RUHR-UNIVERSITÄT BOCHUM

ACS Automatic Control and Systems Theory

RUB

Assessing rheological properties of fluids through torque dynamics in progressive cavity pumps

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Motivation & Goal

- **Understanding** the physical influence of fluid properties on the torque of a progressive cavity pump
- **Modeling** of the progressive cavity pump with viscosity, total suspended solids and wear as explicit parameters
- **Laying the foundation** for using this model as a basis of a Softsensor

State of the art techniques

- Classical measurement of total suspended solids is ex-situ and slow, as water needs to be evaporated from a probe for each test
- Measuring the total suspended solids or viscosity in-situ is expensive due to highly complicated measurement devices
- Modeling of progessive cavity pumps for flow or torque monitoring [1,2,3] already established

Methods

- Data analysis for feature identification
- Lumped parameter modeling of the dynamics of friction torque of progressive cavity pumps
- Identifying components that reveal
  - fluid properties
  - wear
- Parameter identification from lean datasets

Modeling the torque of progressive cavity pumps

Working principle of elastomer progressive cavity pumps

Progressive cavity pumps move fluid by

- rotating a helix-shaped rotor inside a stator slothole and
- generating sealed chambers that move axially towards the pressure side through the eccentric rotation of the rotor relative to the stator.

Progressive cavity pumps usually work on **low rotation speed** and provide **low shear rates** inside the fluid to be able to **handle complex fluids**. Their **tolerance of abrasive and highly viscous fluids is high** due to the working principle.

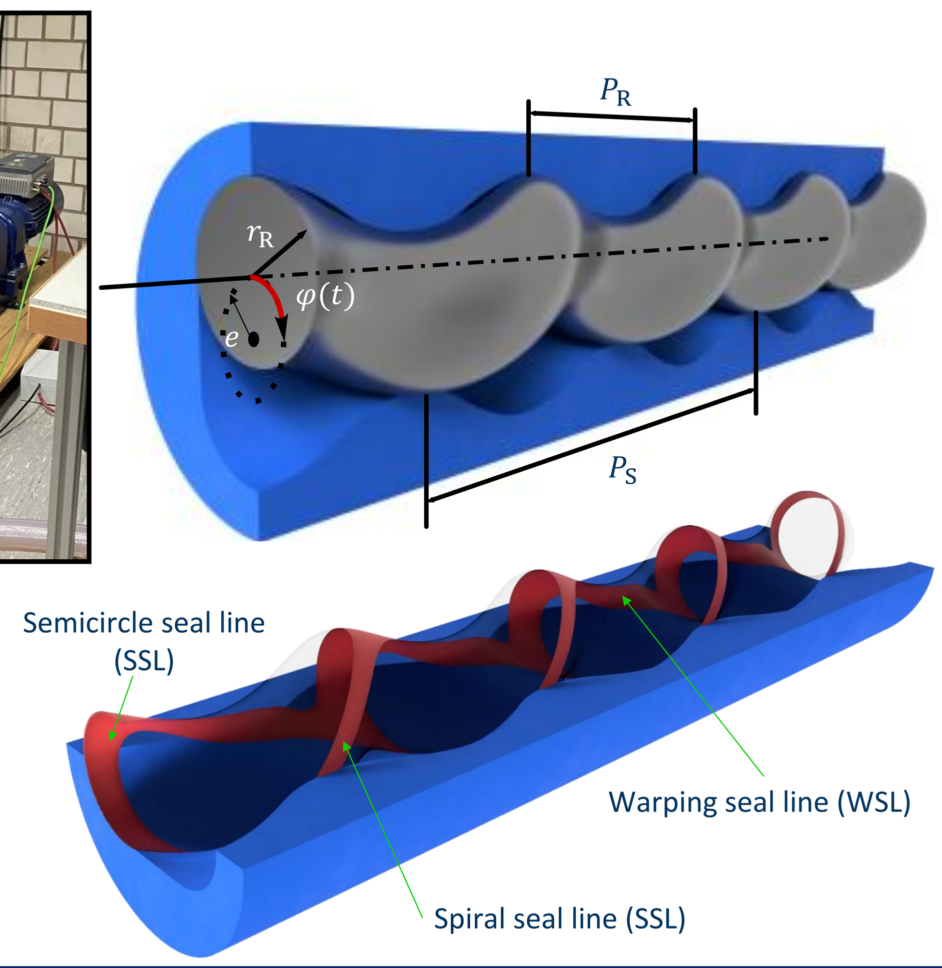

Fig. 1: Kinematics of a 2-stage single lobe PCP with deviation from reference path and the visualisation of the sealing lines

