Unstable Characteristics and Rotating Stall in Turbine Brake Operation of Pump-Turbines

Reversible pump-turbines are versatile in the electricity market since they can be switched between pump and turbine operation within a few minutes. The emphasis on the design of the more sensitive pump flow however often leads to stability problems in no load or turbine brake operation. Unstable characteristics can be responsible for hydraulic system oscillations in these operating points. The cause of the unstable characteristics can be found in the blocking effect of either stationary vortex formation or rotating stall. The so-called unstable characteristic in turbine brake operation is defined by the change of sign of the slope of the head curve. This change of sign or “S-shape” can be traced back to flow recirculation and vortex formation within the runner and the vaneless space between runner and guide vanes. When approaching part load from sound turbine flow the vortices initially develop and collapse again. This unsteady vortex formation induces periodical pressure fluctuations. In the turbine brake operation at small guide vane openings the vortices increase in intensity, stabilize and circumferentially block the flow passages. This stationary vortex formation is associated with a total pressure rise over the machine and leads to the slope change of the characteristic. Rotating stall is a flow instability which extends from the runner, the vaneless space to the guide and the stay vanes channels at large guide vane openings. A certain number of channels is blocked (rotating stall cell) while the other channels comprise sound flow. Due to a momentum exchange between rotor and stator at the front and the rear cell boundary, the cell is rotating with subsynchronous frequency of about 60 percent of the rotational speed for the investigated pump-turbine (nω=45). The enforced rotating pressure distributions in the vaneless space lead to large dynamic radial forces on the runner. The mechanisms leading to stationary vortex formation and rotating stall were analyzed with a pump-turbine model by the means of numerical simulations and test rig measurements. It was found that stationary vortex formation and rotating stall have initially the same physical cause, but it depends on the mean convective acceleration within the guide vane channels, whether the vortex formations will rotate or not. Both phenomena lead to an unstable characteristic. [DOI: 10.1115/1.4003874]

Keywords: rotating stall, stationary vortex formation, unsteady vortex formation, S-shape, unstable characteristic, pump-turbine, turbine brake operation, numerical simulation, CFD, measurement, runaway, speed no load, variable boundary conditions, flow separation

1 Introduction

Reversible pump-turbines utilize excess energy available in electrical grids for pump mode operation and produce peak energy in turbine mode at times of high demand making them competitive in today’s deregulated electricity market. For economical reasons, there is an increasing demand for faster switching times between the different operational regimes. Reversible pump-turbines are designed with more emphasis on the pump flow, since the decelerated pump flow is more sensitive to flow separation and losses. This, however, often leads to stability problems and system oscillations near no load (runaway) in turbine operation, an effect which is not observed for Francis machines built only for turbine operation. The machines are synchronized with the electrical grid at speed no load, a procedure which may be slowed down by system oscillations excited by the unstable behavior of the pump-turbine at runaway. No load or runaway is defined as the torque equaling to zero and corresponds to one single point on each curve of the characteristics with constant guide vane opening (GVO). All no load points for each GVO define the runaway line. Synchronization of the machine is done at speed no load where the GVO is small and the rotational frequency times the number of generator pole pairs corresponds to the grid frequency.

In the case of load rejection, the operating range between no load and zero flow rate in turbine operation of pump-turbines (which is called turbine brake) is of eminent importance. Within short time, the operating point moves from full load to no load and may overshoot to turbine brake in the case that the guide vanes and inlet valves do not close [1]. If the characteristic is unstable in this range, low frequency system oscillations will arise, which might damage the penstock (see Martin [2,3]). For larger GVO, additionally rotating stall (RS) with frequencies in the range of 0.3 to 0.7, the rotational frequency will build up in the transient when shutting down the machine.

All manufacturers of pump-turbines encounter this unstable transient operation when starting-up into turbine mode in power plants. Such oscillations are actuated by an unstable characteristic curve at no load. A characteristic is called unstable if the gradient of the head curve in turbine mode near no load operation is negative.

Thus for pump-turbines at no load the criterion for instability is given by \( \frac{dH}{dQ} < 0 \).
If this criterion is fulfilled, oscillations may be excited in the hydraulic system. The slope gives a necessary but not sufficient condition for the instability of the system. Further criteria lie in the elasticity, inertia and the amount of energy dissipated in the system. Once the transferred energy is larger than the dissipated one, the system becomes self-excited (see Staubli [4] and Widmer [5]).

