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Original Paper

Fast Transition from Pump to Turbine Mode of Operation

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Abstract

The reversible pump turbine (RPT) is a suitable machine to control fluctuations in the energy market. The usage of RPTs for this purpose will increase the number of operational mode changes of the machine. In order to reduce the response time of the machine, fast transitions between the modes of operation are necessary. Therefore, increased knowledge of how the machine operates during these fast transitions is needed. This includes the investigation of the transient characteristics for the whole operating range of the machine. This paper presents experimental results from a fast transition from pump to turbine mode. The flow rate is measured by the use of a modified pressure time method. The resulting transient characteristics are compared with steady state characteristics. Experiments have been preformed on a model scale reversible pump turbine in the Waterpower Laboratory at the Norwegian University of Science and Technology (NTNU). The results show that the pressure pulsations are highest at low discharge in both pump and turbine mode of operation and at runaway speed in turbine. Oscillations at runaway speed is reduced with lower opening degree on the guide vanes. The results also show a difference between the steady state and transient characteristics in the pump mode due to the inertia of the water masses.

Keywords: Reversible pump turbine, Tranient operation, Mode change

1. Introduction

When discussing the future of the energy market, the reversible pump turbine (RPT) are often given the role of balancing the intermittent energy production. The introduction of intermittent renewable energy sources calls for a faster regulating force in the energy market. Balancing the energy production can be done by RPTs and require the machines to change the mode of operation more frequently. In such a scenario, the need is not only for an increase in operational mode changes, but also for a decrease in the transition time between the pump and turbine mode of operation.

In reversible pump turbines, as in Francis turbines, it is well documented that the machines experience high fatigue loads in off-design and start and stop operations [1]–[3].

For reversible pump turbines the instabilities that can occur during start up, both in pump mode [4] and turbine mode [5] are also areas where a lot of research is available.

There is, however, less information about the transition between the different operational modes. Early, and most well-known research on this topic is the four quadrant characteristics of a pump presented in [6] by Stepanoff. The experimental data describing the four quadrant characteristics in [6] was done by Knapp [7].

Ruchonnet and Braun [8] have presented transient characteristics from experimental research done from pump to turbine mode and from turbine to pump mode.

They refer to high-pressure pulsations during the transient period, and especially high amplitudes in energy dissipation mode. Stens and Riedelbauch [9] have investigated the same transient operation using Computational Fluid Dynamics (CFD). They conclude that the fast transitions lead to stall conditions between the guide vanes and vortices in the runner and draft tube, and that this is the cause of the off-design conditions in the runner. Liu et al. [10] have done CFD on a transient process caused by power failure from pump mode to turbine mode. The authors show that the minimum value for the transient head occurs when the flow rate is approximately zero, which means between energy dissipation mode and turbine mode of operation.

When connecting a hydraulic machine to the grid, the rotational speed of the machine have to be equal the synchronous speed. The most common way to achieve the correct rotational speed before connecting the machine to the grid is to let the machine rotate at runaway speed with a given opening of the guide vanes. It is therefore of interest to look closer at the systems behaviour at runaway speed after a fast transition from pump to turbine mode of operation.

This article investigates the transient characteristics of a reversible pump turbine, from pump to turbine mode of operation. An important part of the work has been the development of a reliable method to measure the flow rate in this scenario. The method of measuring the flow rate, together with the resulting transient characteristics, are presented in the article. The transient characteristics are also compared to the steady state characteristics for the same machine.

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2. Experimental setup

When doing transient experiments it is important to remove disturbances on the system caused by the pump, if run in closed loop. The Francis test rig was therefore arranged to operate in an open loop configuration when testing the reversible pump turbine. As seen in Fig. 1 the upper reservoir have a constant head regulated by an overflow valve (1) and fed by a centrifugal pump (6). Upstream the turbine (3) there is a pressurized tank (2) with an air cushion. The draft tube ends up in an outlet tank (4), and the tanks water level is held constant by another centrifugal pump (5). In order to disconnect the hydraulic inertia of the outlet system a weir is installed in the outlet tank. The head is defined as the total energy difference between inlet of the turbine and outlet of the draft tube according to the convention as seen in eq. (1).

