

PREROTATION FLOW ANGULAR FREQUENCY ANALYSES

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Abstract: In this paper the results of the experimental and theoretical analyses of angular frequency of the prerotating flow in the entrance pipe of turbomachines, at different operating regimes are given. The prerotation flow swirl appears far before the impeller blades entrance edge and it is the result of the developed circulation flows in the individual impeller channels that also has an affect on the impeller efficiency.

The mathematical model proposal for the angular flow frequency calculation in the entrance pipes of the model turbomachine (pump-turbine model) is given. The model is based on the determination of the relative circulation flow in the impeller channels that appears as a consequence of the effect of the rotated impeller channels.

The calculation results are compared by experimental data of measured prerotation flow angular frequency by simplified anemometer with six straight blades in the axial pipe direction. The anemometer is placed in the entrance pipe at distance of three-pipe diameters distance before the impeller entrance.

Keywords: radial turbo-machine, prerotation flow measuring, anemometer with straight blades

1 INTRODUCTION

Stewart first examined the prerotation flow at the pump entrance pipe. Stepanoff [1] explained the prerotation flow with entrance flow velocities triangles, considering the theory of the potential flow. Predin [2] examined the causes of direction change of the prerotation flow. Direction of the flow prerotation, compared by impeller rotation direction, is examined at different operating capacities and at different impeller speed. In the present contribution is considered also the proposal of the mathematical model, given by Predin [3], valid for the plain flow and based also on the determined circulation flow at the rotated impeller channels.

The goal of this contribution is to extend the validity of the known mathematical model proposal for wider radial impeller type group, considering also the Nishi [4] cognition for the Francis turbine draft tube. In this flow direction the cavitation process begin at the pump-operating regime at higher operating capacities.

2 MATHEMATICAL MODEL PROPOSAL

The circulation is determined by integrating velocity around the closed curve

$$\Gamma = \oint_L \vec{v} \circ d\vec{l} \quad (1)$$

The relative circulation flow is given as a sum of circulation flow around the impeller blade profile Γ_B and two circulating flows in the impeller channel in radial Γ_{CR} and axial Γ_{CA} direction.

$$\Gamma = \Gamma_B + \Gamma_{CR} + \Gamma_{CA} \quad (2)$$

The angular speed of prerotation swirl is

$$\omega = \frac{\Gamma}{2 \cdot p \cdot r_{imp}} \cdot \frac{f}{r_{imp}} \quad (3)$$

where r_{imp} is the radii of anemometer and f is function that considered the influence of the operating regime and the influence of impeller geometry.

2.1 Circulation flows in the impeller channels

The circulation flow around the impeller blade appears as a consequence of different flow acting at the front and rear side of the blade. The pressure difference Δp is caused by the relative flow that crossed the curved rotated impeller channel and it can be presented as a function of relative flow velocities difference at the entrance diameter w_1 and relative flow velocities at the impeller exit diameter w_2

$$\Delta p = \frac{\rho}{2} \cdot (w_1^2 - w_2^2) \quad (4)$$

The circulation around the blade can be determined by

$$\Gamma_B = \int_L (\vec{w}_s - \vec{w}_t) \circ d\vec{l} \quad (5)$$

Based on the Eckert-Schnel [5] derivations, the relative flow velocities difference at the blade surface can be equalized by the difference of the relative flow velocities at entrance end exit diameter

$$(\vec{w}_s - \vec{w}_t) \equiv (\vec{w}_1 - \vec{w}_2) \quad (6)$$

Considering the Euler main equation of turbomachines the final form of the equation for the circulation flow around the blade calculation can be given as

$$\Gamma_B = 2 \cdot p \cdot (r_2 \cdot c_{2u} - r_1 \cdot c_{1u}) \quad (7)$$

where r_1 is the impeller entrance radii, r_2 is the impeller exit radii, c_{1u} is the absolute flow velocity at the entrance and c_{2u} at the exit impeller diameter, respectively.

The circulation flow in radial direction is a consequence of the flow movement in direction opposite to the impeller rotation direction. For single impeller channel it can be given as

$$\Gamma_{CR} = c_{2u} \cdot t_2 - c_{1u} \cdot t_1 + w_2 \cdot l_{vane} - w_1 \cdot l_{vane} \quad (8)$$

where t_1 is division at the entrance and t_2 at the exit diameter, respectively l_{vane} is blade-curved length.