Due to the shape of the $H(Q)$ curve, such an unstable characteristic is called “S-shaped.” In the hydraulic system slow flow rate and head fluctuations are provoked while at the pump-turbine torsional oscillations result at the same frequency making the synchronization difficult (see Dörfler [6]). Recent publications such as Liang [7] or Staubli [8] show that there is a complex interaction of turbine flow and partial pumping flow within the runner channels of the pump-turbine near no load. This interaction is dominated by stationary or unsteady vortex formations in the runner channels and the vanesless space.

Rotating stall (RS) is an instability phenomenon occasionally found in pumps and often found in compressors at part load (see Tsujimoto [9] or Borresen [10]). Lesser known is that the rotating stall also may occur in pump-turbines in turbine flow direction between part load and zero flow rate. Rotating stall is characterized by a number of runner and guide vane channels being stalled (stalled cell) while the other channels comprise sound flow. The stalled cell (or even several cells) is rotating with a sub-synchronous frequency of the runner rotation. Flow rate and pressure pulsations in the guide vane and runner channels are likewise enforced (see Hasmatuchi [11]), causing highly dynamic stresses on the runner and stator vanes. In case of hydraulic machinery, the high radial forces may cause rubbing of the labyrinth seals.

Rotating stall in part load operation of pumps and compressors is often investigated in the literature. According to Gyarmathy [12,13], the propagation of the stalled cells is activated by a momentum exchange between the runner and the diffuser flow at the front and the rear boundary of the cell. The extension of the rotating stall cell increases with decreasing mass flow.

Less information is available on rotating stall in turbine operation of pump-turbines. Veselý [14] and Staubli [15] both measured one cell rotating stall at large GVOs with a relative frequency of 60% of the rotational frequency. In the presented study the unstable characteristics due to stationary vortex formation and rotating stall will be discussed on the basis of numerical flow simulation and test rig measurements of a pump-turbine model.

### 2 Test Facility of the Investigated Pump-Turbine

Under investigation was a reversible pump-turbine model (Fig. 1, left) of a prototype which proved to be unstable during synchronization. The measured characteristics show considerable S-regions (Fig. 8) at low mass flow rates in turbine operation between 6 and 15 degree GVO. Specifications of the machine are:

- $n_s$=45 specific speed (SI units); $N_s$=2324 (US customary units)
- $D_2$=346 mm impeller diameter
- 9 runner blades
- 20 guide vanes
- 10 stay vanes (including tongue)
- 6 degree GVO synchronization speed
- 17 degree GVO best efficiency point (BEP)
- $n$=1000 rpm rotational speed for every investigated operating point

Water at 18°C for the open turbine circuit was delivered from a feeding pump.

In order to separate rotating stall from other unsteady phenomena, 5 high resolving piezo-electric pressure transmitters (GV1…GV5) were placed in the guide vane channels at different circumferential angle and radius. Their natural frequency is at 80 kHz. Linearity is lower than 1% FSO. In addition, the dynamic torque of 2 guide vanes (MLS1 and MLS2) being 144 degrees apart from each other was measured (Fig. 1, right).

The measurement of the operating points near no load and turbine brake depends on the hydraulic system and varies up to 2% from one to a second measurement. Table 1 gives the operating points of the dynamic measurement.

### 3 Boundary Conditions of the Simulations

The transient numerical flow simulations were performed with Ansys CFX 12.0, whereas the hexahedron grid was manually

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**Table 1 Measured operating points**

<table>
<thead>
<tr>
<th>OP</th>
<th>GVO</th>
<th>Q [m³/s]</th>
<th>n [rpm]</th>
<th>kcm [-]</th>
<th>ku [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>OP1m</td>
<td>6 deg</td>
<td>0.038</td>
<td>1000</td>
<td>0.025</td>
<td>1.13</td>
</tr>
<tr>
<td>OP2m</td>
<td>6 deg</td>
<td>0.031</td>
<td>1000</td>
<td>0.020</td>
<td>1.12</td>
</tr>
<tr>
<td>OP3m</td>
<td>35 deg</td>
<td>0.040</td>
<td>1000</td>
<td>0.036</td>
<td>1.51</td>
</tr>
</tbody>
</table>

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Fig. 1 Outline of the investigated pump-turbine with numbered GV channels (left) including the meridian view (middle) and sensor arrangement in the guide vane channels (right)
The mesh comprised 3.5 million tetrahedral elements. The simulations were conducted with 24 cores (3 nodes with 2 Quad Core processors each) of an Intel Xeon X5550 (2.66 GHz) from Sun. Each core had 2 GB DDR3-1333 RAM. Sim 1 and sim 2 each took around one week to reach convergence and 4 to 5 weeks to run the simulation specified in Fig. 3.

