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A New Concept of Hydraulic Design of Water Turbine Runners

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Abstract

Vibrations at different frequencies with a different intensity as well as a pressure pulsation with different parameters are two phenomena which can be observed at different water turbines. Due to the vibration and the pressure pulsation some restrictions of water turbine operation range are applied. Similar problems with the efficiency level in a wide water turbine operation range are the basic problems which are solved for ages. A theoretical and practical solution of the above mentioned problems is very much time and money consuming. The paper describes a new theoretical solution of the excitation and pressure pulsation decrease as well as extension of the operational range with high efficiency level. The new concept to decrease the vibrations and pressure pulsations is based on a heterogeneous runner blade geometry generation. The new concept of the runner geometry design was numerically tested at a low specific speed pump turbine, see Fig. 1, and basic points of the concept are presented in this paper.

Keywords: Pressure pulsation, Detuning, Heterogeneous blade passages.

1. Introduction

The vibration and pressure pulsation occurrence depends on different reason. One of the most important reasons is the rotor stator interaction (RSI). Under homogenous spacing of runner blades as well as under homogeneous runner channels one sharp peak in the frequency spectra of the excitation predominates. Under heterogeneous spacing of runner blades as well as under heterogeneous runner channels the frequency spectra of the excitation could be less dangerous.

Similar problems appear with operation range with high efficiency level. In the case of the homogeneous runner blades geometry the turbine efficiency for operation points relatively far from best efficiency point decreases considerably. The main reason of the situation is probably in the same flow through each channel of the runner. Therefore an attempt was carried out to test heterogeneous runner blades.

2. Basic principle of the heterogeneous runner design

The basic idea of the presented concept of the water turbines is the application of a heterogeneous shape of the runner blades in the turbine runner.

2.1 Hydraulic design limits

Hydraulic design with heterogeneous blades has been numerically tested in a few steps. Excitation frequencies in vane less area as well as runner natural frequencies and stresses were determinated. There were defined a few limits of the new hydraulic design applied for low specific speed pump turbine: The numerical simulation has been done for prototype of pump turbine with following parameters: H=550m, n=428.6 r.p.m, runner diameter 4.5m. The turbine has 9 runner blades and 20 guide vanes.

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a) Excitation frequencies

Classical hydraulic design allows to apply well known equations: $N.z_g \pm k = M.z_r$, and $f_g = n.z_r.M$; $f_r = n.z_g.N$, where k is number of diametrical nodes; N, M are arbitrary integer; z_g is number of guide vanes; z_r is number of runner blades, n is rotating speed of the runner and f_g , f_r are excitation frequencies of the distributor and

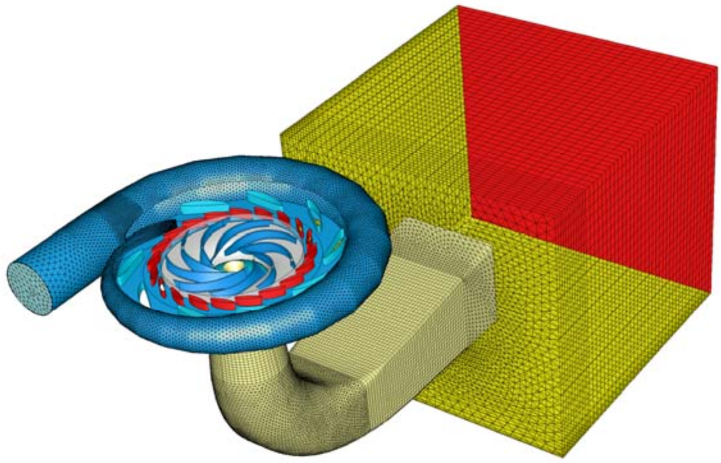


Fig. 1 Low specific speed pump turbine with solution grid

runner, Lit.[1-7]. In the case of reasonable differences of leading edges spacing at the study runner there was found a lot of additional excitation frequencies acting to the runner and distributor. On the other hand the in case of small differences of leadings edges spacing the excitation frequencies are very similar to frequencies with constant of leadings edges spacing. The high number of excitation frequencies appears as serious limitation factor for new runner design.

b) Natural frequencies of the runner

It is commonly known that cracks at pump turbine runners appear at runner hubs and cracks start near a leading edge. These cracks appearance is a consequence of the runner hub vibration. Due to this fact it is very important to know natural frequencies of the runner as well as excitation frequencies in vane les area. Natural frequencies of the runner with different leadings edges spacing were determinated using software Unigraphics, Lit. [8].