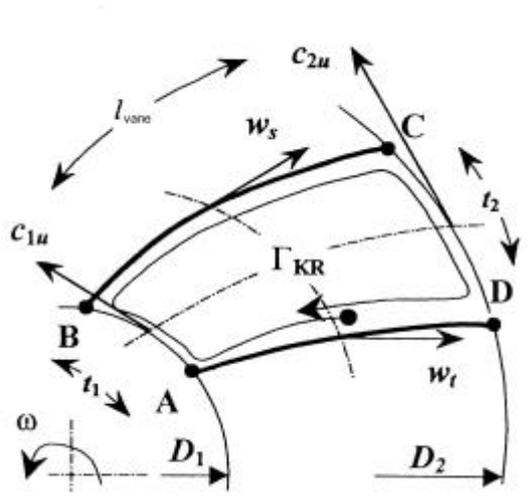


Figure 1. Circulation flows in the impeller channel.

The circulation flow in the impeller width direction appears as a result of the flow inclination at the impeller entrance (almost for 90° degrees by radial impellers). As follows from the measuring results of flow velocities at tip and hub side of the impeller the larger flow velocities at the hub side are determined. The same flow velocity situation is noticed at the exit impeller diameter. Therefore there is always a difference in relative flow velocities:

$$\Delta w_{1,b} = w_{1,hub} - w_{1,tip} \quad (9)$$

and

$$\Delta w_{2,b} = w_{2,hub} - w_{2,tip} \quad (10)$$

where means $w_{1,hub}$ and $w_{1,tip}$ relative flow velocities at hub and tip side of impeller at entrance and $w_{2,hub}$ and $w_{2,tip}$ at exit impeller diameter, respectively. The analogue velocity situation for the flow velocities in the circumferential direction is used. On this base the flowing equation can be written

$$\Gamma_{CA} = \Delta c_{2u,b} \cdot t_2 - \Delta c_{1u,b} \cdot t_1 + \Delta w_{2,b} \cdot l_{vane} - \Delta w_{1,b} \cdot l_{vane} \quad (11)$$

Substituting the equations (7), (8) into equation (2) we get

$$\Gamma = 2 \cdot p \cdot (r_2 \cdot c_{2u} - r_1 \cdot c_{1u}) + (c_{2u} \cdot t_2 - c_{1u} \cdot t_1 + w_2 \cdot l_{vane} - w_1 \cdot l_{vane}) + (\Delta c_{2u,b} \cdot t_2 - \Delta c_{1u,b} \cdot t_1 + \Delta w_{2,b} \cdot l_{vane} - \Delta w_{1,b} \cdot l_{vane}) \quad (12)$$

According to the above-presented equations and by considering the function dependency f the final equation form for the calculation of the circulation flow angular frequency can be given as

$$w = \frac{350 \cdot \Gamma}{2 \cdot p \cdot r_{imp}^2} \cdot \left(-\frac{r_{imp}}{r_3} \cdot \left(\frac{Q}{Q_{opt}} \right)^3 + \frac{l_{vane}}{t_1} \cdot \left(\frac{Q}{Q_{opt}} \right)^2 - \frac{l_{vane}}{b} \cdot \left(\frac{Q}{Q_{opt}} \right) + 1 \right) \quad (13)$$

where Q_{opt} is optimal flow of pump – turbine model.

3 PREROTATION FLOW MEASURING

The angular prerotation flow frequency is measured by using the anemometer with straight blades placed in axial direction of the entrance pipe. The anemometer is placed in the entrance pipe at the distance of three pipe diameters before the impeller entrance (Figure 2).