4 Unsteady and Stationary Vortex Formations

The flow pattern at speed no load is complex and behaves completely different from that in a best efficiency point (see Liang [7]). The hydraulic energy input to the turbine has to be dissipated to achieve no load on the runner shaft. This is a highly unsteady process. The relative streamlines in the rotor and in the vaneless space between guide vanes and runner in Fig. 4 (left) indicates flow separation, recirculation, and vortex formation in any of the 9 runner channels during operation at speed no load at 6 degree GVO. Partial pump flow back into the vaneless space equals the torque on the runner to no load. At this small GVO the flow through the guide vane channels is throttled and convectively accelerated but not disturbed by flow separation or local backflow. Due to geometrical reasons the vaneless space between runner and guide vanes is considerably larger at small GVOs. In this case the onset of rotating stall, two additional simulations with varying boundary conditions were accomplished:

– In the first simulation (Fig. 3, left and Fig. 8) the mass flow was reduced slightly for each time step. The GVO of 30 degree and speed of 1000 rpm were kept constant. This simulation was started at runaway (90 kg/s) where there is no load on the turbine shaft and rotating stall does not exist. The operating curve was varied by reducing the mass flow by 2.5e-3 kg/s per 1 degree runner revolution down to 15 kg/s.

– The second simulation (Fig. 3, right and Fig. 8) was started at no load (31 kg/s) and 6 degree GVO (OP2s). At this operation the characteristic is unstable (S-shape). Using mesh deformation, the guide vanes were then opened by 0.0014 degree per 1 degree runner revolution and the mass flow was increased simultaneously by 0.558e-3 kg/s per 1 degree runner revolution keeping the speed constant. When 30 degree GVO and 40 kg/s were reached, the simulation was continued for a few revolutions of the runner at constant boundary conditions before being aborted.

The simulations were run with 24 cores (3 nodes with 2 Quad Core processors each) of an Intel Xeon X5550 (2.66 GHz) from Sun. Each core had 2 GB DDR3-1333 RAM. Sim 1 and sim 2 each took around one week to reach convergence and 4 to 5 weeks to run the simulation specified in Fig. 3.

Table 2 Simulated operating points

<table>
<thead>
<tr>
<th>OP</th>
<th>GVO</th>
<th>Q [m³/s]</th>
<th>n [rpm]</th>
<th>kcm [-]</th>
<th>ku [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>OP1s</td>
<td>6 deg</td>
<td>0.038</td>
<td>1000</td>
<td>0.024</td>
<td>1.08</td>
</tr>
<tr>
<td>OP2s</td>
<td>6 deg</td>
<td>0.031</td>
<td>1000</td>
<td>0.020</td>
<td>1.08</td>
</tr>
<tr>
<td>OP3s</td>
<td>35 deg</td>
<td>0.040</td>
<td>1000</td>
<td>0.033</td>
<td>1.41</td>
</tr>
</tbody>
</table>

Fig. 3 Varying boundary conditions during simulation 1 (left) and simulation 2 (right), specifying the regions of rotating stall as well as unsteady and stationary vortex formation

Fig. 2 Grid for numerical computation of the complete domain (left) and zoomed to the rotor and stator (right)
It was found that these pressure fluctuations are associated with unsteady vortex formation in the vaneless space and in the runner channels. The vortices which emerge backflow and obstruct the inflow, develop and decay again and, therefore, induce pressure fluctuations. The frequency of 40 percent the rotational frequency of the runner is distinctly below the rotating stall frequency found at larger GVOs. The frequency of the unsteady vortex formation was dependent on the operating point and varied from 30...50 percent. This unsteady vortex formation could be allocated to the “stable” branch of the characteristics (OP1).

When analyzing simulations performed in the “unstable” range of the characteristics (OP2), it was found that the pressure fluctuations in the guide vanes instantaneously cease and the pressure stabilizes on a constant but higher level. The flow pattern in this operating range at small GVO are stable in space and time and the pressure increases due to the permanent inflow obstruction effected by the vortices.

Thus, at small GVOs, this stationary vortex formation in the runner and the entire circumferential space between runner and guide vanes effects the rising total pressure difference over the machine leading to the unstable characteristic.