There were tested two runners:

– homogeneous runner where unified blades shape and constant blade spacing is applied
– heterogeneous runner where unified blades shape and different blade spacing is applied

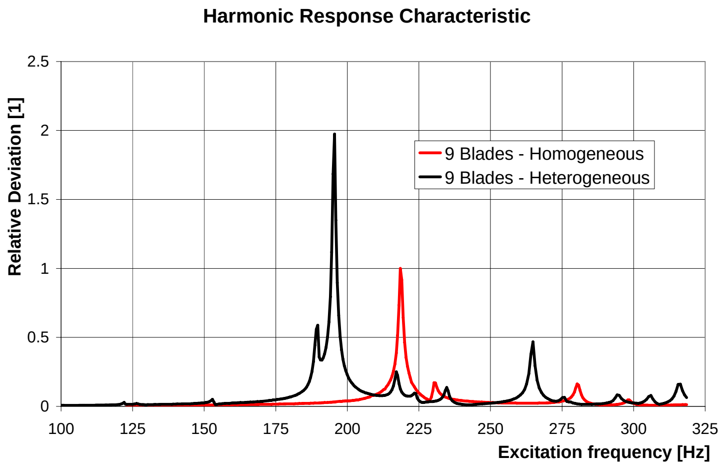


Fig. 2 Harmonic Response Characteristic for 9 blades runners

Results of these numerical simulations are shown in Fig. 2. To save computation time the simulations has been carried out for runners placed in vacuum. At horizontal axis excitation frequencies and at vertical axis relative deviations of the runner hub are shown. Maximal relative deviations for heterogeneous runner are twice higher than maximal deviation for homogenous runner. The number of natural frequencies of the heterogeneous runner is relatively high. Problems with number of natural frequencies as well as runner hub relative deformation size are serious problems for new runners design limitation.

2.2 Hydraulic design limits - summarization

It was found that different runner blade spacing is not applicable at the turbine leading edge area without a serious risk. The reason is that there could be expected resonance at some runner’s natural frequency. On the other hand different blade shape and different blade spacing at turbine trailing edge has not any serious impact on dynamic features of the pump turbine runner. In this case there are not any problems with excitation frequencies as well as with natural frequencies of the heterogeneous runner. These facts were taken in account and these restrictions were applied to the new hydraulic design. Numerical tests, it means CFD (Computational Fluid Dynamic) as well as FEM (Finite Element Method) analyses were carried out for runner with above mentioned restriction application. Outside these restrictions requirements to the runner balancing were taken into account. With respect to that the new runner has 9 runner blades. It was decided to create it from tree groups of different blades – see Fig 3. Marking A,B and C means different blade shape.

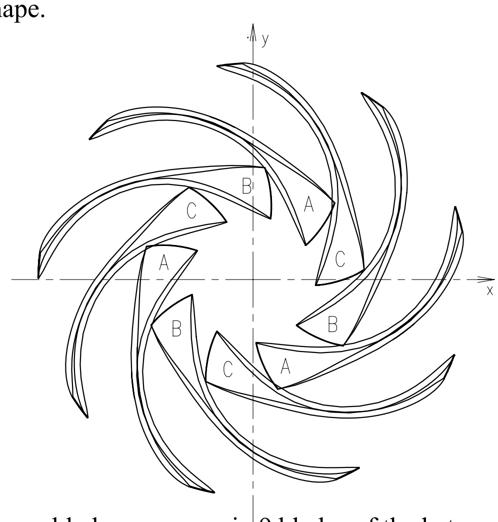


Fig. 3 Runner blade sequences in 9 blades of the heterogeneous runner

The most important and suitable place for the pressure field detuning is the turbine trailing edge area. There are many possibilities for the trailing edge modification to receive detuned pressure field. Due to the different blades shape at the turbine trailing edge, the pressure field has a much more complicated structure. It is the main reason why the necessary energy for the vibration origin is not available. This complicated pressure field reduces the excitation because there is a different portion of energy at the appropriate frequency than at runners prepared by classical design with the same geometry of each blade. Detuned pressure and velocity field at turbine runner output has a positive impact to the efficiency curves of the new runner. The practical solution of the problem is tested at 9 runner blades runner. To this pump turbine runner with three groups of different blade shape was applied and results of numerical simulations are shown in next chapters.

