SOME RESULTS OF FORCE MEASUREMENTS ON THE IMPELLER OF A MODEL PUMP-TURBINE

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SYNOPSIS

An improved measuring shaft for measurement of radial forces and axial thrust is described. This device measures steady and unsteady loading on the rotating shaft close to the impeller of the model machine. The influence of the dynamic behaviour of the measuring system is, to a great deal, eliminated by simultaneous measurement of radial acceleration and by subtracting mass forces of the impeller from the measured force signals.

Force measurements, over four quadrants of operation, on the impeller of a pump-turbine reveal typical deterministic and non-deterministic signal contributions. Peak-to-peak values (defined with the 99.8% confidence interval) give an indication for overall force fluctuations. As an example for deterministic force fluctuations, the forces due to rotating stall detected at turbine brake operation are described in detail.
1. Introduction

Fluid forces on the impeller of a model pump-turbine \( n_s = 190; v_s = 0.33 \) were determined experimentally. These fluid or hydrodynamic forces result from non-uniform pressure loading on the impeller. Measurement of this loading was performed on the shaft close to the fixation of the impeller and included determination of mean static forces as well as of fluctuating force contributions. The loading is described by three force components \( (F_x, F_y, F_z) \) and three torque components \( (M_x, M_y, M_z) \).

**Axial thrust** \( (F_z) \) mainly results from the difference in the pressure distributions on the impeller shroud and hub as well as from the pressure distributions within the impeller channels and thus of the change of momentum of flow through the impeller.

The total of the measured **radial forces** \( (F_x, F_y) \) is composed of contributions due to non-uniform pressure distributions in the volute, due to static eccentricity of the impeller, due to a non-uniform flow field at the impeller inlet, and due to labyrinth forces. Fluctuating components arise from unsteady interaction of impeller and guide vanes, from the so-called "hydraulic unbalance" due to minor geometrical differences of the impeller blades, from flow separation at partial load, and from unsteady flow conditions at the inlet. A discussion and overview of parameters influencing hydraulic forces on centrifugal impellers is given e.g. by Gűlich, Jud and Hughes [5].

The two torque components \( (M_x, M_y) \) allow to determine the point of application of the radial forces and the torque component \( (M_z) \) describes the mean static value and the fluctuations of the driving torque. The data presented in the following text will be restricted to the radial forces.

Measuring points were taken at representative operating conditions in all four quadrants with positive head.

**Data evaluation** was performed in the rotating as well as in the fixed coordinate system. Evaluation included determination of mean static forces and of statistical properties of the fluctuating signal contributions. Samples of spectral density functions in the frequency-domain are given, as well as samples of time domain representations.

Force coefficients employed in this paper are defined as follows:

- for rotating coordinate system
  
  \[
  \begin{align*}
  F_{\xi 11} & = F_\xi (\rho g H D_1^2) \\
  F_{\eta 11} & = F_\eta (\rho g H D_1^2)
  \end{align*}
  \]

- for fixed coordinate system
  
  \[
  \begin{align*}
  F_{x 11} & = F_x (\rho g H D_1^2) \\
  F_{y 11} & = F_y (\rho g H D_1^2)
  \end{align*}
  \]

\( (D_1 = \text{outer diameter of impeller}) \)

2. Measuring system

Forces were measured with an improved version of a rotating measuring shaft, as described e.g. by Bachmann [1]. This device, a Sulzer Escher Wyss development (see Barp [2] and Grein et al. [4]), is basically instrumented with six full bridges of strain gages. Improvements concerned design of the shaft, electronics, data acquisition and analysis, and also included
elimination of mass forces of the runner.

The measuring shaft, completely mounted on its bearing body, was statically calibrated in a test stand at the Swiss Aircraft Factory, Emmen, for all possible combinations of loading and directions of force application. Additional static and dynamic checks of this calibration were performed on site.

Signals are preamplified on the rotating shaft and transmitted over a slip ring assembly to the data acquisition equipment, as schematically shown in figure 1.

Fig. 1: Assembled measuring shaft and schematic display of data acquisition

The rotor's radial acceleration is measured with a total of 4 piezoelectric accelerometers (PCB) which are mounted inside the force measuring device close to the joint of measuring shaft and impeller (indicated in figure 1). This measurement includes acceleration due to gravity since transducers are mounted on the rotating shaft.