On the origin of torque dynamics

- Balance of torque around center axis reads
$$J\ddot{\varphi} = T_{\mathrm{el}} - T_{\mathrm{hyd}} - T_{\mathrm{fric}}$$
with the moment of inertia J, rotative acceleration $\ddot{\varphi}$ and the electrical torque T_{el} generated by the driving motor
- **Hydraulic torque** is goverened by the differential pressure $\Delta\overline{p}$:
$$T_{\mathrm{hyd}} = 2 d_{\mathrm{R}} e P_{\mathrm{S}} \Delta\overline{p} \pi^{-1}.$$
- **Friction torque** is mainly caused by friction between rotor and stator:
$$T_{\mathrm{fric}} = \kappa_{\Delta\overline{p}} r_{\mathrm{r}} \mu F_N$$
with reduction factor $\kappa_{\Delta\overline{p}}$ that accounts for normal force F_N reduction caused by differential pressure across the pump
- Normal force F_N varies with rotation angle φ, the deviation from reference path, differential pressure $\Delta\overline{p}$, and wear
$$F_N = f(\varphi, \tau, \dot{\tau}, x, \dot{x}, \Delta\overline{p}, w).$$
- Friction coefficient μ varies with rotation speed $\dot{\varphi}$ and viscosity η
  - ➢ **Finding appropriate descriptions for μ and f completes the model**

Torque dynamics as function of fluid-rheology and normal force

Modeling of friction factor μ

- Friction coefficient is a function of viscosity or total suspended solids, respectively, and of rotation speed
$$\hat{\mu}(\dot{\varphi}, \eta) = \gamma_1(\tanh(\gamma_2\dot{\varphi}) - \tanh(\gamma_3\dot{\varphi})) + \gamma_4 \tanh(\gamma_5\dot{\varphi}) + \gamma_6\, \dot{\varphi}^n + \gamma_7(\tanh(\gamma_8\eta) - \tanh(\gamma_9\eta)) + \gamma_{10}\tanh(\gamma_{11}\eta) + \gamma_6\, \eta^m$$
  - ➢ Can be parameterized with data from a few operation points.
- Parameterization is performed by solving
$$\min_{\gamma_i, n, m} ||\mu_{\mathrm{meas}}(\dot{\varphi}, \eta) - \hat{\mu}(\dot{\varphi}, \eta)||_2^2$$

