Dynamic Radial Forces and Pressure Fluctuations Measurement at Off-Design Conditions on a Model Scale Pump-Turbine

V Novotný¹, V Habán², A Skoták¹, R Loub¹
¹ Litostroj Engineering a.s., Čapkova 2357/5, Blansko, Czech Republic
² Victor Kapan Dept. of Fluid Engineering, BUT, Technická 2896/2, Brno, Czech Republic

vojtech.novotny@litostrojpower.com

Abstract. The main challenge in the pump-turbines development are hydraulic instabilities in the partial loads in both pumping and generating mode. For the mechanical design of the pump-turbine shaft and bearings, it is necessary to know the magnitude of the dynamic radial forces caused by these instabilities. The radial force magnitude is usually determined on the basis of head and the runner dimensions. The paper focuses on the comparison of the empirical approach to the radial forces and the measurement on the model scale pump-turbine that was carried out in the Litostroj Engineering hydraulic laboratory in Blansko, Czech Republic. There were six strain gauges mounted on the bearing support to monitor the radial forces. In the pumping mode two distributor openings were measured. In the generating mode eight distributor openings were measured with the focus on the S-shaped curves region in the characteristic. Additionally, there were pressure gauges installed in the vaneless gap between the distributor and the runner to monitor the pressure fluctuation and potential presence of rotating stall. The appearance of rotating stall was connected to the magnitude of dynamic radial force.

1. Introduction
When designing a turbine shaft and/or bearings, the magnitude of radial force acting on the runner is an essential value for mechanical engineers. It is usually determined on the basis of head and runner dimensions. This paper focuses on the verification of the empirical approach to the radial forces: by means of the measurement on the model scale pump-turbine, the value of dynamic radial force coefficient k is determined.

2. Model Device Description
Experimental measurement took place in 2018 in Litostroj Engineering hydraulic laboratory on the model scale pump-turbine of ns = 130. The parameters of the model are following:

Runner characteristic diameter \( D_1 = 450 \, \text{mm} \)
Number of runner blades \( Z_R = 9 \)
Number of guide vanes \( Z_{GV} = 20 \)
Number of stay vanes \( Z_{SV} = 19 + 1 \) (spiral nose)
Besides the usual characteristic measurement there were six strain gauges mounted on the bearing support to monitor the radial forces (the arrangement of the strain gauges is in Fig. 1) and pressure sensors placed in the draft tube and in the vaneless gap between the runner and the guide vanes (the arrangement of vaneless gap pressure sensors is in Fig. 2). The sampling frequency of the measuring devices was 4.8 kHz.

Fig. 1 Arrangement of the strain gauges

Fig. 2 Arrangement of the pressure sensors in the vaneless gap

3. Experimental Results
First, the whole four quadrant static characteristic was measured for model scale guide vane opening $a_0 = 1 \div 36$ mm (Fig. 3). Then the dynamic measurement was carried out in unstable operating regions, i.e. in the “S-shaped curves” region in turbine mode and in the “hump” region in the pumping mode. These regions are marked red in Fig. 3.

Fig. 3 Four quadrant characteristic of the investigated pump-turbine
The main goal of the measurement was to determine the magnitude of dynamic radial force coefficient which is usually used for determination of dynamic radial force [3]:

\[ k = \frac{F_D}{D_1 \cdot b \cdot H \cdot \rho \cdot g} \]

where \(F_D\) [N] measured dynamic component of radial force,
\(D_1\) [m] runner characteristic diameter,
\(b\) [m] distributor channel height,
\(H\) [m] net head,
\(\rho\) [kg\(\cdot\)m\(^{-3}\)] density of water,
\(g\) [m\(\cdot\)s\(^{-2}\)] gravity acceleration at model test rig.

The dynamic radial force value for an operating point is an RMS value of the set of values measured during a time interval of \(t = 5\) s. Another goal of the measurement was to determine the presence of rotating stall, its rotational frequency, number of stalled cells, etc.

### 3.1 Turbine Mode

In the turbine mode, the measurement was carried out in the “S-shaped curves” region for model scale guide vane opening \(a_0 = 20 \div 34\) mm (Fig. 4). All the measurements were carried out for constant speed \(n = 1100\) rpm.

The color of individual points in Fig. 4 represents the presence/absence of the rotating stall (RS): blue color means operating point without RS, red color means operating point with developed RS. Pink color means “unstable vortices”, i.e. operating points where the vortices that block the runner channels occur, but there is no periodicity observed, the vortices emerge randomly. Purple color means a transition between “unstable vortices” and developed RS. Black points and black curve represent runaway speed. The identification of RS presence/absence is based on the pressure oscillation records from the pressure sensors in the vaneless gap.

For the detailed description the guide vane opening \(a_0 = 32\) mm was chosen (Fig. 5). Specifically following points: 2405 – operating point in stable region of the characteristic; 2409 – runaway speed; 2425 – turbine brake, “S-shaped curves” region. The numbers in Fig. 5 represent the number of
operating point and the relevant value of dynamic radial force coefficient $k$ respectively. The colors and its meaning correspond to previous graph (Fig. 4); the size of the dots represents the value of coefficient $k$. The comparison of the color and the size of the dots in this case reveals a close relation between rotating stall and the magnitude of dynamic radial force. In turbine mode there is only one zone of rotating stall which causes significant radial forces [5], [2].