3. Numerical simulation

3.1 FEM analyses

The runner was modelled of three different runner blade shapes. This means that three groups of three runner blade shapes were used for the runner design. Each section of three blades has the same geometry. These blades were installed on the runner from each group alternately. The regular blade spacing at the turbine leading edge was chosen because the mode shape of the runner hub near the leading edge should be the same for each blade channel. The application of the above mentioned system to the runner blades positioning means that the balancing of the runner is kept. The geometry and grid applied for the new runner in the numerical model is shown in Fig. 4. These analyses were carried out for runner and appropriate part of the shaft.

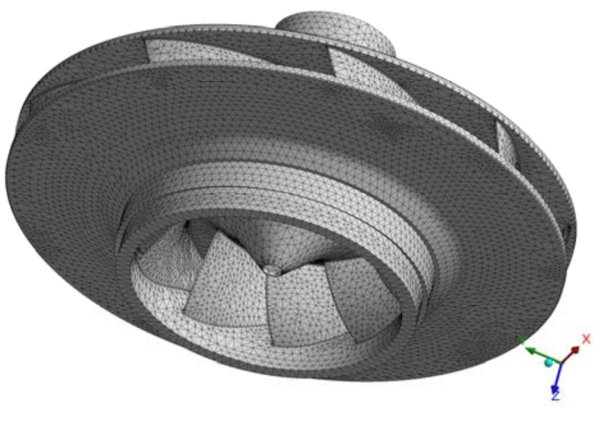


Fig. 4 Grid and geometry for structural analyses

3.1.1 Natural frequencies and mode shape

Comparison of resonance peaks of the homogeneous runner and the heterogeneous one is shown in Fig. 5. An excitation without any diametrical nodes was applied on the hub as well as on crown periphery. The decrease of natural frequencies due to water added mass was taken into consideration. It was found that the maximal relative deviation for the heterogeneous runner is a bit smaller than that for the homogeneous one. On the other hand there appear new resonance peaks.