Table 1 RPT runner parameters

D_1	D_2	B_1	n_{ed}	Q_{ed}	H^*	B_{hp}	B_{lp}	α^*
0.631 m	0.349 m	0.059 m	0.133	0.223	29.3	12°	12.8°	10°

The model runner is designed by Grunde Olimstad [11] during his PhD work. The laboratory set-up is, with some modifications, the same as used by Eve Walseth as part of her PhD thesis [12]. The runner parameters are listed in Table 1.

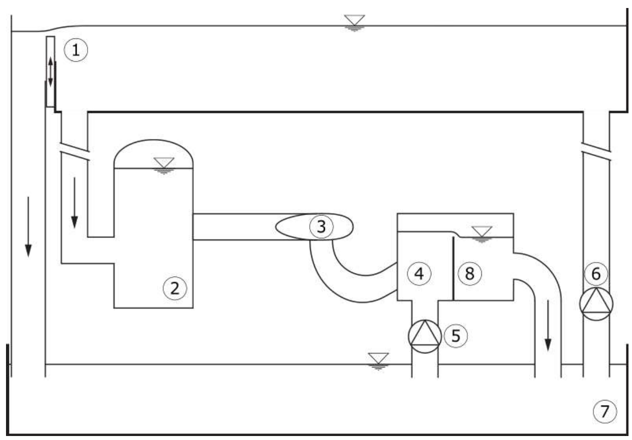


Fig. 1 The open loop configuration of the Francis test rig with; upper reservoir (1), pressure tank (2), RPT (3), draft tube tank (4), downstream feeding pump (5), upstream feeding pump (6), main water reservoir (7) and weir in draft tube tank (8)

2.1 Instrumentation

Flow rate, torque, rotational speed and pressure were measured during the experiment. Table 1 lists the different sensors, and details for each instrument used indicate the placement of each instrument. The head calculated as shown in eq. (1). The sample rate of 5000 Hz was used.

Table 2 Instrument details

Abbreviation	*Description*	*Sensor Type*	*Location*
p_{i1}	Inlet pressure	Druck UNIK 5000	Turbine inlet
p_{i2}	Upstream inlet pressure	Druck UNIK 5000	Inlet pipe
P_{vl}	Vaneless space pressure	Kulite	Vaneless space
P_{dt}	Draft tube pressure	Kisler	Draft tube upper
P_o	Outlet pressure	PTX	Draft tube lower
M	Torque	Hottinger	Main shaft
Q_m	Electromagnetic flowmeter	Krohne	Inlet pipe
n	Rotational speed	Encoder	Generator draft

$$H = \frac{p_{i1} - p_o}{\rho g} + \frac{Q_g^2}{2g} \cdot \left(\frac{1}{A_{i1}^2} - \frac{1}{A_o^2}\right) \tag{1}$$

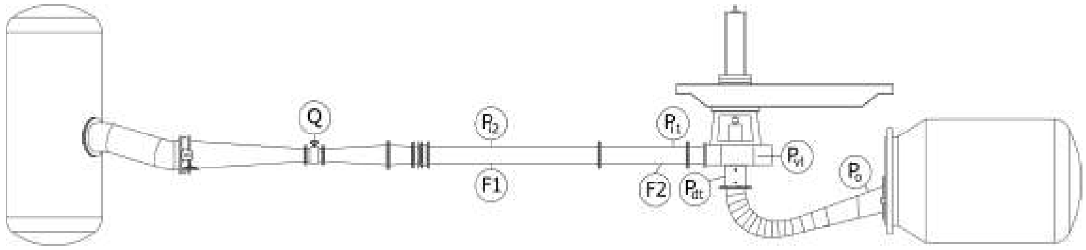
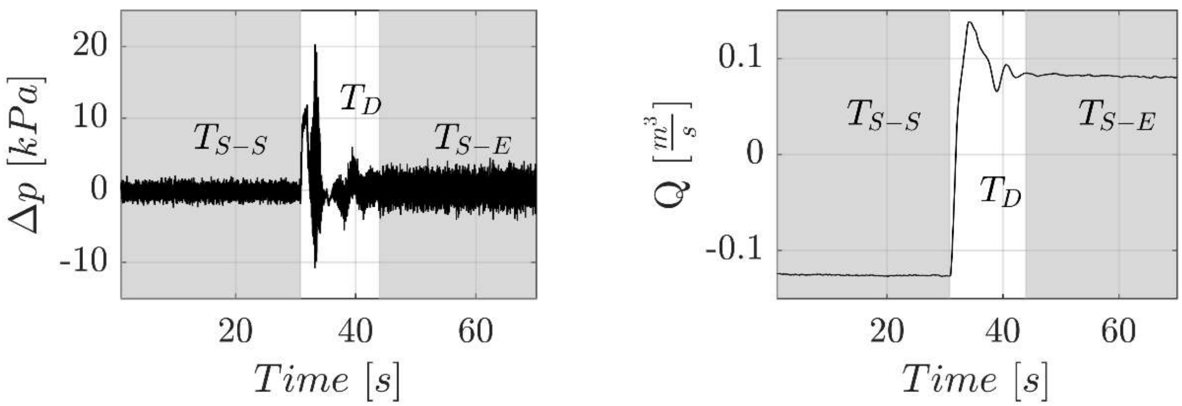