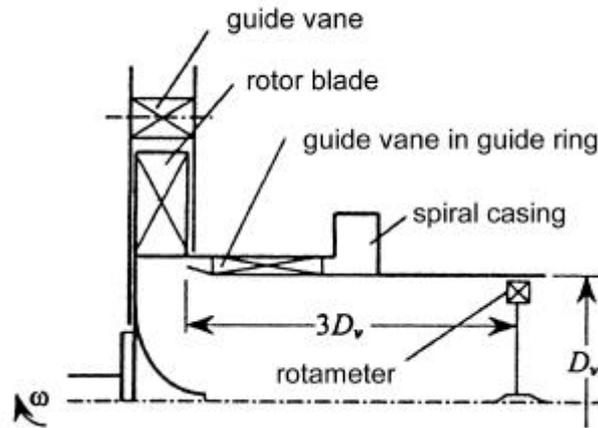


Figure 2. In entrance pipe placed anemometer.

The anemometer is actually the axial impeller with six straight blades made by light plastic mass where the tiny metal plates are placed perpendicularly to the blade as shown at figure 3.



Figure 3. The anemometer photo.

Two inductive sensors type TURCK Bi-M12-AP6X are used for the pulse receiver. The measuring signal was taken by universal counter HP 5325 B, over the measuring unit (Figure 4). The rotation direction of the anemometer impeller is given by the measuring unit where the successive measuring signals from magnetic probes are compared. The angular frequency of the anemometer impeller can be calculated by using equation

$$w = 2 \cdot p \cdot \frac{n_{imp}}{z_{imp}} = 2 \cdot p \cdot \frac{n_{imp}}{6}$$

$$w = \frac{p}{3} \cdot n_{imp} \tag{14}$$

where n_{imp} is the average value of the measured results and z_{imp} is the anemometer impeller blades number.

The flow at the intake pipe is predicted as a stationary flow. The anemometer impeller is put into the entrance pipe in that way that the minimal effect of the flow disturbance is predicted. The impeller blades have the minimal surface area in the axial direction that is in flow direction. The weight of the anemometer impeller is relatively small (less than 0,2 kg) in order to provide a small impeller inertia moment. The impeller bearing is performed by precise bearing boxes at the both sides of the impeller. The phase delay is practically negligible while the rotating speeds are small.

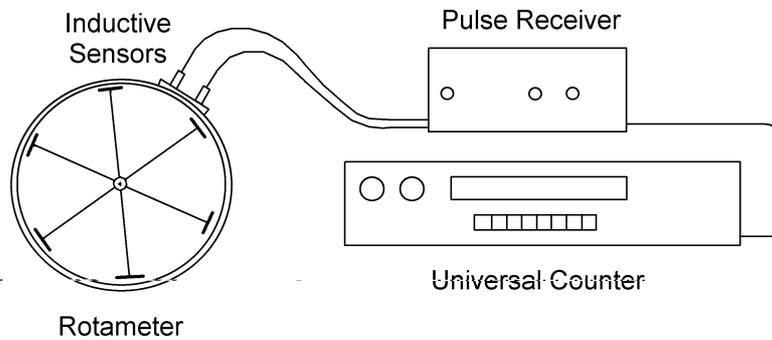


Figure 4. Measuring chain.

3.1 Measuring results

To avoid the measuring errors we take three coincidental measuring values at each measuring and then calculate the average value

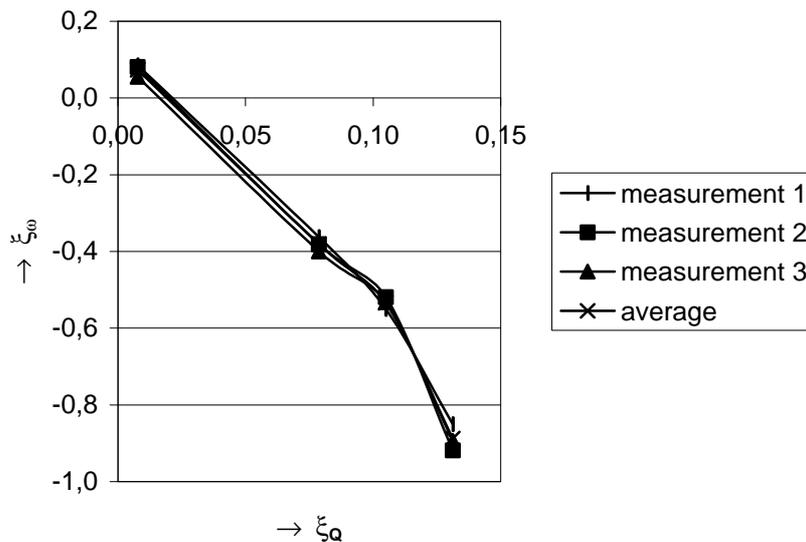


Figure 5. The measuring results.