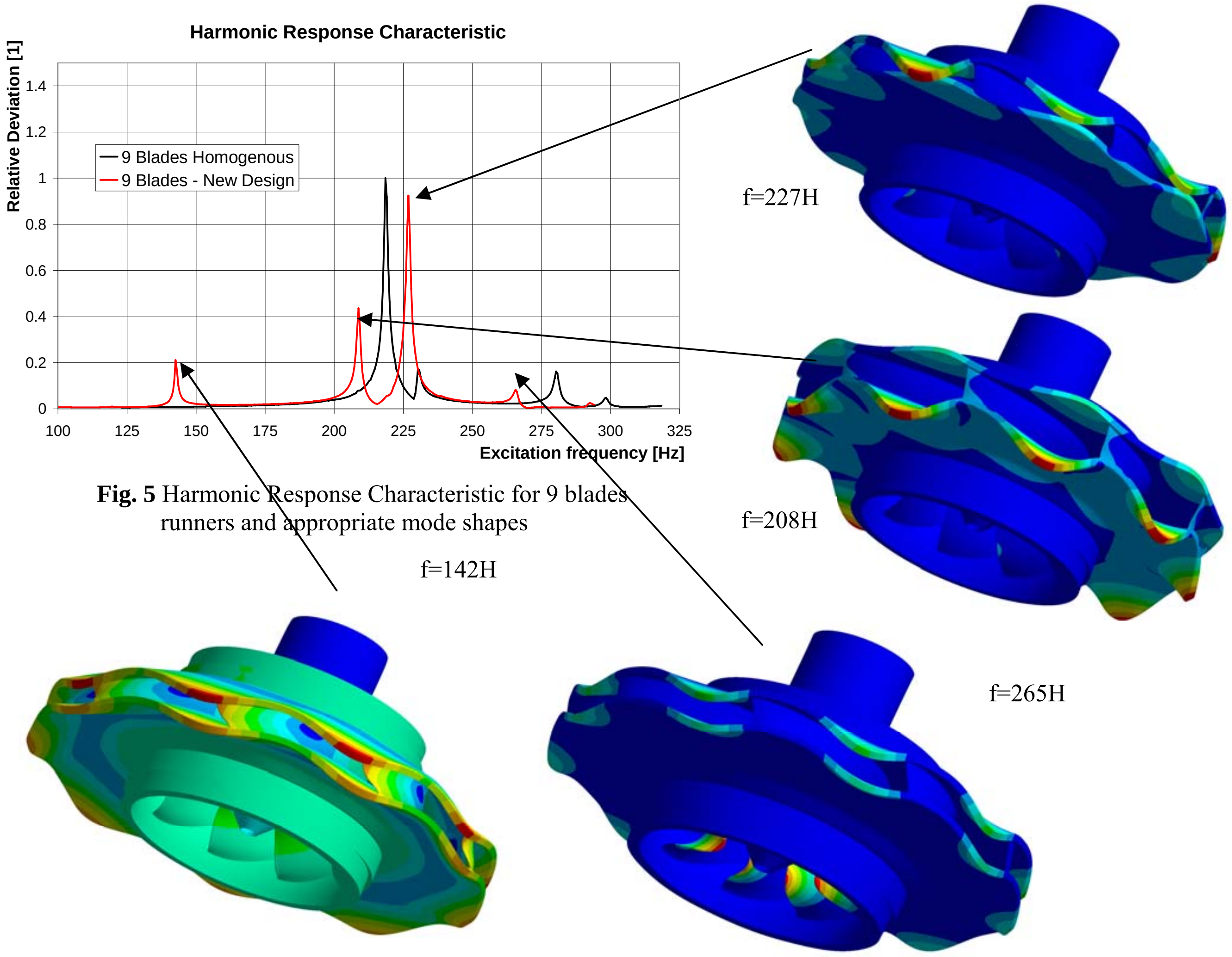


Fig. 5 Harmonic Response Characteristic for 9 blades runners and appropriate mode shapes

The really runner is excited by the pressure field generated by RSI. Its shape is given by number of diametrical nodes, it means by k. The basic excitation frequency in the relative runner coordinate system is $f = n \cdot z_g = 142.85 Hz$.

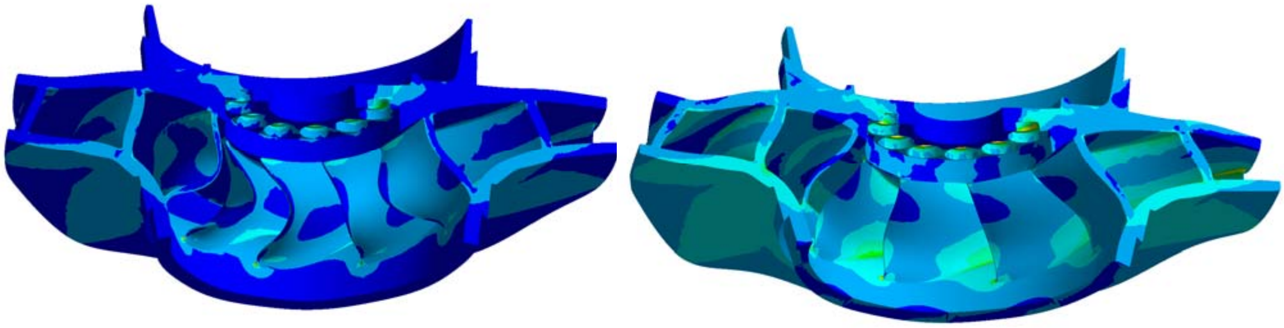
The dangerous vibrations can appear under the following conditions:
The excitation frequency is equal to a natural frequency (resonance)
The excitation shape is similar to the corresponding mode (affinity)
Based on these conditions the peaks in Fig. 5 can be evaluated from the point of the real situation at the turbine in question.

