SIGNIFICANCE

Multiple stage centrifugal pumps comprise of several impellers mounted on a single shaft spinning inside a barrel. These pumps rely on a number of sealing elements that restrict secondary flow. Seals boost pump efficiency and reliability: impeller neck ring (eye) and inter stage annular (non-contact) seals restrict flow from a high pressure discharge region towards an intake section. In straight-through pump configurations, a long seal acts as a balance piston that equilibrates the full pressure rise across the pump. Pump rotors are long and flexible, the operating dense fluid dictates the system natural frequencies and damping ratios. Proper support bearings’ stiffness and damping characteristics ensure adequate placement of a pump’s critical speed, a wide margin away from the operating speed range or with enough damping (log-dec) to ensure safe passage through a critical speed. More crucial; however, is the appropriate design and installation of the balance piston seal as its stiffness, damping and virtual mass coefficients (lateral and tilting) can shift the pump fundamental critical speed and bring either more or less stability to the system. Instances of pump failures due to rotordynamic instability induced by a sealing device are well documented [1].

Unlike in centrifugal compressors where various types of damper seals (textured surfaces) and interlocking labyrinth seals with inlet swirl brakes and/or shunt injection holes are common, balance pistons in pumps retain a basic configuration with deep square grooves breaking the land portions with small clearance (c), see inset on the right. In pumps, rubbing with the rotor surface and wear with clearance enlargement are common happenstances. A smooth land (ungrooved) seal is not a safe choice, even more so as the rapid development of the fluid circumferential velocity in a long seal (L/D~2) will quickly produce too large cross-coupled stiffness and promote rotordynamic instability. Hence, the selection of a grooved pattern that breaks the development of fluid swirl. Compared to smooth annular seals, a grooved seal has less leakage and cross-coupled stiffnesses of lesser magnitudes. Too shallow grooves or too deep grooves do not reduce direct stiffness and damping appreciably. On the other hand, balance pistons holding the full pressure rise across the pump can have a significant direct stiffness due to the Lomakin effect at their inlet, and thus affect the placement of critical speeds [2].

The through flow in pump seals is typically turbulent due to the large pressure drop (ΔP), surface rotational speed (½ΩD), fluid properties (small kinematic viscosity, μ/ρ), i.e., the Reynolds number is rather large, Re>>1. Hence, prediction of the flow, drag power and rotordynamic force coefficients in (rectangular) grooved seals has relied on application of Hirs’ bulk-flow model (BFM) [3], and more recently, on computational fluid dynamics (CFD) solutions. Nordmann et al. [4] and Marquette and Childs [5] present BFM implementations extensively used in industry.

Hirs’ model is simple as it makes no distinction on the type of flow (pressure or shear driven) and relies on empirically based friction factor(s) f=n Re^n. The BFM does well in flows without strong recirculation or curved streamlines, but not in a rectangular deep groove seal which has local recirculation or vortices in each cavity [3]. Friction factors in seals with a complex geometry, such as in labyrinth seals, honeycomb damper seals and deep grooves, are more complicated functions of the Reynolds number, geometry and surface conditions than the simple formulas advanced by Hirs [3].

XLTRC© program XLCGrv© implements a three control volume bulk-flow model [5] to predict the leakage and force coefficients of circumferentially grooved annular liquid seals. XLCGrv© model predictions have been validated to an acceptable accuracy against tests for a short length grooved seal (L/D <0.45) [6,7]. XLCGrv©
predictions for a long seal \((L/D > 1)\) with shallow grooves are regarded as inadequate to reproduce field behavior in a boiler feed pump [8].

CFD solutions to the complex flow in grooved seals are (becoming) common engineering practice as commercial software is readily accessible and computers’ processing speed continuously increases (Moore’s law). However, prediction of seal force coefficients using commercial CFD software still demands of a large intellectual effort and computing time since the unsteady fluid flow process due to small amplitude displacements of the rotor (perturbation analysis) requires of attaining (converged) multiple flow field solutions as time evolves and the clever matching of meshing techniques in a rotating coordinate system [9,10].

Untariou et al. [11,12], following precursor ideas by Arghir and Frene [13] and Villasmil et al. [14], implement a hybrid scheme CFD-BFM to speed up the prediction of reliable force coefficients in grooved seals. In brief, the CFD software predicts the equilibrium flow field for sets of operating conditions and the walls’ friction factors \((f)\) are extracted. Difference friction factors, as numerical functions of changes in pressure \((\Delta f/\Delta P)\), circumferential speed \((\Delta f/\Delta U)\), and clearance \((\Delta f/\Delta c)\), are inserted in the BFM program to predict rotordynamic force coefficients. Untariou et al. [12] claim great accuracy in predictions when comparing to test data (damping ratio) for a commercial pump (Flowserve), with significant savings in computational time.

**PRELIMINARY RESULTS OF A CFD STUDY FOR A LIQUID CIRCUMFERENTIALLY GROOVED SEAL**

A preliminary CFD study for a water lubricated balance piston shows good agreement in predicted leakage versus test results obtained by a TRC member [8]. The seal, with a whole length and diameter equaling 180 mm, has 55 grooves. The groove and land are 1.6 mm in length, a groove depth equals 0.25mm, and the land radial clearance is 0.225mm. The fluid has density \(\rho=935 \text{ kg/m}^3\) and viscosity \(\mu=0.2 \text{ cPoise}\) at 130°C. The balance piston seals 121.5 bar and the pump speed is 2,980 RPM.

Figure 1 illustrates the CFD predicted axial pressure on the seal stator and views of the axial velocity for grooves of decreasing depth, from 1.6 mm to 0.1 mm. Clearly, the shallow groove configurations do not show a (strong) recirculation zone within a cavity. The predictions evidence the need to reformulate the conventional BFM for shallow depth grooved seals.

**PROPOSED WORK 2015-2016**

In year 2015-2016, the objective of the project is to introduce a hybrid method to update the XLGrv® or XLAnSeal® programs with friction factors derived from a number of CFD simulations conducted for well-known balance piston (grooved seal) configurations. In this manner, the updated software will deliver accurate force coefficients for ready implementation in reliable pump rotordynamic analyses. The list of tasks includes:
1. Build mesh-model for grooved seal with deep and shallow depth grooves in CFD software (ANSYS Fluent®). Find flow fields for nominal configuration and with changes in inlet pressure, surface speed and clearance. Calculate friction factors ($f$) along seal length and circumference and its differences, ($\Delta f/\Delta P$), ($\Delta f/\Delta U$) and ($\Delta f/\Delta c$).

2. Implement found (numerical) friction factors and its difference in a BFM program to predict seal leakage, power loss, and rotordynamic force coefficients.

4. Compare predictions of BFM against CFD results (leakage mainly) as well as published test data for force coefficients.

**BUDGET FROM TRC FOR 2015-2016**

<table>
<thead>
<tr>
<th>Year I</th>
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<tr>
<td>Support for graduate student (20 h/week) x $2,200 x 12 months</td>
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<tr>
<td>Fringe benefits (2.7%) and medical insurance ($150/month)</td>
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<td>Tuition three semesters ($363 credit hour x 24 ch/year)</td>
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<td>PC upgrade + monitor</td>
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<td>Attend 2 day short-course in CFD analysis</td>
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<tr>
<td><strong>Total Cost:</strong></td>
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**REFERENCES**


