Simulation and Optimization of Flow inside Claw Vacuum Pumps

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Introduction

- Gardner Denver claw vacuum pumps are built for many applications: Chemical Industry, Environmental Engineering (Aeration, Drying, Dust Extraction, etc.), Industrial Applications, Medical Industry, Packaging Industry, Pneumatic Conveying, Post-Press Applications, Printing Press, Wood Working Industry,….

- They have high efficiency, run dry, variable speeds, robust housing, low sound, and so on

- Market supported by **VLR**, **DLR** & a combination, & in size ranges of 60-1000 m^3/h

- Explosion proof versions with reduced leakage are available as well as ATEX compliant vacuum pumps and compressors

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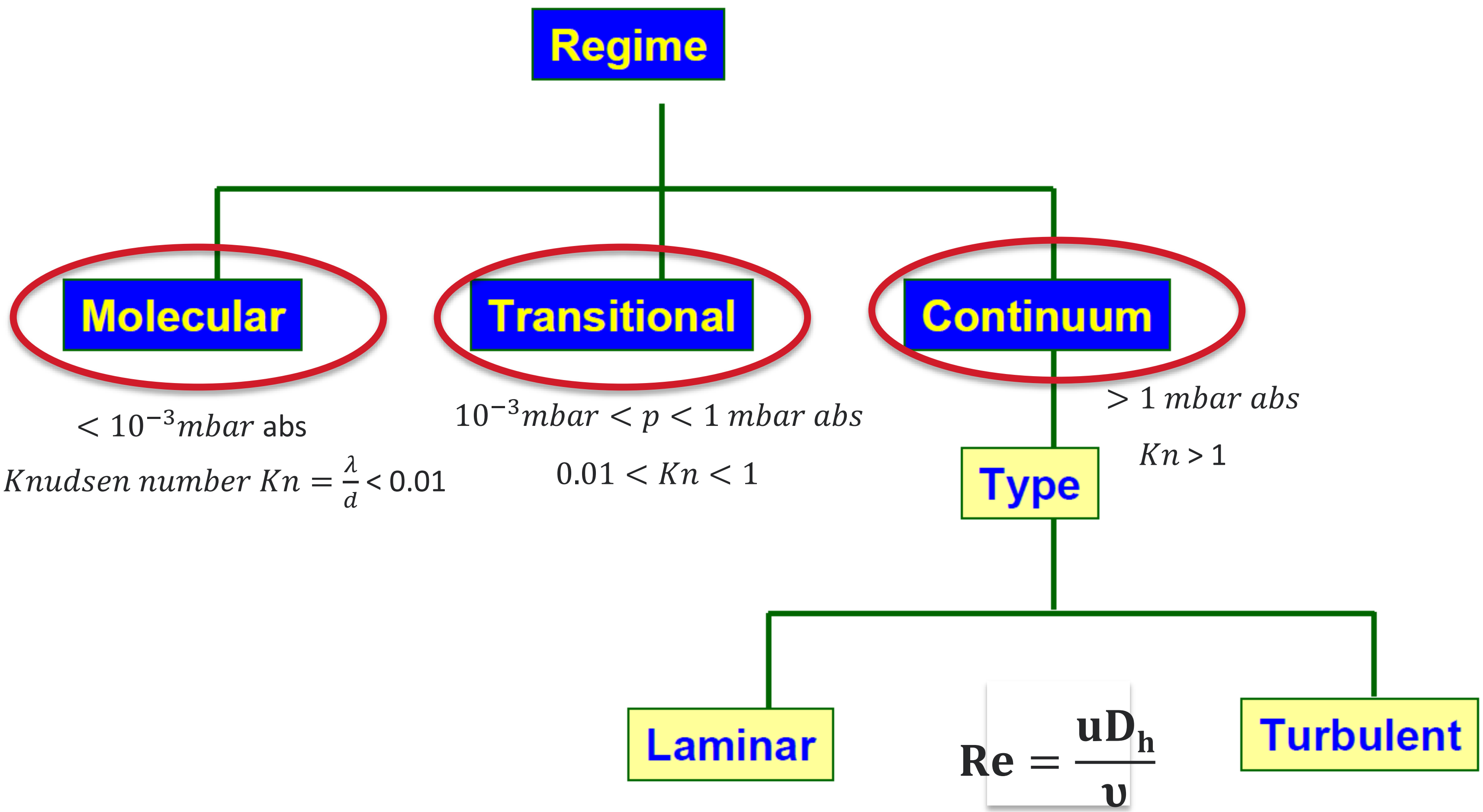
Motivation