5 Rotating Stall in Turbine Brake Operation

In the following, measurement and simulation of rotating stall in turbine brake operation (OP3) at high guide vane opening (35 deg) are compared. Figure 6 illustrates time signals of the 5 pressure sensors (measurement) and probes (simulation) in the guide vanes within 10 runner revolutions. The same running average procedure as documented in Sec. 4 was applied.

One single cell can be concluded from the phase shift between the signals of the sensors mounted circumferentially at different positions. This is visually confirmed in the CFD simulation. In the experiment the rotating stall cell propagates at 59 percent of the runner speed (Fig. 6, right) agreeing well with the simulation where 61 percent were determined (Fig. 6, left).

The simulated amplitude of the pressure fluctuation in the guide vanes due to developed rotating stall reaches up to 40 percent of the turbine head in this operating point. The measurement shows significantly lower amplitudes (5...8 percent of the turbine head). This discrepancy between simulation and measurement might be explained by the different system properties and boundary conditions. Similar results were found when analyzing the dynamic torque on the two guide vanes MLS1 and MLS2.

Signal 1 (GV1) which was recorded before (in the sense of runner rotation) the tongue, shows a remarkable higher amplitude than the other four signals which are taken after the tongue, both in experiment and simulation. Thus, it can be concluded that the tongue represents a barrier where the rotating stall cell is retained shortly and a pressure increase is enforced.
An established rotating stall cell in the guide vanes, post-processed from the simulation results is shown in Fig. 7. Streamlines in the stator point out sections with sound and stalled flow. The contour plot represents the pressure distribution in the runner as well as in the stator.

At the front boundary of the cell, sound stator flow meets stalled runner flow. The physical result is a momentum exchange between stator and runner flow as it is described by Gyarmathy [12]. This momentum exchange decelerates the stator flow and induces a pressure peak at the front boundary (Fig. 7, red area). The opposite takes place at the rear boundary of the stalled cell: Stalled stator flow meets sound runner flow. Hence the stator flow is accelerated and the pressure is depressed. Comparing flow in a pump, the pressure distribution at the front and the rear boundary of the rotating stall cell is opposite.

By analyzing the momentum exchange process, Gyarmathy [12,13], found a dependency of the rotating stall cell propagation on geometric terms uniquely:

\[
\Omega_{RS} = \frac{\omega_{GS}}{\omega_{rot}} = \frac{\sin^2 z}{\sin^2 z + \mu \cdot \sin^2 \beta} \quad (2)
\]

with

\[
\mu = \frac{m_{stat}}{m_{rot}} \quad (3)
\]

Where \( \mu \) means the mass ratio of the fluid contained in the stator channels to the fluid in the rotor channels. Neglecting the vaneless gap between guide vanes and runner and considering the guide vanes and stay vanes as stator channels then the mass ratio of the investigated pump-turbine comes to:

\[
\mu = \frac{9.409 \text{ kg}}{7.527 \text{ kg}} = 1.25
\]

Having the stator blade angle of \( \alpha = 30 \) deg and the rotor blade angle of \( \beta = 21.6 \) deg the propagation of the rotating stall cell can be calculated as \( \Omega_{RS} = 0.596 \). This is in good agreement with the measured (0.59) as well as the simulated (0.61) speed of propagation.

The rotating stall cells block the through-flow entirely on a certain extension of the circumference. Even backflow is observed. In the circumferential average, the blockage also leads to a global pressure rise being the cause for the unstable characteristic at the larger GVO.

6 Transition From Unsteady to Stationary Vortex Formation and Rotating Stall

The simulations with varying boundary conditions, described in Sec. 3, are pointed out with arrows in Fig. 8 (left) which shows the non-dimensional characteristic. Both simulations were started close to the runaway line and extent into the turbine brake operation: simulation 1 at constant large GVO and simulation 2 from small to large GVO. The points represent time averaged measurements of the model pump-turbine. The appearance of the 3 different instability states between rotor and stator – unsteady vortex formation, stationary vortex formation and rotating stall – is schematically highlighted in the characteristic. The results of the simulations are displayed in Fig. 8 (right). The fluctuations of the simulation results indicate time varying head and flow rate. The different slopes in the characteristic of measurement and simulation 1 is caused by stronger rotating stall in the simulation (blocking the trough flow) compared to the experiment, which in turn can be explained with different hydraulic systems (different boundary conditions at inlet and outlet) which play an important role in this operating range (see Staubli [4] and Widmer [5]).