Fig. 2: Forces acting on measuring system
The new concept employed here and described in the following, is based on the elimination of mass forces, caused by vibration of the measuring system, from the force signals measured with strain gages. This subtraction of mass forces (including weight) of the runner is performed in the time domain.

When measuring fluctuating fluid forces it is important to define exactly the boundaries between the fluid dynamic system and the measuring system; Fluid forces result from non-uniform pressure loading on the impeller, i.e. the wetted surfaces.

The separation of measuring system and fluid system is depicted in figure 2 (fluid system is hatched; measuring system is dotted). The wetted surface forms the boundary between the two systems. The fluid forces, \( F_{\text{Fluid}} \) are defined as the integral of the instantaneous pressure distributions along the wetted surfaces.

The total fluid forces are composed of external fluid forces, \( F_{\text{Fluid ext}} \) and, to a certain extent, of response forces of the fluid on vibration of the measuring system, \( F_{\text{Fluid resp}} \). Figure 2 also indicates external mechanical forces on the measuring system, \( F_{\text{mech ext}} \).

The external fluid forces, \( F_{\text{Fluid ext}} \) are caused by external pressure and velocity fields, which are not influenced by vibrations of the impeller. Such pressure and velocity fields can arise from turbulence in the oncoming flow, from swirl vortices, or from fluctuations in the closed test circuit. Furthermore, external fluid forces are induced by externally excited vibration of the housing (spiral casing, draft tube).

On the other hand, response forces of the fluid, \( F_{\text{Fluid resp}} \), are those contributions to the total fluid forces which vary under influence of impeller vibration. Typical example for such force contributions are forces caused by pressure fluctuations in the labyrinth seals, which are induced by vibrations of the measuring system. Furthermore, forces due to acceleration of the so-called "hydrodynamic mass" contribute to \( F_{\text{Fluid resp}} \).

Coupling of the fluid system and the measuring system is schematically depicted in figure 3. The measuring system is a purely mechanical system, which is composed of the impeller, the measuring shaft, the shaft plus bearings, the coupling, and further elements. This measuring system is forced to vibrations under influence of fluid forces and of external mechanical forces. Simplifying, the vibrations of the measuring system are indicated in figure 3 as bending vibrations \( \xi(t) \) and \( \eta(t) \).

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Fig. 3: Coupled fluid and mechanical system
Fig. 4: Simplified model of the measuring system

In order to show the principles of elimination of mass forces in case of radial force measurement, the measuring system is approximated as a system of several masses. The mass \( m_1 \) in figure 4 symbolizes the mass of the impeller (effective at the measuring shaft) while the spring \( c_1 \) corresponds to the spring of the relatively flexible measuring shaft. The mass \( m_2 \) corresponds to the shaft; further elements, as listed above, could be attached. The equation of motion of the first two masses can be written as follows:

\[
\begin{align*}
\ddot{m}_1 \xi_1 + d_1 (\dot{\xi}_1 - \dot{\xi}_2) + c_1 (\xi_1 - \xi_2) &= F_{\text{Fluid}}(\xi) \\
\ddot{m}_2 \xi_2 - d_1 (\dot{\xi}_1 - \dot{\xi}_2) + d_2 (\dot{\xi}_2 - \dot{\xi}_3) - c_1 (\xi_1 - \xi_2) + c_2 (\xi_2 - \xi_3) &= 0
\end{align*}
\]

With acceleration transducers we are able to measure the acceleration \( \ddot{\xi}_1(t) \) (actually two radial directions on the rotating shaft). The signals we get from strain gages on the measuring shaft are proportional to the displacement difference \( \xi_1(t) - \xi_2(t) \).

Thus, assuming small mechanical damping and neglecting the term \( d_1 (\dot{\xi}_1 - \dot{\xi}_2) \), we can directly determine the fluid forces solving equation (1). For the second, orthogonal direction \( \eta \) analogous equations can be formulated. This operation for determination of the fluid force is digitized performed in the time domain for each sampling point.

The effectiveness of this elimination of mass forces due to impeller vibration and gravity was proved in experiments of rotation in air and in experiments of free vibrations in air. In both cases fluid forces \( F_{\text{Fluid}} \) go to zero or can be neglected while measured mass forces \( m_1 \ddot{\xi}_1 \) and spring forces \( c_1 (\xi_1 - \xi_2) \) are of the same magnitude but \( \pi \)-out-of-phase.

