible Multistage Pumps

Hydraulic development project done by CTD was entirely CFD-simulations driven. Design models were systematically developed and evaluated. The process resulted in the design solution which:
- did not require any optimization by testing
- all performance targets were achieved by the design selected for the manufacturing and were confirmed by the first test made, on each of the FOUR different pump sizes.

The performance targets included:
A. $H_{oo}$, $Q_{oo}$ (very strict requirement on exact values of $Q_{oo}$ (and $H_{oo}$ in a certain range))
B. Shapes of $Q$-$H$, and $Q$-$ETA$ characteristics, with ETA values prescribed over a certain range $Q$ range
C. Excellent hydrodynamic stability in tested pump assemblies with up to twelve stages; the tests were confirmed at the end client’s own facilities and these pumps performed better than comparable pumps delivered for this client.
D. Fulfilment of manufacturing related constraints
E. Fulfilment of requirements related to multistage applications.

Note: the pumps operate flawlessly for the 5th year now, in major power stations in Germany
CTD: Submersible Multistage Pumps

Andritz and VDMA images

Q – H – curves of a 3 - stage pump with trimmed impellers

Q – η – curves of a 3 - stage pump with trimmed impellers
Computable pump performance curves
CTD in Automotive

- SUTER Racing (CH-Turbenthal)
- RICARDO (UK)
- Pierburg (D-Neuss)
- BITTER (A)
- ....

Radial Flow (2D) $\rightarrow$ Mixed Flow (3D) $\rightarrow$ $3D^2$
**Impellers – design and manufacturing problem**

<table>
<thead>
<tr>
<th>Impeller style</th>
<th>efficiency</th>
<th>Curvature of the impeller blades</th>
</tr>
</thead>
<tbody>
<tr>
<td>open</td>
<td></td>
<td>„one“-dimensional (1D)</td>
</tr>
<tr>
<td>half-open</td>
<td></td>
<td>two-dimensional (2D)</td>
</tr>
<tr>
<td>closed</td>
<td></td>
<td>three-dimensional (3D)</td>
</tr>
</tbody>
</table>

Manufacturing costs:

- **40**
"1D"  
2D  
3D  

1985  
41  
2005  

> 2016
New generation impeller for automotive cooling

BITTER 3D² impeller
evaluation of numerical simulation
experimental validation
fabrication of prototype
preparation for production

experience, measurements, empirical values
Spezifische Drehzahl

**KFZ/NFZ Haupt-Kühlmittelpumpen**

NFZ: 50 m³/h = 833 l/min

KFZ: 20 m³/h = 333 l/min

* Bitter 3D² Laufräder
  - 3D² (1) hallopen radial 1-st Prototype
  - 3D² (4) hallopen radial 2-nd Prototype
  - 3D² (5) hallopen radial 3-nd Prototype
  - 3D² (9) hallopen radial 4-th Prototype
  - 3D² (11) quasi closed radial 5-th Prototype
  - 3D² (12) quasi closed radial 6-th Prototype
  - 3D² (13) hallopen radial 7-th Prototype
  - 3D² (13a) hallopen radial reworked 7-th Prototype
  - 3D² (16/2) quasi closed radial 11-th Prototype
  - 3D² (16/2a) hallopen radial modified 12-th Prototype
  - 3D² (18) hallopen radial 15-th Prototype (based on rough concept status)
  - 3C² (18a) hallopen radial 16-th Prototype
  - 3D² (19) hallopen radial 17-th Prototype
  - 3D² (20) hallopen radial 18-th Prototype (without thermostats)
  - 3D² (21a) hallopen radial CFD Bitter
  - 3D² (23a) quasi closed basis on 3D² (13a) hallopen radial 21-st Prototype
  - 3D² (23b) quasi closed basis on 3D² (7c) quasi closed radial 22-nd Prototype
  - 3D² (23c) quasi closed radial 23-rd Prototype
  - 3D² (24c) hallopen (thesis V.3) hallopen CFD HTL-Salzburg
  - 3D² (24f) open (thesis V.4.1) axial CFD HTL-Salzburg
  - 3D² (24p) open (thesis V.6) axial 24-th Prototype
  - 3D² (24q) open (thesis V.8) axial 25-th Prototype
  - 3D² (25p) hallopen radial 26-th Prototype
  - 3D² (27) quasi closed radial 28-gmt Prototype
  - 3D² (28) quasi closed radial CFD HTL-Salzburg
  - 3D² (30) hallopen radial 30-th Prototype

**Flügelräder**

$\omega = 4$

bis 45 l/min

**Radialräder**

$n_r = 8$

bis 160 l/min

**Halbaxialräder**

$\omega = 40$

bis 300 l/min

**Axialräder**

$n_r = 100$

bis 160 l/min
BITTER GmbH 3D² Impellers – projects summary; shown in frames are detailed developments incl. optimization by prototyping and testing.
Automotive pump – inlet conditions
Automotive pump – inlet and outlet system configurations (installed conditions)
Zufluss: CV1 – CV3
Laufrad und Ablaufgehäuse: CV3 – CV4
Abfluss: CV4 – CV6


Im Bild links verursacht die ungünstige Zu- und Abströmung, vorgegeben durch den Einbauraum des Kunden, eine Wirkungsgradeinbusse von 11 %.

Häufig ist Optimierungspotential vorhanden, das aber auch Änderungen beim Kunden verursacht.

CV: control volume, Kontrollvolumen bei Strömungssimulationen
Pump & System Modeling for CFD at BITTER

Konstruktion entsprechend Auslegung
Laufrad und Ablaufgehäuse

Strömungssimulationen
Berechnete Drosselkurven, CV1-CV6 (inkl. Zu- und Abströmung)
5. Conclusions

- CFD, thanks to extensive development accompanied by validation effort, proves to be reliable and cost-effective pump development technology.
- CFD is a core performance analysis and predictive technology. It is an integral part of the Design Cycle, human centric, or AI-based.
- It allows to:
  1. **improve** even best-in-class pump design solutions from the past.
  2. **develop** new pumps for increasingly demanding applications and challenging, multi-objective, performance targets.
  3. **innovate**, in which the design solutions combine:
     - optimized blading hydrodynamics,
     - integrated pump and system approach,
     - specialized and cost effective manufacturing technology.