Main Goal: Optimization of Claw Vacuum Pumps for lower end vacuum→ **Next Generation Claw**

- CFD is playing an increasing role in the design & manufacture of claw pumps at GD
  - In the past empirical and semi-empirical models were used and design was driven by an iterative process of built and test→ Long development time & cost & risks of accident due to many testing
  - Use of physical models to drive the design process → Simulation Driven Design (SDD) → Reduces the dev. time & lower cost and lower risk of accident due to limited testing in the lab
  - Eliminate trial & error in the design process
- Benchmarking is crucial in the design process:
  - Identify benchmark machines and simulate and validate with measurement data → If necessary, tune model in order to match measurement data
  - Fluid flow, noise & vibration and cooling are crucial since they affect the performance of the machine → Develop models for all crucial processes in the machine
  - End goal is to have a robust mechanical design that is suitable for the intended application

Vacuum Operation

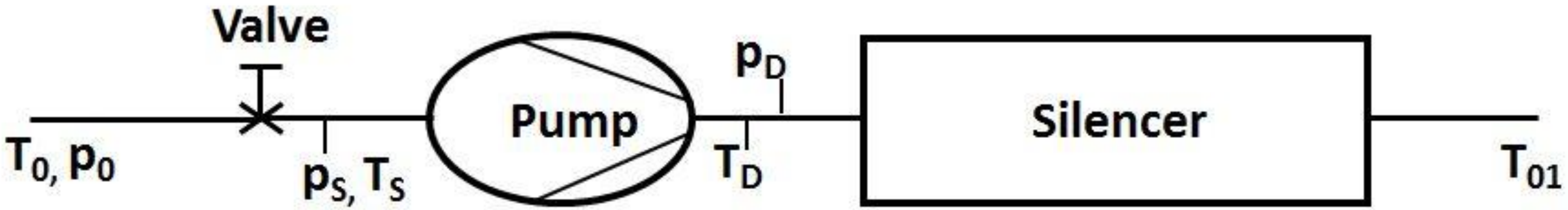
Flow Regimes and Types



Re=Reynolds number, υ is kinematic viscosity in $m^2/s = \mu/\rho$, μ=Dynamic viscosity in Ns/m^2

