Numerical Simulations of Unsteady Flow Instabilities (Rotating Stall) in Pumps

by

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The stable behavior of pumping systems is in general a basic condition for a reasonable operation and high operating hours of pumps. Instabilities at part load can lead to reduction of efficiency, high dynamic load and cavitation up to the destruction of the pump. The operating range is mostly limited by the existence of instabilities. One of the well known instabilities of technical importance is rotating stall.

The existence of the phenomenon has been long acknowledged and numerous have investigated its various forms. Much of their efforts have concentrated on understanding the conditions which lead to the formation and propagation of stall cells. They have also tried to derive procedures to predict when or where stall will arise. Further efforts have been developing design methodologies to avoid the problem and to evaluate various techniques to eliminate or minimize the effects of stall which could arise in a compressor. Despite the substantial amount of research completed, no universally accepted methods for stall prediction or avoidance have been derived.

Rotating stall can occur in both axial and centrifugal compressors. However most of the research has been done on axial compressors. Rotating stall (RS) is an instability where the circumferential flow pattern is disturbed. This is manifested through one or more stall cells of reduced, or stalled, flow propagation around the compressor rotation at a fraction (20-70%) of the rotor speed. This leads to a reduction of the pressure rise of the compressor, and in the compressor map this corresponds to the compressor operating on the so called in-stall characteristic, see Figure 1.

Figure 1: Physical mechanism for inception of rotating stall

The basic explanation of the rotating stall mechanism can be summarized as follows. Consider a row of axial compressor blades operating at a high angle of attack, as shown in Figure 1. Suppose that there is a non-uniformity in the inlet flow such that a
locally higher angle of attack is produced on blade B which is enough to stall it. The flow now separates from the suction surface of the blade, producing a flow blockage between blades B and C. This blockage causes a diversion of the inlet flow away from B towards A and C, resulting in an increased angle of attack on blade C, causing it to stall. Thus the stall cell propagate along the blade row.

As already mentioned most of the research has been done on axial compressors. However there are theories about rotating stall in centrifugal compressors. It is suggested that impeller rotating stall could result from flow perturbations at the impeller exit that would not allow the flow to follow the blading. These perturbations may be a consequence of disturbances within the impeller passages or strong interactions between the impeller and the diffuser. Researchers have identifies the probable causes for rotating stall as incidence angles at the impeller leading edge or pressure disturbances caused by the impeller blade geometry. Based on the published works, impeller rotating stall can manifest itself at frequencies from 26 % of running speed up to 100 % of running speed.

Newly published works show a growing interest in this area. Main focus is in new measurement techniques. With the use of Particle Image Velocimetry (PIV) it has become possible to determine the velocity flow. Figure 2 shows cells with Rotating Stall and Sound Flow.

![Figure 2: PIV-Results, Rotating Stall (left) Sound Flow (right)](image)

**Simulations at IHS**

First simulations with an axial compressor have been carried out. Experiments have been done by ETH Zürich and compared with the simulations. The axial compressor consists of an impeller with 30 blades. The Diameter at the hub is 299 mm and at the tip 380 mm with a ratio of 0.82. The rotational speed is 120 r/min. The design value corresponds to a flow coefficient $\phi = 0.45$.

The unequal distribution of the velocity and the pressure field in the different passages disables the assumption of periodicity, so that the numerical model consists of the entire impeller with 30 blades. Figure 3 shows the impeller and diffuser. Inflow and outflow region are elongated to avoid influences of the inlet and outlet boundary condition to the structure of the rotating stall cell. The hub is also enlarged to the inlet and outlet. The entire model is divided into 60 parts, two for each passage, and simulated parallel. In total about 600000 elements are used to discretize the simulation domain.
The flow field is simulated for two operating points, \( Q/Q_0 = 0.63 \) and \( Q/Q_0 = 0.38 \). For these operating points experimental results exist, thus comparison is possible. The pressure in the interspace between impeller and diffuser is plotted in Figure 4. In the interspace a pressure peak and a pressure valley are found, which bound the Rotating Stall cell. For the chosen operation condition, the computation shows one single stall cell of the full span type. This agrees well with the experiment.