- 142 Hz – the frequency practically equals to the real excitation frequency 142.85 Hz. However, the runner oscillates above all in the axial direction. Such type of excitation (k = 0) generated by RSI cannot be expected under this frequency $(z_r \cdot N = z_g \cdot M)$.
- 208 Hz and 227 Hz – such mode shapes can be excited by RSI. However, there is a sufficient difference between these frequencies and the frequency 142.85 Hz.
- 265 Hz – the maximal deformations are observed at turbine trailing edges. This shape is practically the same for any blades used. In this case, there is also a sufficient difference between this frequency and the frequency 142.85 Hz.

It can be concluded that the mechanical design of the heterogeneous runner is comparable to the homogeneous one.

3.1.2 Stress analyses

There were provided two variants of the stress analyses for maximal power and for runaway speed. Static pressure fields as a loading field of the runners were taken from CFD analyses. This analysis was carried out with the same grid which is shown in Fig. 4.



Maximal power Runaway speed

Fig. 6 Results of strength calculation

Maximal difference of the stresses for different blade shape for maximal power is up to 1% of the maximal stress and up to 3% for runaway speed stresses. It was found that the stress of the new designed runner corresponds with that at homogenous runner.

3.2 CFD analyses

The numerical simulations of the flow through the turbine were carried out by means of CFD Fluent commercial package, see Lit [8]. These numerical simulations have been carried out with model parameters and size. The whole flow profile of the turbine was modelled by CFD for the pressure field evaluation. For the efficiency evaluation the rotating symmetry in the runner was used because of the processor time saving. The grid with three different runner channels was used for evaluation of the efficiency curves. Dynamical behaviour of the new design was numerically tested with complete grid without application of the rotation symmetry. A lot of different trailing edge shapes were numerically tested and the most suitable variant for the pressure field detuning as well as efficiency level was chosen for detail numerical tests.

Following parameters for describing of the received results are used:

Unit speed: $n_{11} = \frac{n \cdot D}{\sqrt{H}}$ and unit discharge: $Q_{11} = \frac{Q}{D^2 \cdot \sqrt{H}}$.

Where: H turbine net head [m]

n rotation speed [min^{-1}]

Q discharge [$\mathrm{m}^3\mathrm{s}^{-1}$]

a_0 wicket gate opening [%]

Each parameter shown in following text, except efficiency, is shown as a relative parameter according to prototype parameters. Efficiency curves published in the paper are shown as a relative efficiency regarding to turbine, respective pump best efficiency point received from the model test of the 9 blades pump turbine with homogeneous blade passages.

3.2.1 Pressure fields at turbine trailing edge

U The pressure fields at turbine trailing edges obtained from the numerical simulations for the new pump turbine designs are presented in the following pictures. Different pressure distribution at trailing edge for each blade is a source of the pressure and velocity fields detuning.

The static pressure distribution for optimal turbine operation point is shown in Fig. 7.

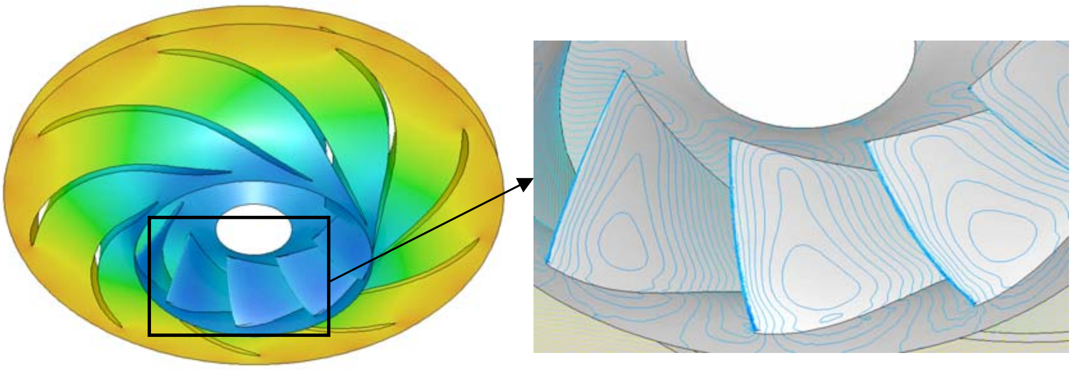


Fig. 7 Static pressure at turbine trailing edge – turbine mode – optimum parameters