Measurement Set Up

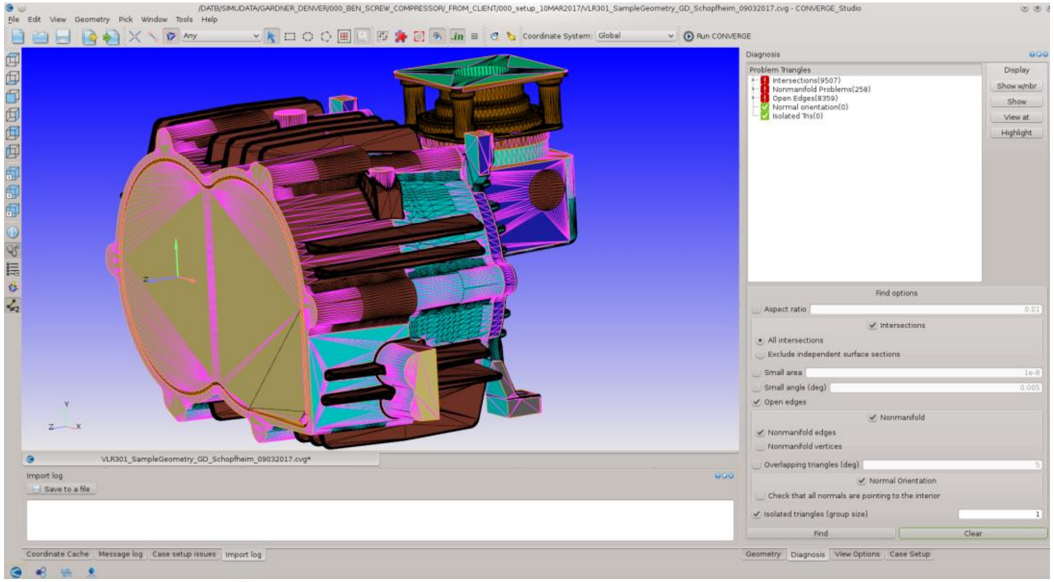
Test stand-Performance measurement



- Sensors are used to measure temperatures and pressures as shown in the sketch above
- Flow meters are used for measuring the volume flow rates
- Input shaft torque is measured to determine the power consumption
-
- Case considered has a rotational speed of 60Hz
- VFDs are used to enable run in a specified speed range as required by the customer

CFD Case Set Up

Pre-processing➔ Geometry cleaning in CONVERGE STUDIO



Fluid Volume Extraction

Define boundary conditions & Materials

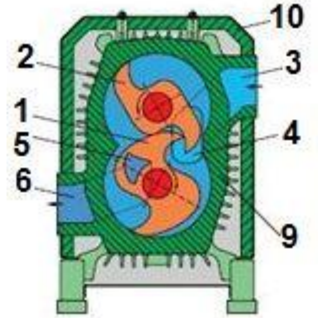
Assign meshing parameters

Solver (8 Cores Linux Server with 256 GB RAM)

Post-processing & validation with measurement data

Design of Experiments (DoE)

Sectional view of a typical claw pump



- For the case presented, heat transferr is not accounted for
- Sealing used to mimic the effect of thermal expansion of the rotors & housing on the axial & radial gaps
- Base grid and level of embedding determined to ensure a mesh independent solution
- AMR used to allow adaptive mesh refinement depending on the temp & vel. gradients in the flow

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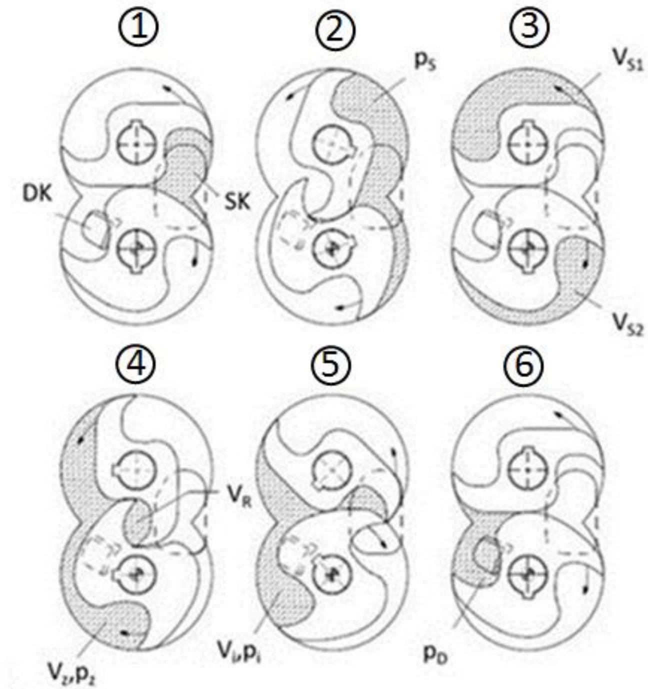
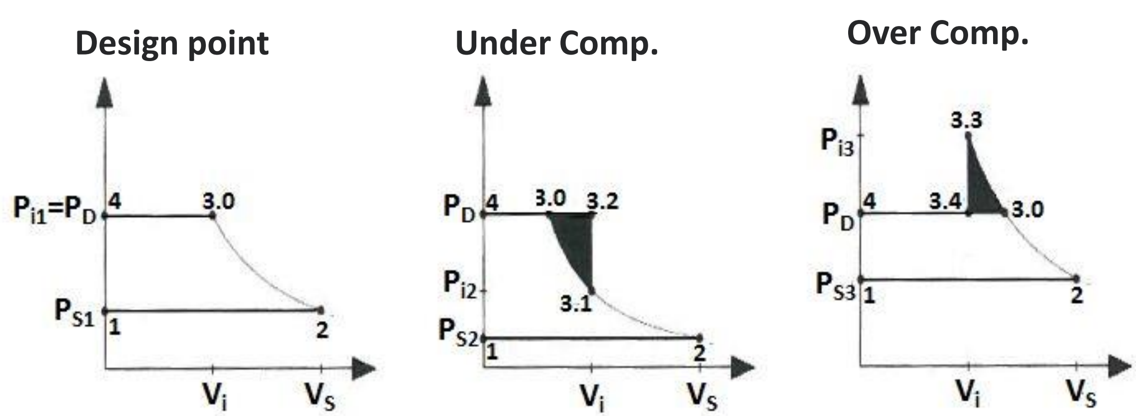
0-D Chamber Model Set Up