Fig. 2 Instrument locations in the test section

2.2 Transient flow rate measurement

To measure the discharge fast and with high accuracy, a modified pressure-time method was used. The pressure-time (Gibson) method referred to in IEC 60193 [13] is used for measuring the initial flow using the transient pressure difference as the guide vanes closes to zero opening. In this experiment however, a modified pressure-time method was used. By measuring the pressure difference over a certain pipe length, the pressure difference can be used to calculate the transient flow during the mode change, using the electromagnetic flow meter (EMF) as a verification of the flow at steady state, both at the start and the end of each transient sequence. Nielsen [14] was the first to use the pressure-time method to find the flow at each increment of time in a transient sequence, while doing tests in the turbine mode of operation.



(a) (b)

Fig. 3 The differential pressure ΔP (a) used to calculate the flow Q (b) where the steady state area at start and end are given by the EMF

Figure 3(a) shows the measured differential pressure between two pressure sensors (p_{i1} and p_{i2}) with a distance of 5.345 m, and with a constant cross sectional area between them. The transient flow rate as seen in Fig. 3(b) is found by integrating the pressure difference in accordance with the pressure-time method as stated in eq. (3) and eq. (4),

$$\Delta Q_i = \frac{A}{\rho L}\int_{t_{i-1}}^{t_i} (\Delta p + \zeta)\, dt \tag{2}$$

$$\zeta = kQ^2 \tag{3}$$

where ζ is the friction loss in the pipe section, k is the friction factor, ΔQ is the change in flow in one time step. A is the cross sectional area and L is the length of the pipe section. ρ is the water density and Δp is the differential pressure of the two pressure sensors. Each measurement is divided into three parts. $t_0 \leq T_{s\text{-}s} < t_1$ is the time period from the start of the steady state measurement up to the start of the transient period. $t_1 \leq T_D < t_2$ is the transient period where the RPT changes from pump to turbine mode of operation. $t_2 \leq T_{S\text{-}E} < t_3$ is the steady state period from the end of the transient measurement to the end of the measurement.

The flow rate in the transient section Q_g was obtained as follows:

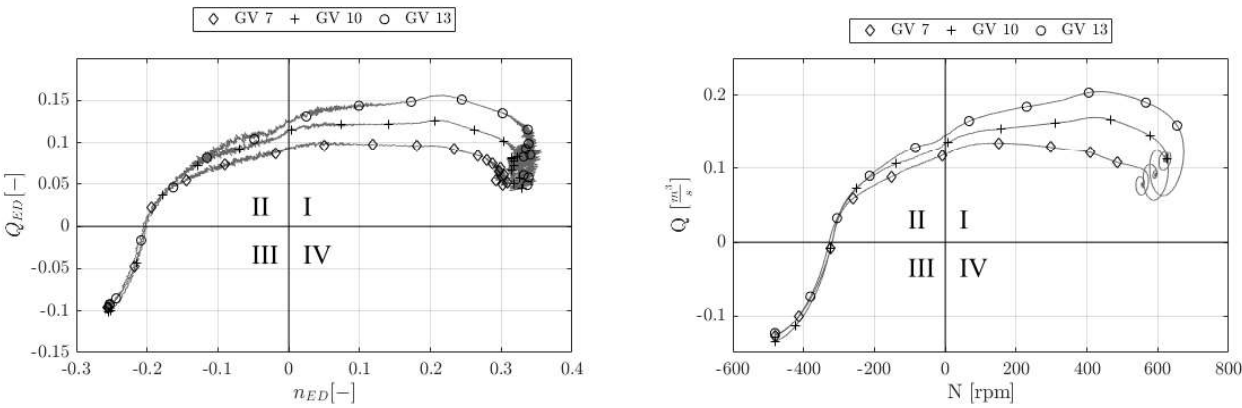
- The flow rate, Q_m at $T_{S\text{-}S}$ and $T_{S\text{-}E}$ were measured by the flow meter.
- The average friction factor for the time period $T_{S\text{-}S}$, $k = \frac{\Delta p_{Ts-s}}{Q^2_{Ts-s}}$
- The friction loss was found at $T_{S\text{-}S}$ and $T_{S\text{-}E}$ using the relationship given in eq. (3).
- $\zeta_{T_{S-s}}$ was used until the rotational speed changed direction, then $\zeta_{T_{S-E}}$ was used for the rest of the measurement.
- ΔQ was calculated by the pressure difference for each time step and $Q_i = Q_{i-1} + \Delta Q_{g_i}$