![Figure 3: Impeller and diffuser with 30 blades (left), Impeller Rotating Stall (right)](image)

Experimental data is also shown in Figure 4. The comparison shows good agreement between simulation and experiment. The blockage factor (\( \lambda = \) blocked/entire region) due to the pressure maximum and minimum is predicted to \( \lambda = 0.45 \). The experimentally found value varies between \( \lambda = 0.34 \) and 0.4. The rotational speed of the RS-cell is simulated to \( \Omega_{RS}/\Omega_R = 0.51 \). This agrees well with the experimental value \( \Omega_{RS} = 0.54 \). The difference between pressure peak and pressure valley is simulated to \( \Delta P = 1650 \text{ Pa} \) slightly lower compared to the experimental value of \( \Delta P = 1830 \text{ Pa} \).

![Figure 4: Pressure in the interspace near casing (left), experimentally determined pressure near casing (right)](image)

![Figure 5: Pressure distribution (left), Transport velocity in front of the impeller (right)](image)
To get an insight into the flow structure the simulated results will be discussed based on two unrolled cylinder cuts. The cuts are located 10% of the blade height away from the hub and from the casing. The pressure distribution is shown in Figure 5. The unequal distribution around the circumference can be detected. In front of the impeller a region with low pressure is built, which indicates the existence of one full span RS-cell. Behind the diffuser higher pressure can be found.

Significant differences are found for the flow pattern near hub and near casing. While throughflow with high velocity in the RS-cell is determined near hub, the region near casing is characterized by considerable backflow. This leads to strong three-dimensional flow pattern in the rotor. In the diffuser passages throughflow without backflow is predicted in both cases. The region with sound flow is characterized by throughflow in impeller and diffuser passages and shows low differences between hub and casing. The transport velocity in Figure 5 shows a significant split between sound flow and RS cell. For the sound flow $v_z$ shows nearly identical inflow for all three profiles. In the RS-cell the velocity profiles are separating. While near hub a velocity rise can be recognized a marked backflow is predicted near casing with an average velocity of $v_z=-0.4 \text{ m/s}$ with significant high frequented wakes near the leading edge. The transition between sound flow and blocked region is at both frontiers characterized by a steep velocity gradient.

After investigating RS in axial compressors simulations in radial pumps have been carried out. The actual measured runner consists of 6 blades. The Inlet diameter is 71 mm, the outlet diameter is 190 mm. The inlet height is 13.8 mm, the outlet height 5.8 mm. At the inlet the angle is 19.7° and at the outlet 18.4°. The pump has a discharge of $Q_n = 11 \text{ m}^3/\text{h}$ and a head of $H_n = 1.76 \text{ m}$. The rotational speed is 725 r/min. Figure 6 shows the velocity field at a nominal discharge of $Q/Q_n = 100\%$.

![Figure 6: Velocity field for 6 blade runner at $Q/Q_n = 100\%$ (left), Velocity field at $Q/Q_n = 25\%$ (right)](image)

In order to get rotating stall the discharge is decreased. Figure 6 plots the velocity field at a discharge of $Q/Q_n = 25\%$. Here rotating stall can be seen. The runner now consists of three sound regions and three blocked regions. They rotate stable with the runner speed. The blocked regions reach from hub to shroud. To get a more detailed view of RS the runner is modified to 7 blades. In comparison to Figure 6, the velocity field for the new runner for the two operating points are shown in Figure 7. Here at least three blocked and three to four sound regions exists. This means, that the rotating stall cannot stay stable in one channel, but it has to rotate. The rotational
speed is approximately 50% of the rotational speed of the runner. As the measurement only provides results for the 6 blade runner, this cannot be compared with measurement results. As seen in Figure 2 (PIV measurements for another impeller) the creation of the rotating stall can be seen. At the impeller exit a swirl is built. It seems to start closing the channel by moving forward to the leading edge. This leads to the foundation of another swirl built by separation on the leading edge suction side of the blade. These two, sometimes even three swirls totally block the channel. A change in incidence brakes the blockade.

Figure 7: Velocity field for 6 blade runner at Q/Q_N = 100% (left), Velocity field at Q/Q_N = 25% (right)

Figure 8 underlines the mentioned statements. As for 6 blades the blocked cells remain stable, for 7 blades the blocked cells rotate around the channels.

Figure 8: Discharge for each channel for Q/Q_N = 25%, 6 blades (left), 7 blades (right)

**Conclusions and further work**

Rotating stall in axial compressors as well as in radial pumps have been calculated. The accordance to measurements is satisfactory. In order to learn more about the foundation of rotating stall in radial pumps, simulations with other already measured impellers are to be done. Therefore the whole characteristic diagram will be simulated.