- **Steps 1 & 2**: Air is taken into the pump
- **Step 3**: Two closed volumes are transported by the rotors using an isochoric process
- **Step 4**: Internal compression starts & carry-over volume is transported to the suction side
- **Step 5**: Outlet opens
- **Step 6**: Discharge to the atmosphere





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0-D Chamber Model Set Up

Model Equations

$$p_z = p_s \left(\frac{V_{S1} + V_{S2}}{V_Z + V_R}\right)^{\kappa}$$ → **Intermediate pressure**

$$V_{RS} = V_R \left(\frac{p_z}{p_s}\right)^{1/\kappa}$$ → **Carry-over volume**

$$\dot{V}_{th} = z(V_{S1} + V_{S2} - V_{RS})n$$ → **Real pumping speed**

$$\dot{V}_S = z(V_{S1} + V_{S2})n$$ → **Ideal pumping speed**

$$p_i = p_z \left(\frac{V_z}{V_i}\right)^{\kappa} = \left(\frac{(V_{S1} + V_{S2})V_z}{(V_Z + V_R)V_i}\right)^{\kappa}$$ → **Max. internal pressure**

0-D Chamber Model Set Up Cont'd

Model Equations

$$\pi = \frac{p_i}{p_s}$$ → **Compression ratio**

$$\theta = \Pi^{1/\kappa} = \left(\frac{(V_{S1} + V_{S2})V_z}{(V_Z + V_R)V_i}\right)$$ → **Internal volume ratio**

$$P_{is} = \frac{\kappa}{\kappa - 1} p_s \dot{V}_{th} \left[\left(\frac{p_D}{p_s}\right)^{(\kappa-1)/\kappa} - 1\right]$$ → **Pump power**

$$P_{is*} = \frac{\kappa}{\kappa - 1} p_s \dot{V}_{th} \left(\left(\frac{p_i}{p_s}\right)^{(\kappa-1)/\kappa} - 1\right) + (p_D - p_i)\dot{V}_i$$ → **Pump power with under or over compression**

$$P_{is*} = p_s \dot{V}_{th} \left[\frac{\kappa}{\kappa - 1}\left(\frac{1}{\kappa}\theta^{\kappa-1} - 1\right) + \frac{p_D}{p_s}\frac{1}{\theta}\right]$$ → **Pump power with under or over compression**