2.2 Experiment procedure

The procedure consisted of first setting the required head in the upper and lower reservoir. Then the rotational speed in pump mode was adjusted and the system was given time to stabilize. The generator torque was disconnected, causing the hydraulic pressure to force the RPT from pump mode to turbine mode. The transient measurement ended at turbine runaway speed. Measurements were preformed for constant guide vane angle (GV) 7º, 10º and 13º, for each GV the measurement was repeated seven times.

3. Results

3.1 Transient characteristics



(a) (b)

Fig. 4 Transient measurements for three different guide vane openings in non-dimensional (a) and physical (b) units. I is turbine mode, II is dissipation mode, III is pump mode and IV are reverse pump mode.

The four quadrant, transient characteristics for three different guide vane openings are presented in Fig. 4, shown as machine characteristics Fig. 4(a) and system characteristics Fig. 4(b). Machine characteristics have non-dimensional units Q_{ED} and n_{ED} as defined in eq. (4) and eq. (5). The non-dimensional units are independent of both the system around the machine and the dimensions of the machine parameters.

$$Q_{ED} = \frac{Q}{D^2\sqrt{gH}} \tag{4}$$

$$n_{ED} = \frac{nD}{\sqrt{gH}} \tag{5}$$

In Fig. 4(b) the different guide vane openings give a similar flow rate per rotational speed in pump mode (III). When the flow changes direction in quadrant II the effect of the guide vane positions becomes apparent. Finally, in turbine mode of operation the result of an increased guide vane opening can be seen giving an increased spiral around the runaway position.

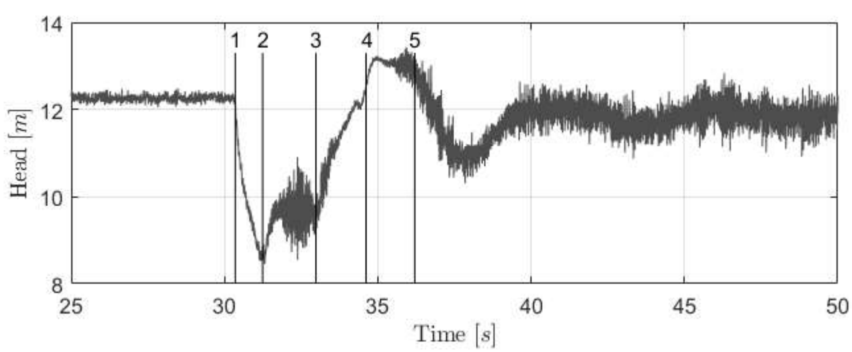
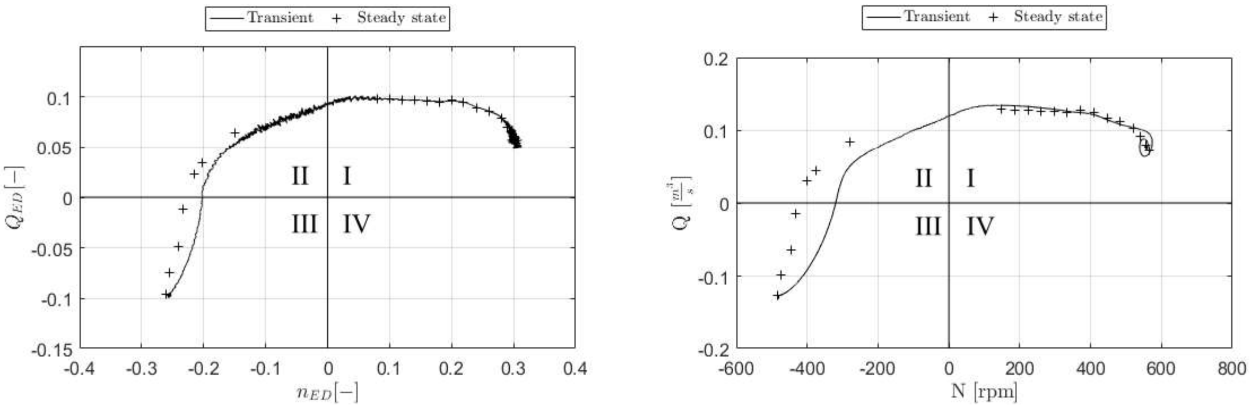