In addition, employing a simple model, it was possible to determine the so-called "hydrodynamic mass" from the fundamental natural frequency (bending mode) determined from free oscillation in air \( (f_0 = 138 \text{ Hz}) \) and from free oscillations in stagnant water \( (f_0 = 95 \text{ Hz}) \). The determined ratio of "hydrodynamic mass" over runner mass equal to 2.11 is in good agreement with that found by Thuss [7] or Bolletter et al. [3].

**Transformation** of force components in rotating coordinate system into fixed coordinate system is given by:

\[
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} =
\begin{bmatrix}
\cos 2\pi f_0 t & -\sin 2\pi f_0 t \\
\sin 2\pi f_0 t & \cos 2\pi f_0 t
\end{bmatrix}
\begin{bmatrix}
F_\xi \\
F_\eta
\end{bmatrix}
\]

where \( f_0 \) is the rotational frequency, and \( F_x \) and \( F_y \) are the fluid forces transformed into the fixed coordinate system.
This transformation from the rotating coordinate system into the fixed coordinate system is performed digitally in the time domain for each sampled point.

3. Data acquisition, statistical properties, and frequency spectra of signals

Analog amplifier outputs were filtered with identical filters in order to avoid aliasing effects and undesired phase shifts between the signals. All signals were digitized with an Hewlett-Packard Multiprogrammer equipped with two scanning units, two 12-bit A/D converters, and two 512-kByte Buffer memory. Signal analysis was performed on a HP-9000 computer with an installed RAM-Memory of 5 Mbyte.

For each measuring point and channel 16384 data points are sampled, covering a total of 256 revolutions of the rotor.

Evaluation of statistical properties includes evaluation of mean values from which the static forces on the impeller are calculated. The peak-to-peak values of the unsteady signal components provide an indication of the level of vibration at a given measuring point. The peak-to-peak analysis is a representative one if peak values are constant or vary only little with time. The analysis of effective values (standard deviation) is specially useful for analysis of essentially random signals (stochastic fluctuations). The standard deviation generally gives values which are smaller than half the peak-to-peak values. The confidence interval gives a band within which all sampled points lie with a given probability (e.g. 99.8%). This interval was chosen here to provide a reasonable estimate for overall fluctuations of signals which are composed of deterministic plus random signal contributes. Thus, in accordance with publications e.g. by Liess et al. [6], the 99.8% confidence interval is used to characterize the fluctuating signal components. Actually, this interval corresponds to the peak-to-peak values assuming that in each ensemble of 1024 sampled points one single point with the smallest value and one single point with the largest value has to be eliminated, e.g. due to faulty measurement or A/D-conversion. Plots of histograms as shown in figure 5 (averaged from 16 ensembles) confirm the suitability of the 99.8% confidence interval.

In the following, maximum values of force fluctuations (index max) are defined employing the 99.8% confidence interval of the signals divided by two.

![Histogram of \( F_x \)](image)

Fig. 5: Example of histogram
Besides analysis of signal properties in the time domain, analysis in the frequency domain is performed. Displayed are the so-called linear spectra (= square root of power spectral density function). An example of a linear spectrum of $F_{x11}$ at normal pump operation is displayed in figure 6. Amplitudes of important discrete frequency components (e.g. at rotational frequency $f_0$ and higher harmonics and at the blade passing frequency $f_1 = 7 f_0$) were determined automatically and listed in print outputs.

![Linear spectrum example](image)

Fig. 6: Example of linear spectrum (same point of operation as shown in figure 5)

4. Measurements

Measurements were taken at different steady-state points of operation on the Ku/Kcm-characteristics, including normal pump or turbine operation, reverse pump operation, pump brake or turbine brake operation, as well as operation at runaway speed. The grid of measuring points, indicated in figure 7, includes 4 different guide vane openings ($F_1 = 5^\circ; 15^\circ; 25^\circ; 35^\circ$). These representative steady-state points were selected so that the extreme static and fluctuating forces, occurring at steady and transient operating conditions, could be determined.

5. Representation of model test results

The following graphs contain material on mean static forces (index s) and maximum force fluctuations (index max). These forces are averaged from 256 revolutions of the rotor. Maximum force fluctuations correspond to values evaluated from half the 99.8% confidence interval.

Four quadrant overviews

In figure 8 mean static values and maximum force fluctuations of radial forces are displayed as a function of the speed coefficient $Ku$. The chosen typical example shows data taken at a wicket gate opening of $35^\circ$.