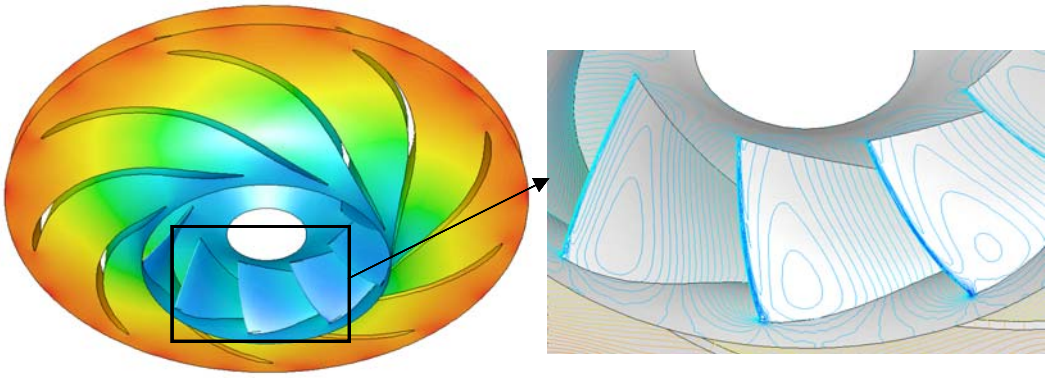


Fig. 8 Static pressure at turbine trailing edge – pump mode – optimum parameters

The static pressure distribution for optimal pump operation point is shown in Fig. 8. Different blades geometry causes different pressure distribution at turbine trailing edge in turbine as well as in pump operation mode. The pressure field near pump turbine runner is due to above mentioned pressure distribution more complicated. On the other hand the hydraulic design of the heterogeneous blade passage has been prepared to the pressure distribution at pump leading edge corresponds with cavitations requirements.

3.2.2 Model test and CFD experiment comparison

Model test and CFD experiment for homogenous 9 blades runner has been carried out. Efficiency level for three guide vane openings for turbine mode and for optimal guide vane opening for pump mode is compared with efficiency curves received for the new design. These comparisons are shown in the next picture. Expected efficiency curves were prepared by CFD simulations after that the model test at hydraulic laboratory was carried out. Comparison of these curves is essential for CFD analyses verification. Numerical simulation for both – homogeneous and heterogeneous blade passage has been done with application the same boundary conditions and similar grid. In similar cases the relative comparison of the received results CFD is a standard method. It means that expected results at the model test, respective at the prototype could be taken with application of relative modification between these two CFD simulations to model test results.

The new design shows efficiency increase in whole operation range for turbine as well as for pump operation mode.

For small wicket gate opening a0/a0opt=0.6 and optimal unit speed in turbine mode efficiency increase is received by numerical simulation approximately 2%. For smaller unit speed (higher head) the efficiency increase is reasonable up to 84 % of the prototype unit speed, see Fig. 9.

For optimal wicket gate opening a0/a0opt=1.0 and optimal unit speed in turbine mode efficiency increase is approximately by 1%. For higher unit speed (smaller head) the efficiency increase is reasonable up to 90 % of the prototype unit speed. For smaller unit speed (higher head) the efficiency increase is more than 1% for any measured and numerically simulated unit speed, see Fig.9.

In the case of wicket gate opening a0/a0opt=1.4 and optimal unit speed in turbine mode efficiency increase is approximately 0.5%. For higher unit speed (smaller head) the efficiency increase is reasonable and more than 1% for any measured and numerically simulated unit speed, see Fig. 9.

