A HIGH-FREQUENCY-RESPONSE PRESSURE PROBE FOR THE MEASUREMENT
OF UNSTEADY FLOW BETWEEN TWO ROTORS
IN A HYDRODYNAMIC TURBOMACHINE

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ABSTRACT
A new, specially-developed high-frequency-response pressure probe was used to measure the unsteady flow in the interaction region between the pump and the turbine in a hydrodynamic torque converter. In order to reduce the probe diameter, a single-hole, single-sensor cylindrical probe (\(\varnothing=1.33\)mm) was developed, to replace the standard multi-hole probe. The smaller the probe the higher the accuracy in unsteady flow. Therefore this is an improvement over three-hole probe. Three-hole probe measurements were simulated by recording data in three different angular positions. The time variable velocity vectors were determined using the probe's calibration coefficients and the knowledge of the rotor positions (measured by angle-encoders) for every measurement value. During the data processing, a double ensemble averaging was carried out, taking into account the positions of the pump and the turbine.

LIST OF SYMBOLS:
- \(m\) mass flow [kg/s]
- \(\alpha\) flow angle [\(^\circ\)]
- \(\eta\) efficiency
- \(\mu\) torque ratio
- \(v\) speed ratio
- \(\theta\) flow angle (yaw) relative to the probe
- \(\varnothing\) diameter

Sub- and superscripts
- (...)\(_p\) counting index pump
- (...)\(_t\) counting index turbine
- (...)\(_m\) meridional
- (...)\(_m\) measured
- (...)\(_h\) harmonics
- (...)\(_p\) pump
- (...)\(_t\) pressure medium
- (...)\(_t\) turbine
- (...)\(_h\) circumferential
- (...)\(_o\) e.g. total pressure \(p^o\)

INTRODUCTION
Velocity and pressure measurements are carried out in many areas of technology. Different measurement methods are used, depending on the type of application. Accessibility of the measurement region, the flow medium and the required level of accuracy and precision are among the factors which determine whether standard test equipment can be used, or whether a new system must be specially developed.
Velocity measurements are very important when investigating the flow field in turbomachinery, where the flow is usually non-steady and three-dimensional. Only if there is a thorough understanding and knowledge of the flow field is it possible to develop better machines in the future. In the past, it was only possible to investigate the quasi-steady mean values of the flow conditions, but modern measurement technology enables the study of non-steady flow phenomena and thereby the improvement of turbomachinery design.

Hydrodynamic torque converters are used as a link between motor and driven machine. Their flow field is highly unsteady [14,15], particularly in the interaction regions between the pump, turbine and stator. Earlier investigations concentrated on lab-model simulations using laser velocimetry [3,6,7,8] and hot-film techniques [4,5]. The shortcomings of the laser methods in highly turbulent wakes and of destruction of hot films by high fluid velocities have been overcome by the present type of sensor. This work aims at a far-future target of improving the prediction of the changing torque-converters characteristics under unsteady operating conditions. For this reason measurements of the unsteady flow field between the two rotors have been carried out. Numerical flow calculation have illustrated the need for an unsteady flow measurement system, which is also capable of recording time-variable pressure profiles for comparison with pressure-velocity computer codes [16]. Therefore a measurement system with a newly-developed, fast response pressure probe has been developed and employed for this research.

This report will describe the measurement system, which includes a newly-developed probe, and the data acquisition system. The results of the measurements between the pump and the turbine demonstrate the capabilities of the concept and offer new insights into the flow characteristics in a torque converter.

EXPERIMENTAL HYDRODYNAMIC TORQUE CONVERTER

The experimental studies were carried out in the flow circuit of a single stage, single phase hydrodynamic torque converter, filled with water which is convenient for mining purposes (environmental reasons). Fig. 1 shows the cross section of the torque converter and the location of the measurement plane (1) between the pump and the turbine, both rotating at different relative speeds [2,18] The unshrouded pump impeller has 16 backswept blades and an outlet diameter of 340 mm. The shrouded turbine rotor carries 40 blades and the stator has 32 vanes. The unsteady flow investigations were carried out at pump speed \( n_p = 400 \, \text{min}^{-1} \) with the speed ratios (turbine/pump) of \( v=0.55 \) and \( v=0.9 \).

Pressure probes equipped with semi-conductor pressure transducers are suitable for the measurement of time-variable pressure and velocity. A silicon or metallic diaphragm in the sensor serves as a conversion element between the pressure and voltage signals. Mechanical stress in the diaphragm is measured utilizing the piezoresistive effect. The high frequency pressure probe has been developed under the requirements of smallest possible size for minimum flow disturbance, use in water, high sensitivity and accuracy, an elevated frequency range, an unambiguous calibration characteristic and absence of dynamic flow separation by choosing a cylindrical probe shape [10,12,13].

Fig. 2 shows the cross section of the new probe. In order to reduce the probe diameter a single-hole, single-sensor cylindrical probe (\( \Theta=1.33 \, \text{mm} \)) was developed, rather than a multi-hole probe with a typical size of about 3 times the cross section. With a single-hole probe, a three hole probe measurement can be carried out by taking three separate measurements in different angular positions (\( \theta=0^\circ, \theta=\pm 45^\circ \)). Commercially available miniature pressure transducers are either difficult to install or unsuitable for use water. Therefore a modified miniature pressure sensor chip, which was originally created for the biomedical industry, is used.