$$\eta_V = \frac{\dot{V}_{th}}{\dot{V}_S}$$ → **Volumetric efficiency**

$$\eta_{is} = \frac{P_{is}}{P_{shaft}}$$ → **Isentropic efficiency**

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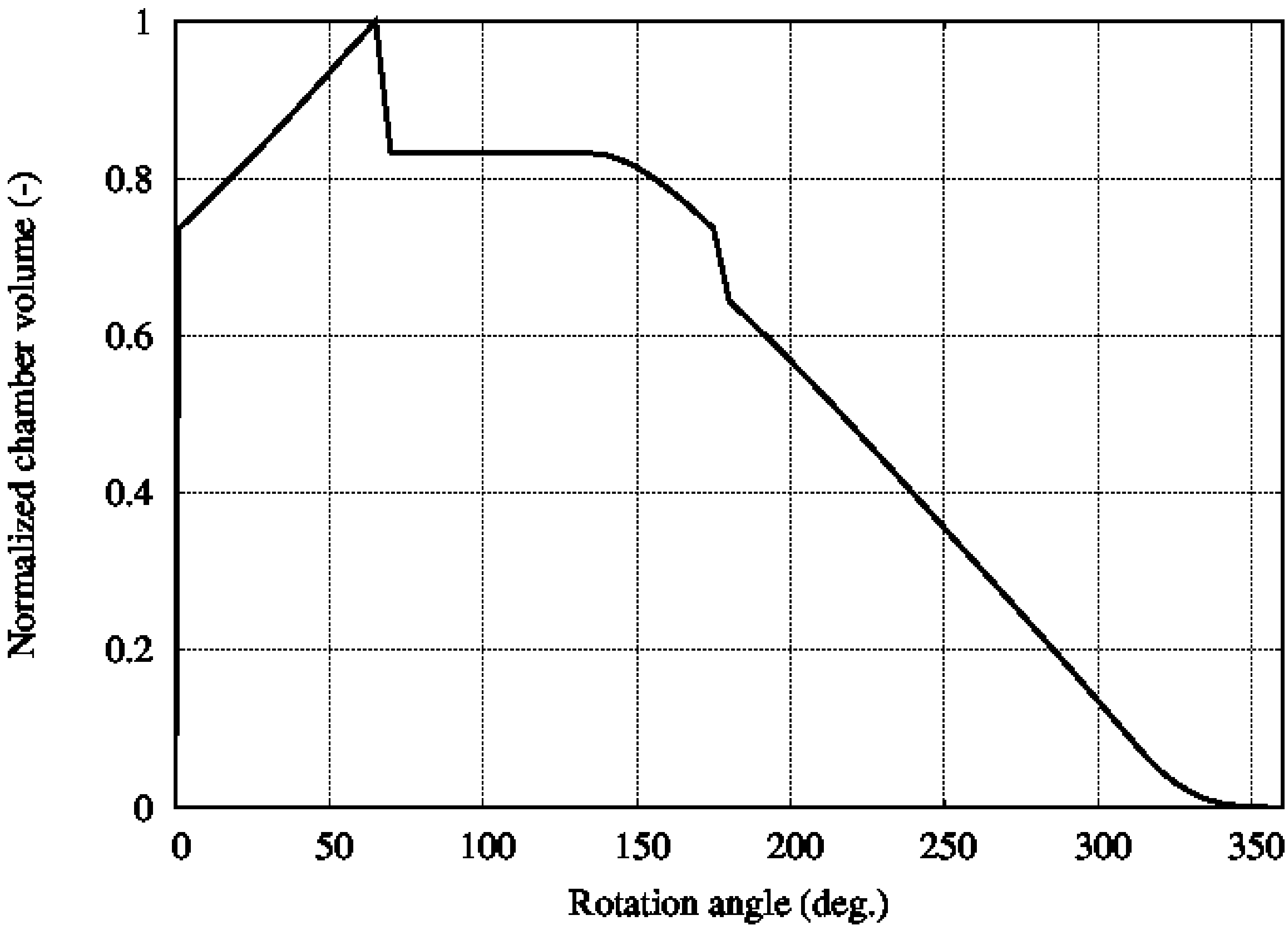
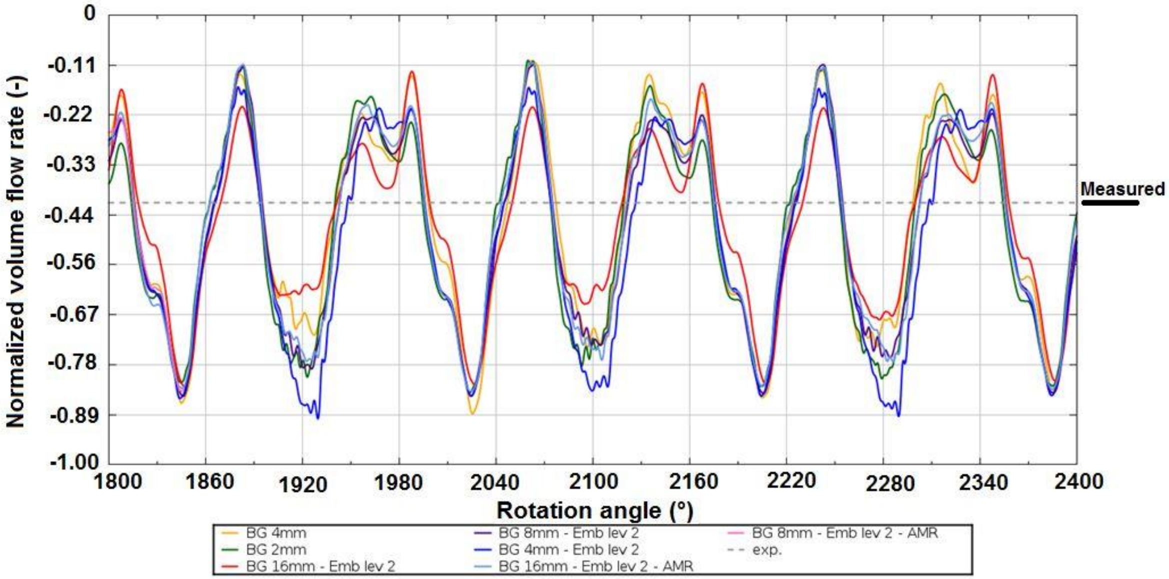
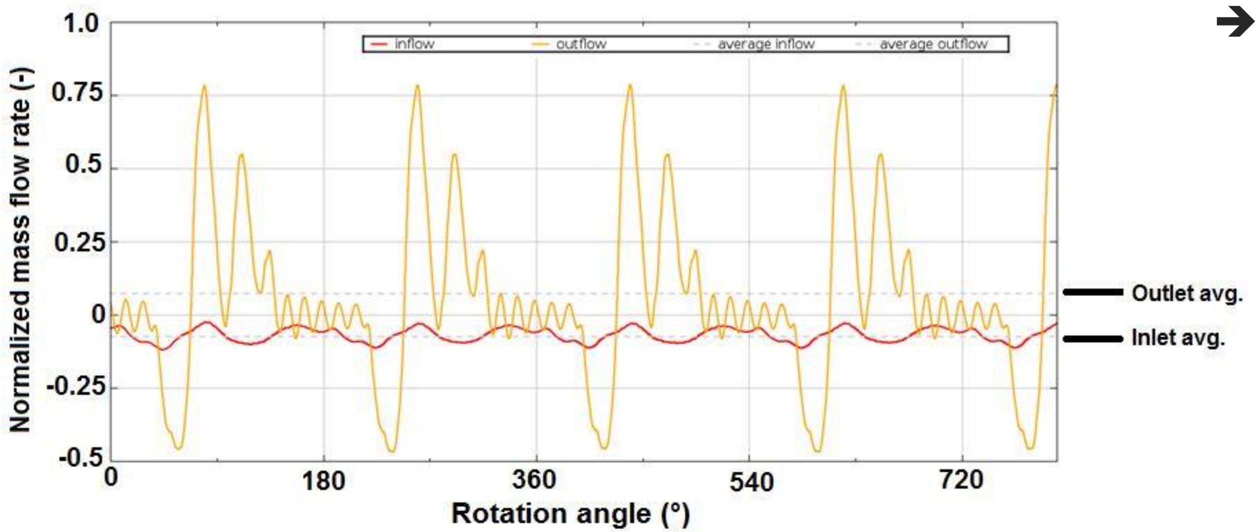
0-D Chamber Model Set Up