Fig. 5 The change in head (H) during the transient experiment. The numbering show start of T_D (1), flow rate zero (2), rotational speed zero (3), flow rate maximum (4), rotational speed maximum (5).

The pressure oscillations in the system can be seen in Fig. 5 and show the changes in head over time. Since the transient measurements shown in Fig. 4(a) include H, the same oscillations can be seen in this figure. The highest oscillations are found in the dissipation mode in quadrant II and at turbine runaway in quadrant I. All of the guide vane openings show similar magnitude in the oscillations in dissipation mode. At runaway speed the oscillations increase with increasing opening.



(a) (b)

Fig. 6 Transient and steady state characteristic in non-dimensional (a) and physical (b) units

Figure 6 compare the transient measurement with steady state measurements. Figure 6(b) has two areas where the transient and steady state measurements differ significantly, when the rotational speed is negative and around runaway speed. In Fig. 6(a) the difference are concentrated in quadrant II and III.

The difference between the steady state and transient measurements in turbine mode of operation has been addressed by Nielsen [14]. By removing the inertia of the water masses as in eq. (6) when calculating the head, he showed that the transient and steady state characteristics becomes equal around runaway speed.

$$H_t = H - I\frac{dQ}{dt} \qquad (6)$$

$$I = \Sigma\frac{A}{gL}$$

In eq. (6) H is the measured head, I is the hydraulic inertia, dQ/dt is the change in flow rate over time and H_t is the transient head. A and L is the cross-sectional area and the length of the pipe segment respectively.

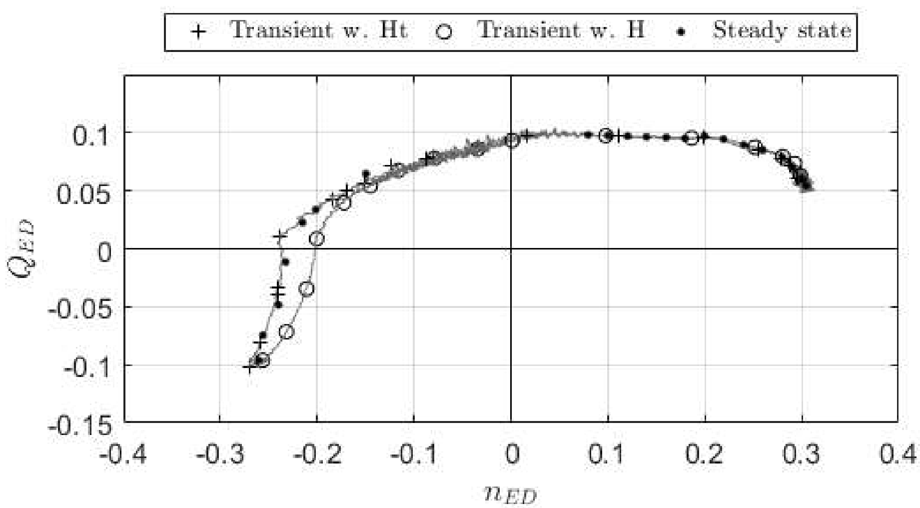


Fig. 7 The difference between the measured transient characteristics with H, the adjusted transient characteristics with H_t and the steady state characteristics.

Figure 7 shows that the difference between the transient and steady state characteristics in quadrant II and III are also removed by Nielsens method. Calculating Q_{ED} and n_{ED} from eq. (4) and eq. (5) for the transient characteristics use the measured H, the adjusted Q_{ED}-n_{ED} characteristics use H_t. The transient characteristics with H_t and its correspondence to the steady state measurements and can thus be explained by the inertia of the water masses.