The pressure sensor itself is fixed on a support, over which the outer probe body is mounted. Due to the orientation of the sensor chip - the electronically active side of the diaphragm is located against the probe interior - it is possible to use the probe in water.

The ratio of the distance \( a \) of the pressure tap from the probe tip to the probe diameter \( d \) is \( a/d=1.6 \). This depends upon the size of the pressure transducer inside the probe head and determines the characteristics of the cylindrical probe [19].

![Fig. 1. Experimental hydrodynamic torque converter with pump impeller P, turbine T and stator guide vanes S](image-url)
Fast frequency response pressure probes, as used in turbomachinery, should be capable of a resolution of at least ten times of the characteristic blade passing frequency (fp=106.6 Hz in this case). Due to the flushness of the pressure-sensor installation to the probe wall (set back by 0.7 mm from the probe body) and its high natural frequency of 500 kHz, the signals within the frequency range 0-2.4 kHz can be measured without amplitude or phase errors. Calibration takes place in a steady channel flow by varying the yaw angle of the probe [1].

**DATA ACQUISITION AND TEST PROCEDURE**

The data acquisition system, which corresponds widely to the procedure described in [1,4], is shown in Fig. 3.

![Fig. 3: Test facility and measuring equipment](image-url)

The data acquisition system consists of a computer, a 12 bit analogue/digital conversion card with 150 kHz maximum sampling rate and several interface cards. An incremental angle encoder with one pulse every 0.5° and one reference pulse per revolution is connected to the pump shaft. It triggers the A/D-converter. Due to the reference pulse, sampling always starts at a defined zero position of the pump impeller. It is therefore possible to calculate the real pump impeller position. The actual position of the turbine is measured with an absolute encoder (1000 steps per revolution), which is driven by the turbine shaft. After the reference pulse of the incremental encoder has started conversion, the analogue values of the probe and the turbine encoder are measured for every trigger signal by the A/D converter. Additionally the voltages of the resistance thermometers, the steady pressure transducers and the frequency/voltage converter, which produces speed-proportional voltage values from the signal of the encoders, can be recorded by a data logger.

Beginning with the flow angle, which is determined by a 5-hole-probe for the average flow direction, a total of 5 measurements are carried out at the angular positions $\theta = 0^\circ, \pm 10^\circ$ and $\pm 45^\circ$. Total pressure, flow velocity and direction can be calculated, as described in [1] from these measurement, downstream of the impeller. 216000 values (=300 pump revolutions) are acquired per measurement and saved as binary data. At a pump speed of 400 min$^{-1}$, this necessitates a measurement time of 45 s. The data are finally converted and processed on a Personal Computer.

**DATA EVALUATION**

The data evaluation can be subdivided into three stages. In the first step, the raw data are analysed with a Fast Fourier Transform (FFT). The voltages are then converted into pressures and the constant pressurisation, required to prevent cavitation, is subtracted Eq. (1).

\[ \Delta p = \Delta p_{me} - \Delta p_{p} \]  

(1)

The values are analysed in the second step with reference to the angular position of the pump impeller and the relative position of the turbine, using ensemble averaging, as described in [1,4,5].

\[ \overline{P}_i = \frac{1}{n_y} \sum_j P_{ij} \]  

(2)

In order to reduce measurement time and quantity of data, each blade pitch is considered to be identical. The pump and turbine positions are superposed with 2 pump pitches enclosing 5 turbine pitches. There are a total of 720 circumferential triggering pump positions (16 blades * 45 samples). The turbine angle is positioned with a resolution of 25 samples times 40 blades equal to 1000 increments. Therefore the data are classified in $i \times j = 1125$ different positions, i.e. the distributions can be represented in 25 different relative positions over a pump pitch. The pressure fluctuations $\delta P_{ij}$ are also calculated for every relative position Eq. (3).

\[ \delta P_{ij} = \frac{\sqrt{(P_{p} - P_{i})^2}}{\overline{P}_i} \]  

(3)

In the third step, the flow data are calculated for every relative position using the sorted and averaged pressures $P_{ij}$ and the calibration coefficients. The results correspond to the grid in Fig. 4.
MEASUREMENT RESULTS BETWEEN PUMP AND TURBINE

The total error of the pressure measurements can be calculated from the systematic residual error due to drift and creep of the sensor, which is considered to be a random error, and the random error itself, which is quantified with the aid of statistical methods (calculation of the convergence interval for every sorted and averaged value). In the worst case, the error is ±0.32 mbar. Under consideration of the error reproduction, an error of ±0.2% results in the velocity measurement and ±0.08° in the angular measurement. With superposition of the static systematic angular error due to velocity gradients and dynamic effects (as identified by [12]), the accuracy of the flow angle measurement is given to be ±0.84°. The total error of the velocity measurement is calculated with the pressure measurement error (±0.2%) and the error due to the probe blockage (±1.25%). This amounts to ±1.7%.