If force levels are small enough and can be accepted from the technical point of view, there is no need for further analysis of the data, except for research purposes. However, in the chosen example strong peaks appear in the shape of a loop for turbine brake operation. (Although the forces are high, for prototype operation these peak values cannot be of any danger, because steady state operation of the prototype for these turbine brake operating points is not possible).
Kul = \frac{UL}{\sqrt{2gH}} \quad ; \quad Kcml = \frac{Q}{D^2 \frac{\pi}{4} \sqrt{2gH}}

Fig. 7: Four quadrant characteristics showing the discharge coefficient Kcml as a function of the speed coefficient Kul

Fig. 8: Mean static values (index s) and maximum fluctuations (index max) of radial forces as function of the speed coefficient Kul

Displaying the turbine brake and reverse pump operating points as a function of the discharge coefficient Kcml, figure 9, eliminates the loop of the force coefficients and clarifies the influence of discharge variation.
Fig. 9: Mean static values (index s) and maximum fluctuations (index max) of radial forces as function of the discharge coefficient Kcml

The following discussion will focus on these high amplitude fluctuations because they represent a rather interesting phenomenon.

Rotating stall

The force fluctuations observed for turbine brake operation are periodical and there is, as can be concluded from measurement of pessure fluctuation at various locations, a strong indication of rotating separations in the region of the wicket gate and impeller vanes. Thus, we term in the following, in analogy to similar effects observed for unstable pump characteristics, the cause for these high amplitude fluctuations rotating stall.

Amplitude spectra, figure 10, of the measuring point with the largest amplitudes in figures 8 and 9 reveal the dominance of a peak at the frequency $f_s$ at about 60% of the rotational frequency $f_0$. At the blade passing frequency $f_p$ no significant peak can be observed.

Indicated in figure 10 is the frequency $f_n = 95$ Hz determined from free oscillation in stagnant water. This frequency corresponds to the natural frequency of the coupled system (measuring system and fluid) with the water at rest and is not necessarily the same for the actual flow condition. However, we see that no frequency which could be related to $f_n$ is dominantly excited.

Interesting is also the fact that the spectra in x and y-direction are almost identical. We can conclude from this identity a constant force vector rotating with the frequency $f_s$ with respect to the fixed coordinate system.

The question whether this force vector rotates in the sense of rotation of the impeller, can be answered by looking at the time domain representations, as displayed in figure 11.

Figure 11 shows the force fluctuations in x and y-direction over a period of 16 revolutions of the rotor. Fluctuations $F_x$ clearly precede the fluctuations $F_y$ by quarter a period. Knowing that the sense of rotation was anticlockwise, we can conclude that the force vector and therefore the rotating separation rotates in the same sense as the impeller.

Figure 12 shows the same force fluctuations but here in the rotating coordinate system, i.e. in $\xi$ and $\eta$-direction. The frequency detected here is $f_0 - f_s$ and the fluctuations $F_\xi$ lag obviously the fluctuations $F_\eta$ by quarter a period.
Fig. 10: Frequency analysis (linear spectra) of $F_x$ and $F_y$ at the operating point where maximum forces are detected ($f_0 = 16.7$ Hz).

Fig. 11: Time domain representation of $F_x$ and $F_y$ (same measuring point as in figure 10).

Fig. 12: Time domain representation of the forces $F_\xi$ and $F_\eta$ in the rotating coordinate system (same measuring point as in figure 10).

6. Conclusions

The improved concept of measuring and analysing the loading on the impeller of model machines proved, in a first set of measurements, to be adequate. The influence of the dynamic behaviour of the measuring system could be, to a great deal, eliminated by simultaneous measurement of radial acceleration and by subtracting mass forces from the measured force signals.

Force measurements, over four quadrants of operation, on the impeller of a pump-turbine reveal typical deterministic and non-deterministic signal
contributions. However, maximum force fluctuations detected at turbine brake operation were dominated by periodical signal contributions. These fluctuations are associated with the instability in the Kcml/Kul-characteristic and are caused by rotating separation effects in the region of the wicket gate and impeller vanes, i.e. rotating stall. For these maximum force fluctuations spectra and time domain representations reveal, with respect to the fixed coordinate system, a force vector of, in good approximation, constant length which rotates in the same sense as the impeller does but at lower frequency \( f_s = 0.6 f_0 \).

References