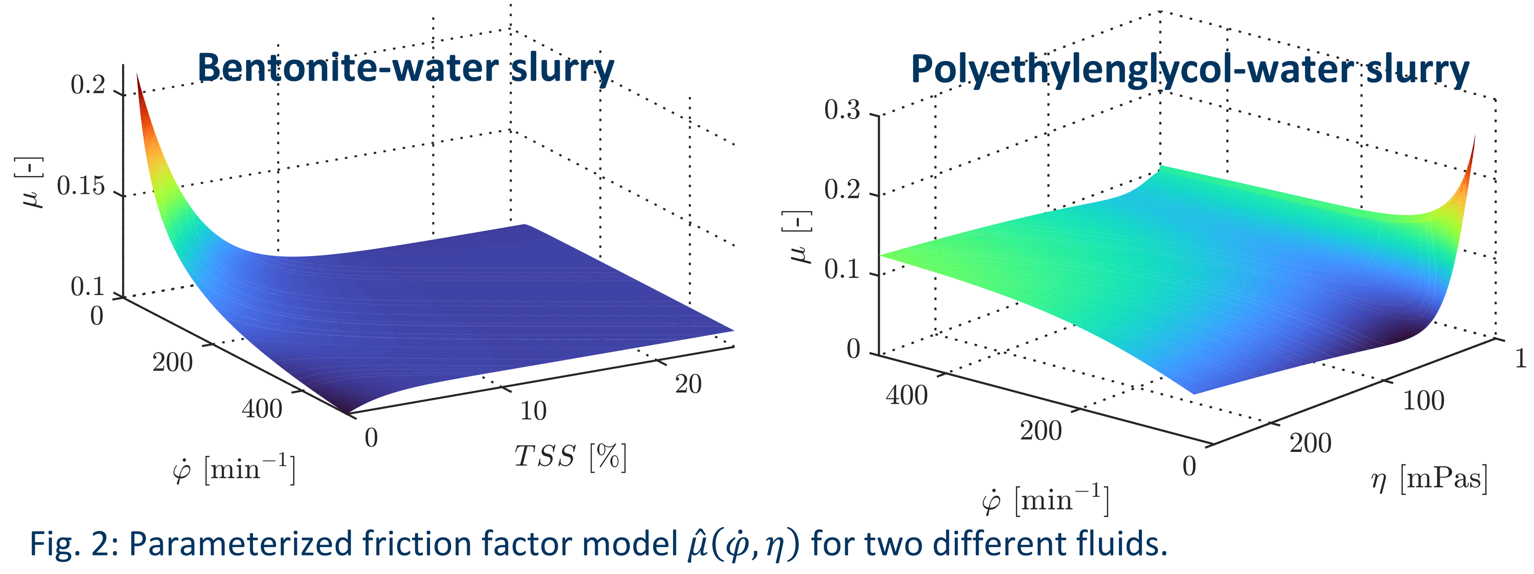

Fig. 2: Parameterized friction factor model $\hat{\mu}(\dot{\varphi}, \eta)$ for two different fluids.

Modeling of normal force F_N

- Modeling of the torque caused by the SSL sealings as moving springs which simultaneously act as rotor bearing and cause normal forces
- Lumping dynamics into the center of gravity
$$\vec{x} = [x, \dot{x}, \tau, \dot{\tau}]$$
- Position s_i of the springs and their stiffness c_i are modeled as functions of rotation angle φ with $s_i = -\frac{l}{2} + \frac{(i-1)P_S}{2\pi} \bmod (\varphi, \pi)$ and $c_i = \frac{E^* \pi l_{\mathrm{seal},i}}{4}$ with the bulk modulus E^* and the instantaneous length of the sealing $l_{\mathrm{seal},i}$
- Forces acting on the rotor are caused by differential pressure, uncertainty in rotation angle position and spring-damper forces.
- Using actual position of the rotor to calculate
$$\boldsymbol{F_N} = \sum \boldsymbol{c_i(\tau s_i + x_i + (w - w_0))}$$

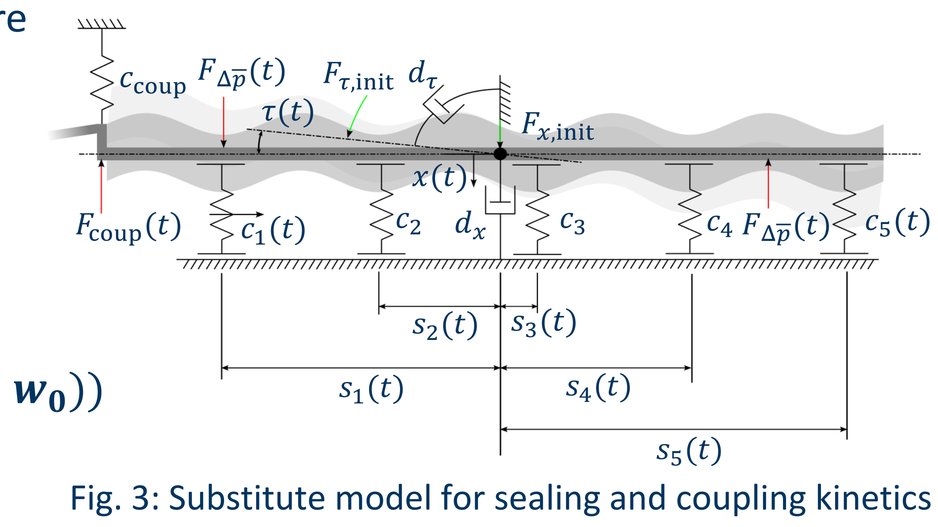

Fig. 3: Substitute model for sealing and coupling kinetics

Validation of the Model for multiple operating points and fluid combinations

- Parameterization of friction torque model at different standard operating points $\mathcal{O}(\dot{\varphi}, \Delta\overline{p})$ with water
- Simulation is performed in MATLAB
- Validation of the model against different fluids (water-bentonite and water-PEG), and wear states
  - ➢ **Model shows sufficient accuracy for further usage in control and softsensor design**