$$n_{imp} = \frac{1}{3} \cdot \sum_{i=1}^3 n_{imp,i} \quad (15)$$

that is taken as a representative measuring value. At figure 5 the measuring results of the averaged representative values are given in dependency of the operating capacities. The values are given in non-dimensional form, by

$$x_Q = \frac{Q}{n_{imp} \cdot D_2^2} \quad (16)$$

and

$$x_w = \frac{w}{w_{max}} \quad (17)$$

where w_{max} is the maximal measured value of the angular impeller speed. From diagram at figure 5 a small deviation of measuring values is evident.

4 COMPARED EXPERIMENTAL AND CALCULATED RESULTS

Decreased flow velocities of the prerotation flow are considered according to the flow friction decrease with the squared radii of the entrance pipe. The radial impeller imperfectness and the impeller efficiency are also considered.

From the results of measuring and calculation can be concluded, that the course is almost identical. On that base the good agreement between measuring and calculated results can be concluded. Therefore the proposed mathematical model for prerotation flow angular frequency calculation, gives satisfactory results.

The relative larger disagreement is evident at the area of the larger operating capacities, where also larger flow turbulence exists. The proposed mathematical model considered the stationary un-viscous flow. This can be the reason for the calculation error in this operating area.

By rigorous analyses the real flow should be considered.

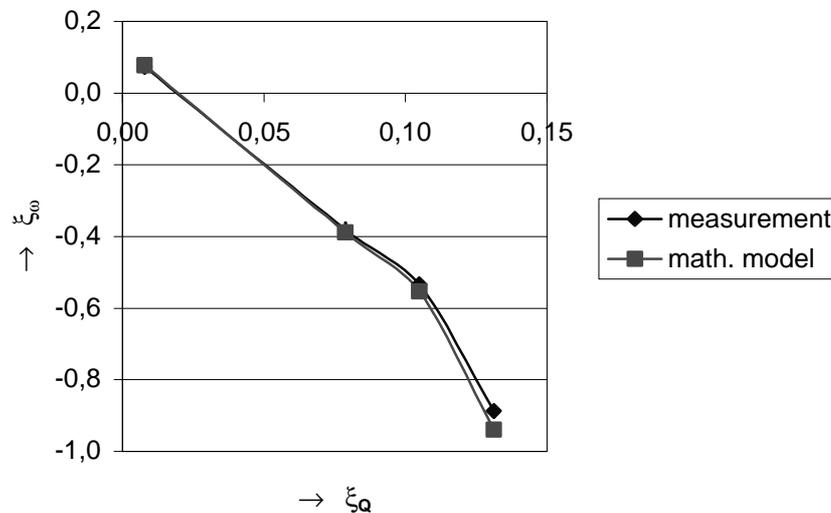


Figure 6. The results comparison.

5 CONCLUSIONS

The stability of measuring results is acceptable. It may be concluded that the chosen measured method is correct. Discrepancy of the measuring results by measuring repetition is small.

The measured results prove that the prerotation flow at the pipe entrance changes direction and the strength in which appears. The strength and direction of the prerotation flow depend directly on the impeller-operating characteristic. Therefore the prerotation flow depends directly on the strength of the circulation flow, which is created in the individual impeller channels. The prerotation flow depends also on the operating pipe characteristic.

The swirl flow at the entrance pipe of radial turbo machines appears far from the impeller entrance diameter or entrance of the impeller channels. At small operating capacities the prerotation flow

direction is the same as the direction of impeller rotation. The cause may be found in the flow friction. With larger capacities the prerotation flow changes its direction into the opposite impeller rotation direction. The reason for this direction change may be the flow diversion in direction of the smallest resistance, what in this case is the opposite direction as the impeller rotating direction is.

The mathematical model proposal gives good results and can be used for the first approximation of the prerotation flow determination for the group of impeller types with specific speeds between 20 and 35 min^{-1} , but for the general model application more experimental tests must be done.

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