3.2 Uncertainty

Figure 8 show the repeatability for Q_{ED} vs n_{ED} for four repetitions at 10º guide vane opening.

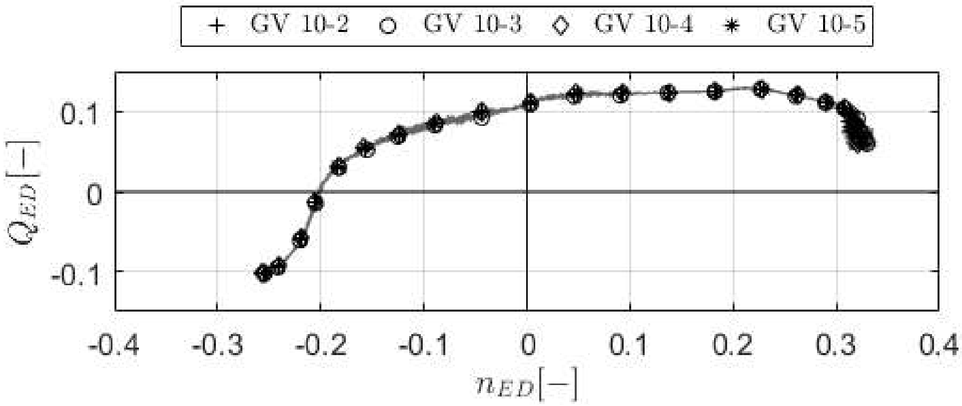
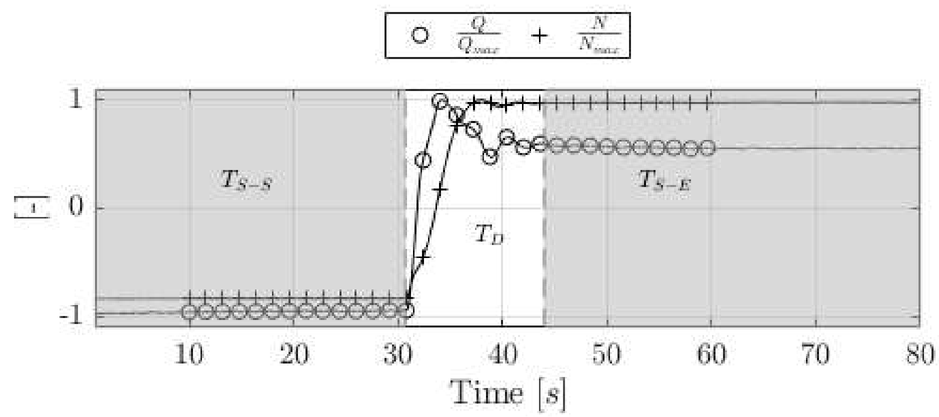
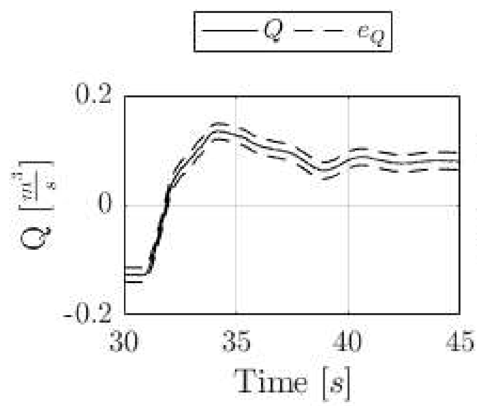


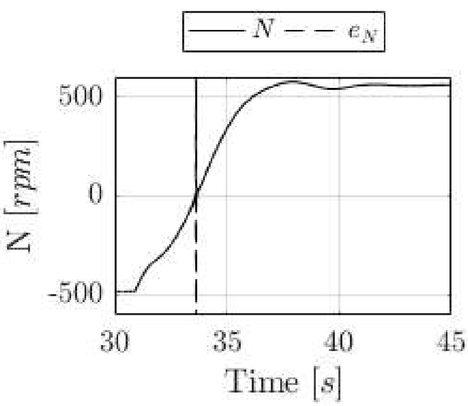
Fig. 8 The repeatability at 10º guide vane opening for measurement series 2 to 5