Fig. 5 shows the double ensemble average distribution of the absolute flow angle α as an example of the influence of the relative motion between the rotors. It is recorded at the speed ratio v=0.9 in the mid-channel for every relative position of pump and turbine, so that the time-periodic changes for the time frame in which the pump has moved one blade relative to the turbine can be observed in Fig. 5. The periodic angular variations caused by the turbine blades can be observed, as well as the variation of the angle in the region of the pump blade wake. These move counter-clockwise relative to the pump and there is a brief superposition of the pump and turbine blade wake (e.g. pump angle 15°-22°). It can be clearly seen that the turbine blades then ‘cut off’ the pump blade wakes. This effect of ‘wake cutting’ is also observed in a similar form at the interaction of the rotor and stator in axial compressors [17].

Further information about the flow field in the interaction area between the pump and turbine is delivered by the double ensemble average of the total pressure fluctuation coefficient. This can be seen for one relative rotor position at the speed ratio v=0.55 and v=0.9 in Fig. 6.

At v=0.55 a higher intensity fluctuation zone in the region of the channel front wall and the blade suction side points to a low energy area, a so-called ‘wake zone’, which is often found at the exit of radial impellers [5]. This wake zone retains its basic shape, despite the relative motion of the two impellers. The influence of the turbine blades on the fluctuation can only be detected in the region of the fluctuation core.

At the speed ratio v=0.9, the flow is, in contrast, less disturbed. An increased fluctuation is noticeable at the suction side zone at s/b=0.8-0.9. Since the wake area is principally caused by the flow bend in the radial pump impeller, the two distributions allow it to be assumed that the pump works only at part-load at v=0.55 (0.865 m/(v=0.55)=m/(v=0.9)). For this reason, the wake area increases in size between v=0.9 and v=0.55 and originates from the channel front wall in the region of the blade suction side mid-channel.
Fig. 6. Profile of the total pressure fluctuation coefficient $f_{kij}^0$ for one relative impeller position: $v=0.55$ (top) and $v=0.9$ (bottom).

Fig. 7 shows the double ensemble average of the meridional velocity $c_{mij}$ for a single relative position of the pump and turbine.

For $v=0.55$, the wake area also can be seen. Beginning from the suction side, it fills up to $2/3$ of the blade pitch and more than $50\%$ of the channel width. It can be characterised by the low meridional velocity, which also exists in the region of the pump blade wake. The blade thickness of the turbine causes a blockage at both speed ratios, which is enlarged by a short-term superposition of the pump and turbine blade wakes. The influence of the turbine can also be observed in the wake-area at $v=0.55$. This further reduces the local meridional velocity.

The percentage deviations between the doubly and simply ensemble averaged meridional velocities can be seen in Fig. 8. They show the influence of the turbine separately from that of the pump. The wakes caused by the turbine blades are approximately 25-30\% and 35-40\% smaller than the corresponding simple ensemble average of the values for $v=0.55$ and $v=0.9$, respectively. It can be seen that the turbine blades form obstructions for the pump flow. Channel size reductions and expansions arise due to the relative motion of the rotors, which periodically, as in the area of the pump angle between $-5^\circ$ to $-1^\circ$, lead to an increase of the meridional velocity of up to $50\%$ of the current simple ensemble averaged value.

Fig. 7: Meridional velocity flow field $c_{mij}$ for one relative position of the pump/turbine at $v=0.55$ (left side) and $v=0.9$ (right side); SS/PS = suction side/pressure side of the pump, TB = turbine blade.
CONCLUSIONS

Unsteady flow investigations were carried out in the interaction region between the pump and the turbine in a single-phase, single-stage hydrodynamic torque converter. For this purpose, a high-frequency-response single-sensor pressure probe was developed and used. Three-hole probe measurements were simulated by recording data in three different angular positions. The time variable velocity vectors were determined using the calibration coefficients and the knowledge of the rotor positions for every measurement value. During the data processing, a double ensemble averaging was carried out, taking into account the position of the pump and the turbine. Important conclusions drawn from this investigation are:

1. The miniature probe developed here proved to be a valuable tool in the investigation of the unsteady flow under the constrains of high frequency range and small-size flow domain.
2. The flow field at the design point (v=0.55) is characterised by a wake structure in the region of the pump blade suction side of the mid-channel. This structure is little influenced by the relative motion of the rotors.
3. The wake blockage which occurs at v=0.55 is already visible at v=0.9. It is mainly caused by the pump with the radial impeller, operating off-design at part-load for v=0.55. Thus the wake blockage size and shifts in position.
4. The upstream influences caused by the turbine blades are smaller by approximately 25-30 % for v=0.55 and 35-40 % for v=0.9 when compared as double ensemble average with the corresponding simple ensemble average values.
5. At the interaction area between the pump and turbine, there is a superposition of the pump and turbine blade influences ('blade cutting'). There is also channel narrowing and widening, which periodically leads to a local increase in the meridional velocity component of up to 50 % of the current average flow value.

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