Simulation 1 was started close to the runaway line at 30 degree GVO where unsteady vortex formation is found. Rotating stall establishes with reduced mass flow. Simulation 2 was started near the runaway line of the 6 degree curve where stationary vortex formation is developed in the unstable branch of the characteristic (OP2s). When opening the guide vanes, the vortex formation first
destabilizes and unsteady vortex formation arises. Then, from a specific guide vane angle, rotating stall appears.

The pressure signals in the guide vanes were processed by a joint time frequency analysis (JTFA) which results in a spectrogram showing the time dependent frequency of the signals. The quadratic JTFA algorithm was computed by a short time fourier transform (STFT) which gives the energy density of the signal and depends on the window function. The window length (Hamming) was equal to 10 periods of rotating stall and 97% of the values were overlapped in each window. Figure 9 displays the spectrogram of the pressure signal GV1 for both simulations with constant GVO (left) and with increasing GVO (right). When passing the characteristics through at 30 degree GVO (sim 1), the first sign of rotating stall arises after about 50 revolutions (85 kg/s) which is just below no load. From about 70 revolutions (70 kg/s), rotating stall is fully established and it abates from about 120 revolutions (25 kg/s). The frequency is 10.2 Hz which equates to 61% of the runner speed for all mass flows.

When opening the guide vane angle (sim 2), rotating stall starts from about 50 revolutions (25 degree GVO). At this, the frequency changes from initially 11 Hz at 25 degree to 10.2 Hz at 30 degree GVO. A first sign of rotating stall can be observed already at about 30 revolutions (15 degree GVO) but the fluctuations disappear again after a few revolutions. Computing at 6 degree GVO (till revolution 12), the vortical structure and hence the pressure is stable (stationary vortex formation) which effects the unstable characteristic, as it is described in Sec. 4. As soon as the guide vane angle is larger than 6 degree (after 12 revolutions), the operating point moves above the branch of the unstable characteristic. Now the vortical structure destabilizes and in phase pressure fluctuations are induced (unsteady vortex formation). The frequency of these fluctuations is typically lower than the rotating stall frequency and comes to 8.5 Hz (51 percent of runner revolution speed).

The mass flow through any of the 20 guide vane channels was analyzed and plotted as a function of runner revolutions. When rotating stall is fully established, negative mass flow (dark blue regions in Fig. 10) is observed in the stalled channels while the healthy channels show increased mass flow. The diagonal lines of low and high mass flow in the hill chart of Fig. 10 (detailed view for a better sight) prove rotating stall. As in the JTFA of the pressure already found, the fully established rotating stall extends from 70 kg/s to 25 kg/s (revolution 70 to 120) in simulation 1 (Fig. 10, left) and from 25 degree GVO (revolution 50) in simulation 2 (Fig. 10, right). It can be observed that rotating stall is especially intensive in front of the tongue up to guide vane channel 19 which means the gradient of mass flow change increases near the

Fig. 8 Turbine quadrant with measured characteristics for different guide vane angles and the results of the transient simulation with varying flow rate (sim 1) and with varying GVO and flow rate (sim 2), (left) overview, (right) zoom on turbine brake range

Fig. 9 Spectrogram of the pressure signal GV1 processed by a JTFA during simulation 1 (left) and simulation 2 (right)
tongue. Immediately before the tongue (channel 20) there is less through flow even in the case of sound flow.

Synchronized fluctuations during unsteady or stationary vortex formation effect no mass flow changes through the guide vanes in the simulation since the mass flow is fixed by the boundary conditions. Rotating fluctuations however produce local mass flow fluctuations through the guide vane channels (Fig. 10) while keeping the sum through all channels constant.

The number of stalled channels is rising with decreasing mass flow (decreasing kcm) as it was also proven by Gyarmathy [12]. As the streamlines in Fig. 11 illustrate, there are about 4 guide vane channels (20%) completely stalled at 30 degree GVO and 56 kg/s (left) while at 30 degree and 16 kg/s (right) about 14 guide vane channels (70%) are completely stalled.

To localize the onset of rotating stall and stationary vortex formation in the investigated machine, simulation 2 was performed from the known stationary vortex formation at small guide vane angles (OP2) to the known rotating stall range at large guide vane angles. Figure 12 displays the pressure signals of the probes GV1 (positioned at guide vane outlet) and GV5 (positioned at guide vane inlet) during this simulation with increasing GVO. Obviously, there is a pressure drop through the guide vanes at small GVO (start of simulation) due to flow acceleration and losses which is brought out by the different pressure levels of GV1 and GV5. This pressure decrease between guide vane entry and exit abates and disappears with larger GVO.