The absolute uncertainty for N and Q in the transient period T_D for one measurement series can be seen in Fig. 9. Where the uncertainty in the flow rate is shown in Fig. 9(b) and the uncertainty in the rotational speed is shown in Fig. 9(c). It is prudent to note that the uncertainty in flow measured during a transient phase makes it difficult to decide the standard deviation. In this experiment it was decided to use the standard deviation from T_{SS} when calculating the uncertainty in T_D. Due to the chosen standard deviation the uncertainty is probably marginally higher than calculated. For the rotational speed, it is noticeable that the uncertainty increases rapidly around zero. This is due to the encoders dependence on the pulses to determine the rpm. There are 1024 pulses for one rotation, but with close to zero rotation the result is a high increase in the uncertainty around that point. Excluding the uncertainty at zero, the mean absolute uncertainty, e_N for N is 0.0034 rpm. The mean absolute uncertainty, e_Q for the flow rate is 0.000156.



(a)





(b) (c)

Fig. 9 (a) is the time dependent change in Q and N, (b) and (c) show the absolute uncertainty and the real value in the transient period TD for respectively Q and N. The uncertainty band is multiplied with 100 to enhance visibility

4. Discussion

The use of a modified pressure-time method for measuring the flow rate in a transient phenomena with big changes in the flow is previously unpublished. Nielsen [14] and Walseth [12] have used this method in the turbine mode of operation, but to the authors knowledge this is the first time the method is used to measure the transient flow from pump to turbine mode of operation. The uncertainty shown in Fig. 9(b) shows this method to be a reliable way to measure the flow.

There is a significant difference between the steady state and transient measurements in pump and pump break mode. Where, for the same flow, the rotational speed is smaller for the transient measurements than the steady state measurements. This phenomenon was also observed by Liu et al. [10] and Stens [9], but dismissed as an inaccuracy of the CFD in pump mode. The inertia of the water masses explains the difference. Nielsen [14] showed that by taking into account the inertia of the water masses removes the difference between the transient and steady state characteristics. The transient measurements are influenced by the system dynamics.

By accounting for this and subtracting the hydraulic inertia in the system from the head, we get the same results as in a steady state situation without system dynamics.

The pressure fluctuations that can be seen in Fig. 4(a) and Fig. 5 are highest where the discharge is low and at runaway speed in turbine mode of operation. This is in accordance with previous research by Zuo [1] and Tanaka [3], where it was found that these areas give the highest vibrations due to the off-design conditions.

In Fig. 4(b) the flow and rotational speed oscillations at runaway speed is reduced with lower opening degree on the guide vanes. These flow and speed oscillations are caused by the slower head oscillations seen in Fig. 5. The decrease in these oscillations dependent on the decrease in guide vane opening degree is because of the reduced flow through the turbine, and a smaller amplitude of the head oscillations after reaching the turbine quadrant (I). When doing a fast mode change ending at runaway speed it can therefore be recommended to do this in a manner that ends up at a low guide vane angle when the RPT reaches runaway speed. This can be done with a constant low GV angle from pump to turbine mode of operation, or with a gradual reduction in the GV angle during the transition.

5. Conclusion

This article presents an investigation of the transient characteristics of a reversible pump turbine, from pump to turbine mode of operation measured experimentally. It has been shown that the modified pressure time method is a reliable method to measure the discharge in this transient phase. Pressure fluctuations are highest where they can be expected to be, i.e at low discharge in both pump and turbine mode of operation and at runaway speed in turbine. The comparison of the steady state and transient characteristics show a difference in the pump mode due to the inertia of the water masses.

Nomenclature

A	Area	[m]	Q	Flow rate	[m3/s]
B1	Inlet runner height	[m]	p	Pressure	[kPa]
D	Diameter	[m]	α	Guide vane angle	[º]
g	Gravitational acceleration	[m/s2]	β	Mean velocity component	[º]
GV	Guide vane angle	[º]	ρ	Density	[kg/m3]
H	Head	[m]	ς	Friction loss	[-]
I	Hydraulic inertia	[1/s2]	dt	Position indication	[-]
k	Friction factor	[-]	i1	Position indication	[-]
L	Length	[m]	o	Position indication	[-]
M	Torque	[Nm]	vl	Position indication	[-]
N,n	Rotational speed	[rad/s]	1	Inlet position turbine direction	[-]
nED	Non-dimensional speed	[-]	2	Outlet position turbine direction	[-]
QED	Non-dimensional flow	[-]	*	Best operation point	[-]

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