Typical Volume Curve of a Claw Pump



Results-CFD Grid Dependency Test





- ➔ Good mass balance achieved @ 150 mbar suction pressure
- ➔ Results obtained is physical due to the grid independent solution
- ➔ Good correlation between measured and predicted pump power
- ➔ Female rotor torque higher than male rotor torque as expected➔ Volume of fluid transported by female larger than male

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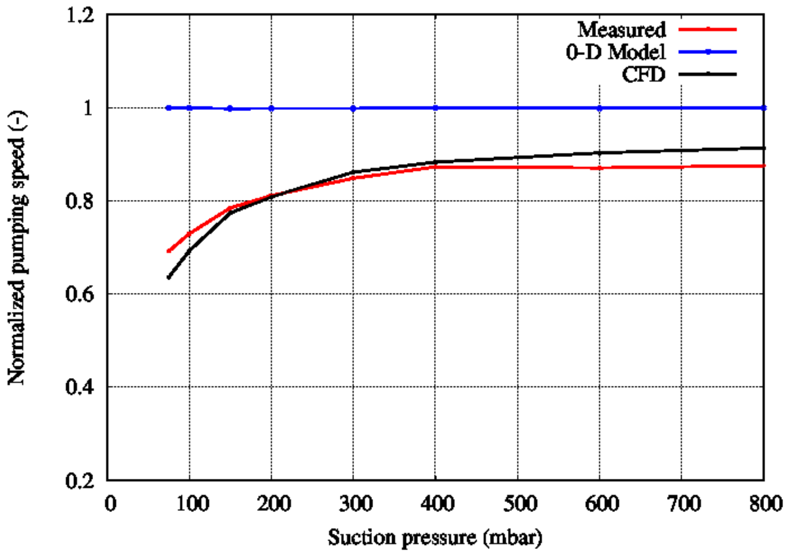
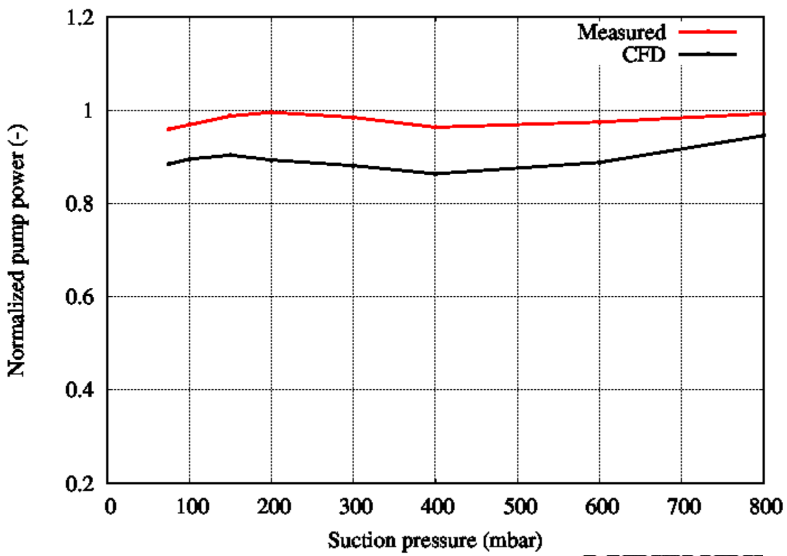
Results-Performance Curves

- Pumping speeds normalized using max. pumping speed from 0-D model
- Pumping speed predicted by the 0-D model is constant as expected ➔ No losses due to gaps inside the machine
- Trend in the pumping speed measured and simulated using CFD are similar up to 300 mbar suction pressure
  - ➔ At low suction pressures flow losses are higher and gradients are larger
- Pump power normalized using max pump power from measurement
- CFD model not accounting for power consume in bearings and gears
- Trend is similar between CFD and measurement
- 0-D pump power increases linearly with decrease in pumping speed