![Fig. 5 Turbine mode, $a_0 = 32$ mm](image)

On the following figures (6 – 8) there are records of pressure fluctuations for specific operating points and corresponding radial force orbits. The thin pale lines represent the actual pressure fluctuation records. Its high frequency represents impeller rotation frequency multiplied by the number of guide vanes. The thicker lines are numerically filtered values to depict low frequency which helps to identify presence/absence of rotating stall: in first two cases (OP 2405 and 2409) there is no RS present, on the records there is no periodicity apparent. In the third case (OP 2425) there is developed RS present. The periodicity and the phase shift between the subsequent sensors are evident. The frequency determined by FFT corresponds to 60 % of runner rotational frequency, which corresponds to literature [4].

In the stable region of the characteristic the values of dynamic radial force coefficient are around $k = 0.1$, in optimum the value is probably even lower. In “S-shaped curves” region the value is up to $k = 0.55$. According to [1] and [3] the value grows with guide vane opening. This trend is, however, not confirmed by the measurement.

![Fig. 6 OP 2405: a) pressure fluctuations record; b) dynamic radial force orbit](image)
3.2 Pump mode

In the pump mode, the measurement was carried out for guide vane opening \(a_0 = 20\) and 24 mm. Both measurements were carried out for constant speed \(n = 1200\) rpm. For the detailed description the guide vane opening \(a_0 = 24\) mm was chosen (Fig. 9).

The characteristic can be divided into three regions: right branch of the characteristic curve (stable region, operating region of the machine), rotating stall region (instability) and the left branch of the curve. One representative operating point was chosen from each of these three regions (OP 2219, 2227 and 2253; marked bald in Fig. 9). On the following figures (10 – 12) there are pressure fluctuation records of the chosen points.
In the right branch of the characteristic (OP 2219, Fig. 10), there are no significant pressure fluctuations present. In Fig. 11 there are pressure fluctuations in rotating stall region depicted. The phase shift between the subsequent sensors is clear. There are three zones of rotating stall with the frequency of 3.6 % of runner rotational frequency. The small value of coefficient $k$ indicates that the radial forces caused by three high-pressure zones act against each other and the resultant force is therefore relatively small. In the left branch of the curve (OP 2253, Fig. 12) there are significant pressure oscillations, but the resultant force is relatively small. Again, the individual forces act against each other.

For model scale guide vane opening $a_0 = 24$ mm the value of the coefficient is in the range $k = 0.03 \div 0.06$. For $a_0 = 20$ mm the value of the coefficient is in the range $k = 0.02 \div 0.08$. 

![Fig. 10 OP 2219, right branch of the characteristic curve: a) pressure fluctuations, b) dynamic radial force orbit](image)

![Fig. 11 OP 2227, rotating stall region: a) pressure fluctuations, b) dynamic radial force orbit](image)

![Fig. 12 OP 2253, left branch of the characteristic curve: a) pressure fluctuations, b) dynamic radial force orbit](image)
4. Conclusion
The experimental measurement of dynamic radial forces was carried out on the model scale pump turbine. The findings are following:

- in the turbine mode in the stable region of the characteristic, the dynamic radial force coefficient value is around \( k = 0.1 \)
- in the “S-shaped curves” region the rotating stall with one stalled zone occurs and its frequency corresponds to \( 57 \div 63 \% \) of runner rotational frequency
- the RS causes significant radial force, in the worst case the coefficient value is \( k = 0.55 \)
- the assumption that the coefficient value grows with guide vane opening was not confirmed by the measurement

- in the pump mode for model scale guide vane opening \( a_0 = 24 \text{ mm} \) the value of the coefficient is in the range \( k = 0.03 \div 0.06 \); in the rotating stall region there are three stalled zones with the frequency of \( 3.6 \% \) of runner rotational frequency
- for model scale guide vane opening \( a_0 = 20 \text{ mm} \) the value of the coefficient is in the range \( k = 0.02 \div 0.08 \); in the rotating stall region there are three stalled zones with the frequency of \( 4.7 \% \) of runner rotational frequency

Acknowledgement
The authors of this paper would like to thank the Technology Agency of Czech Republic for the support in the project No. TE02000232.

References
[1] Etter S, Gummer J H, Seidel U, 2000: Measurement of the Forces on the Shaft of a Model Hydraulic Turbine and their Application to the Prototype. 20th IAHR Symposium, Charlotte.
[2] Ješe U, Novotný V, Skoták A, 2018: Development Trends in the Field of Reversible Pump Turbines – Study of Pumping and Generating Mode Off-design Conditions. 29th IAHR Symposium, Kyoto.
[3] Liess C, Jaeger E U, Klemm D, 1984: Hydraulically Induced Radial Forces on Francis Turbines and Pump Turbines: Measurement, Evaluation and Results. The 12th IAHR Symposium, Stirling.
[4] Skoták A, Mikulášek J, Střecha L, 2017: Metodika řešení problematiky vibrací vodních turbín vznikajících následkem rotujícího odtržení při turbinovém chodu čerpadlových turbín v oblasti „S“ křivek s cílem eliminovat „S“ tvar. Součást projektu Centra kompetence. Č. zprávy VAV-2017-1232, Blansko.
[5] Veselý J, Půlpitel L, Troubil P, 2005: Model Research of Rotating Stall on Pump Turbines. HYDRO 2005, Villach.