Results for standard rotor

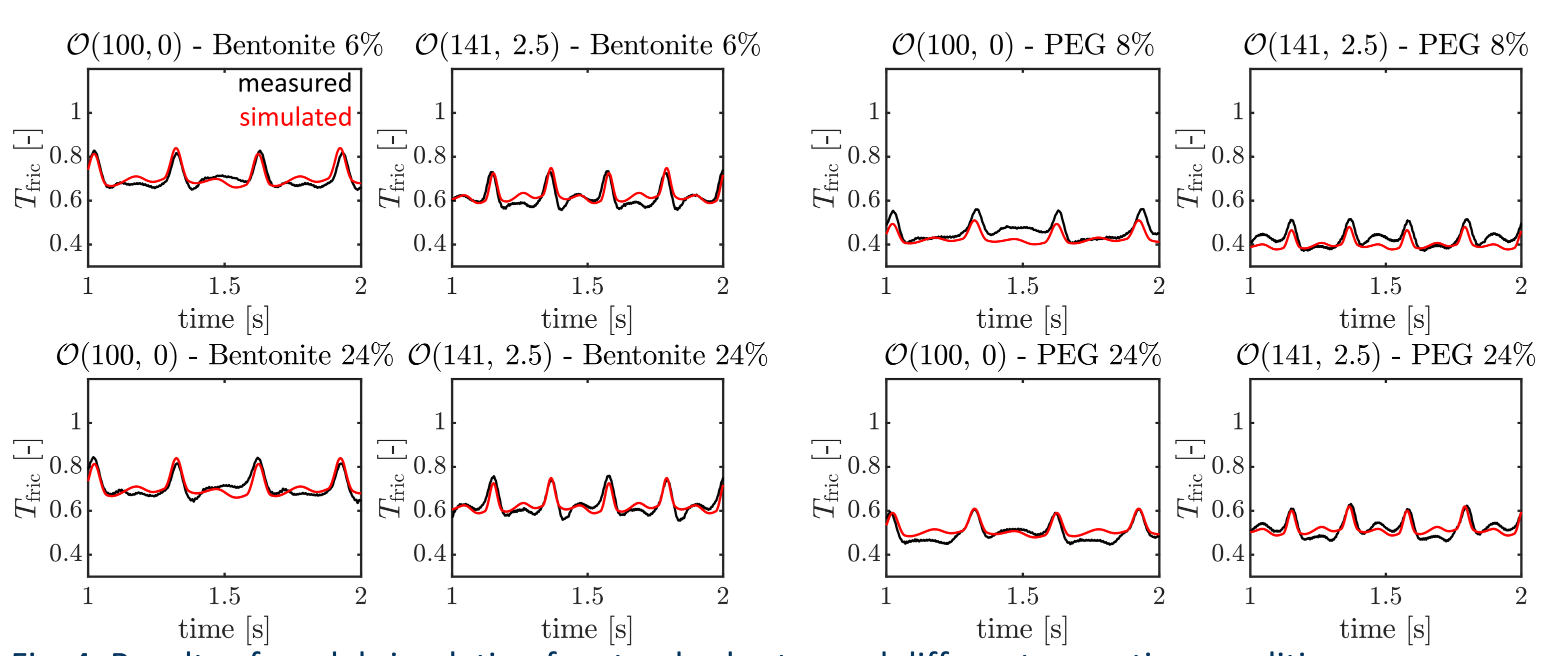

Fig. 4: Results of model simulation for standard rotor and different operation conditions

Results for worn rotor

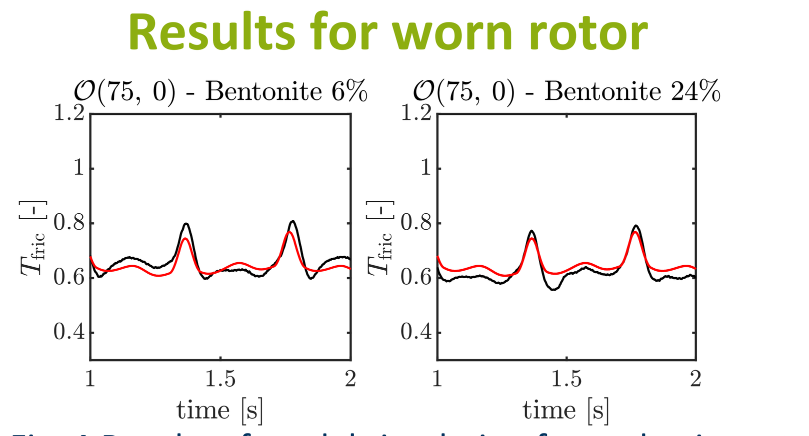

Fig. 4:Results of model simulation for undersize rotor

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