Pumping Speed @ 60 Hz



Pump power @ 60 Hz



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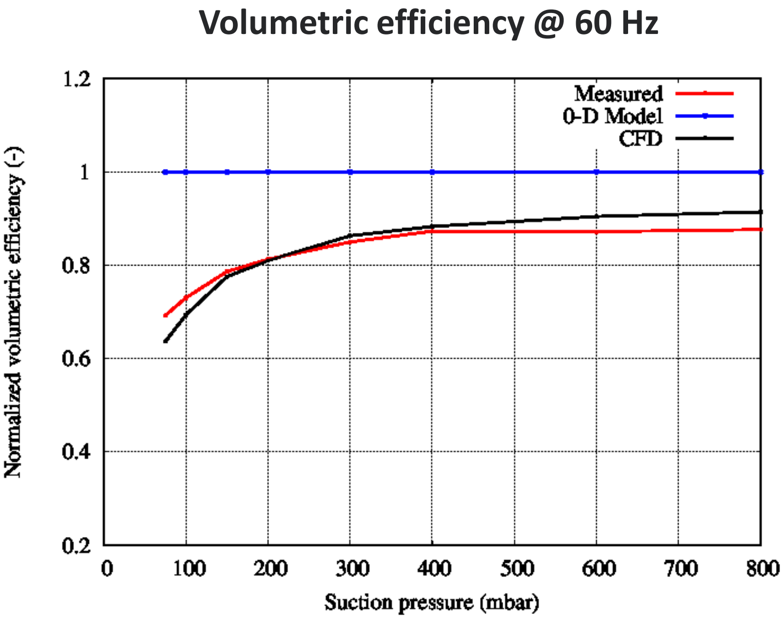
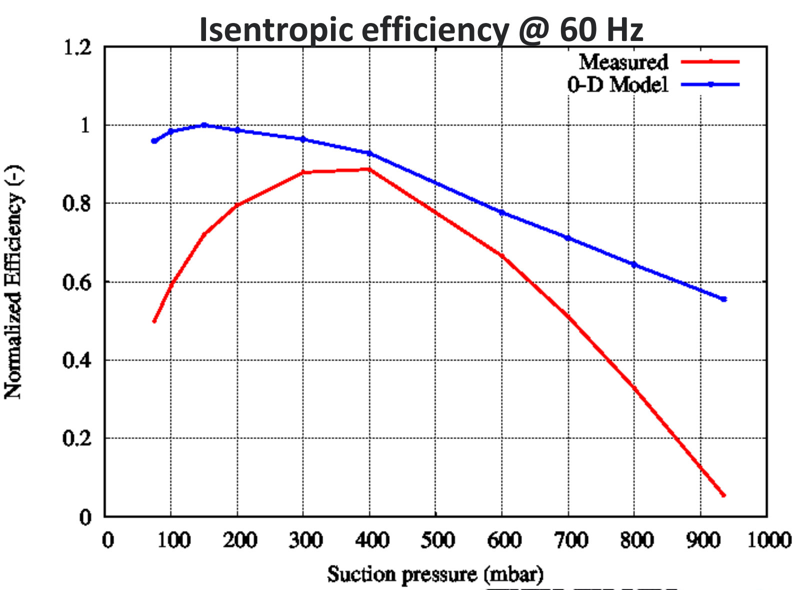
Results-Performance Curves Cont’d

- Volumetric efficiencies normalized using max. from 0-D model
- Trend is similar to the pumping speed curve
- In 0-D model losses occurring due to gaps not accounted for but accounted for in measurement and CFD
- Isentropic efficiency shows similar trend➔ Lower on the high pressure side due to high over compression
- It is max at the design point where there is no over or under compression





Conclusions

- The use of 0-D chamber models successfully integrated in the design process of claw pumps at GD
- CFD is also successfully integrated into the design process including the use of Design of Experiments to optimize the design
- The added benefits of a reduction in the development time and product optimization being realized
- CHT models are being used to determine thermal loads on the rotors and sealing and bearings and helping in the optimization of the cooling and the determination of the hot gaps
- Optimization work like the reduction in the carry-over volume, over & under compression prevention, optimized cooling, etc., are possible using CFD/0-D thermodynamic models

Future Work

- Design improvement/Optimization using Design of Experiments (DoE)
- CHT + FSI simulation to enable us to predict the thermal expansion of the rotors
- Item 2 is critical for determining the hot gaps inside the machine
- Extension of the 0-D chamber model to include gap flow and losses occurring through the gaps

Acknowledgment

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