After about 30 runner revolutions (15 degree GVO) the pressure levels equalize at the guide vanes inlet and exit. Between 30
and 35 revolutions rotating stall establishes intermittently. Between revolution 10 and 30 synchronized pressure fluctuations due to unsteady vortex formation occur which is pointed up with the phase shift between the two signals being close to zero. On the other hand, the phase shift during rotating stall ranges around \( \pi \) corresponding approximately to geometric terms.

An indispensable condition for the appearance of rotating stall in turbine operation is that the flow in the guide vanes is prone to flow separation. This is the case for very low mass flow operation with pronounced flow separation and recirculation in the runner at larger GVOs. The guide vane channels are prone to flow separation when the negative pressure gradient is absent hence the flow is not accelerated. Typical flow separations in the guide vanes can be seen in Fig. 11, for small and large rotating stall cells.

On the other hand it is evident that stationary vortex formation only occurs if there is enough space between runner and guide vanes in order to have fully established vortices around the inlet edge and the vaneless space (see Fig. 4, right). This is given for small GVOs where the flow is always accelerated through the guide vane channels.

7 Conclusions

Stationary vortex formation and rotating stall may occur in the wide range from part load to turbine brake operation of the unstable branch of a pump-turbine characteristic. Both phenomena can be traced back to flow separation and vortex formation in the runner and the stator due to low mass flow. During unsteady vortex formation, the vortices in the runner and the vaneless space fluctuate in time and induce in-phase pressure fluctuations in the vaneless space. This occurs on the stable branch of the characteristic (Fig. 8).

During stationary vortex formation the disturbed flow pattern enlaces the entire circumference but is limited to the runner and the vaneless space (Fig. 13, left). Since the fluid through the guide vane channels at small GVO is convectively accelerated, the flow will not separate. For small guide vane angles the vaneless space is large enough to allow there the formation of circumferentially extended vortices.

Rotating stall is usually established at larger GVO where the vaneless space becomes small and the negative pressure gradient through the guide vanes lacks. The disturbed flow pattern in this case is limited to several channels (rotating stall cell) but extends from the runner to the guide and stay vane channels and rotates with subsynchronous frequency (Fig. 13, right) due to a momentum exchange between rotor and stator. The simulated rotating stall phenomena could be confirmed by measurement and the estimation based on momentum exchange of the mass in the runner and guide vane channels.

The unstable characteristic can be traced back to two phenomena of identical source, which both increase the total pressure difference over the machine: on the one hand a circumferential stabilization of the vortical structure in the runner channels and the vaneless space leads to a blockage of the through flow (stationary vortex formation) and on the other hand at larger GVO these vortical structures may change to rotating stall cells, which block the through flow only in a certain number of runner, guide and stay vane channels. This increasing pressure is cause for a change of sign of the characteristic being responsible for possible oscillations of the hydraulic system in a hydropower plant. These hydraulic system oscillations complicate or avoid the synchronization to the electrical grid at small GVO and compromise the machine in the case of load rejection at large GVO and large mass flow.

Acknowledgment

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Nomenclature

- \( \text{BEP} \) = best efficiency point
- \( D_2 \) = impeller diameter
- \( \text{GVO} \) = guide vane opening
- \( H \) = head
- \( H_0 \) = mean head at actual operating point
JTFA = joint time frequency analysis

\[ kcm = \frac{4Q}{(D^2 \pi \sqrt{2gH})^{1/2}} \] specific discharge factor

\[ ku = \frac{u_2}{(2gH)^{1/2}} \] specific speed factor

m = mass

\[ N = \text{number of data} \]

\[ n_q = n(Q[\text{m}^3/\text{s}])^{1/2}/((H[\text{m}])^{3/4} \text{ specific speed in SI units} \]

\[ N_s = n(Q[\text{GPM}])^{1/2}/((H[\text{ft}])^{3/4} \text{ specific speed in US customary units} \]

n = rotational speed [rpm]

\[ p_j = \text{running averaged static pressure} \]

\[ p_j = \text{static pressure raw data} \]

Q = volume flow

RS = rotating stall

\[ u_2 = \text{rotational velocity at outer runner diameter} \]

\[ \alpha = \text{guide vane angle} \]

\[ \beta = \text{runner blade angle} \]

\[ \Omega = \text{frequency relative to rotational frequency} \]

\[ \omega = \text{angular frequency} \]

\[ \mu = \text{mass ratio} \]

Subscripts

rot = rotational/rotor

RS = rotating stall

stat = stator

References