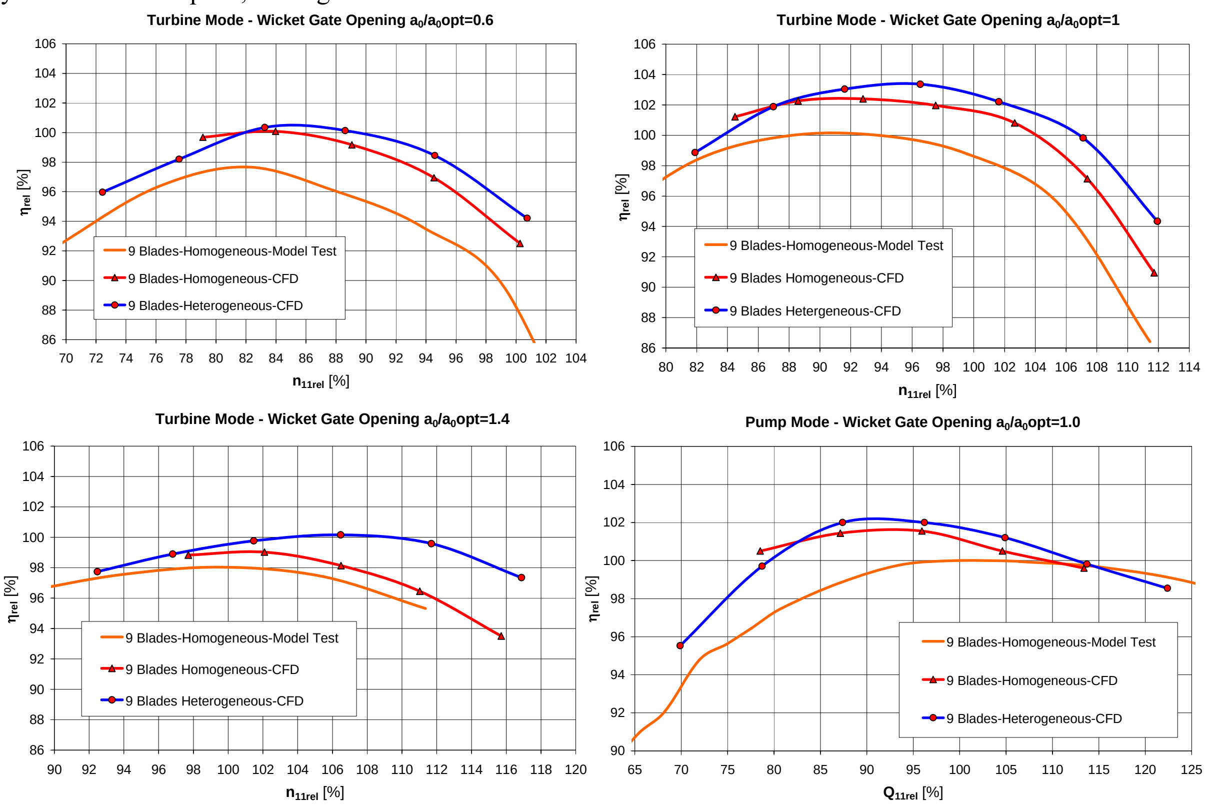


Fig. 9 Efficiency curves for homogeneous and heterogeneous runner design

In turbine mode it is expected that for small discharge (small wicket gate opening) the net head increases and contrary for higher wicket gate opening the net head decreases with the discharge increase. The new heterogeneous blade passage design and its hydraulic features correspond with these facts especially for higher wicket gate opening where the efficiency level increases for net head decrease.

For the pump mode expected efficiency level in a relative wide operation range from optimal discharge higher is by approximately 0.5%.

3.2.3 Hydraulic loses analyses

For above mentioned wicket gate openings hydraulic losses in pump turbine with heterogeneous blade passage are evaluated. Determination of losses is provided for complete spiral case and draft tube geometry. In the runner conditions of rotating periodicity are applied. Complete spiral case means range from inlet to spiral case to inlet to the runner. The efficiency levels and energy losses described in this paragraph correspond with energy in the turbine model size received by CFD simulation. Curves in the Fig. 10 describe energy losses for above mentioned turbine parts.

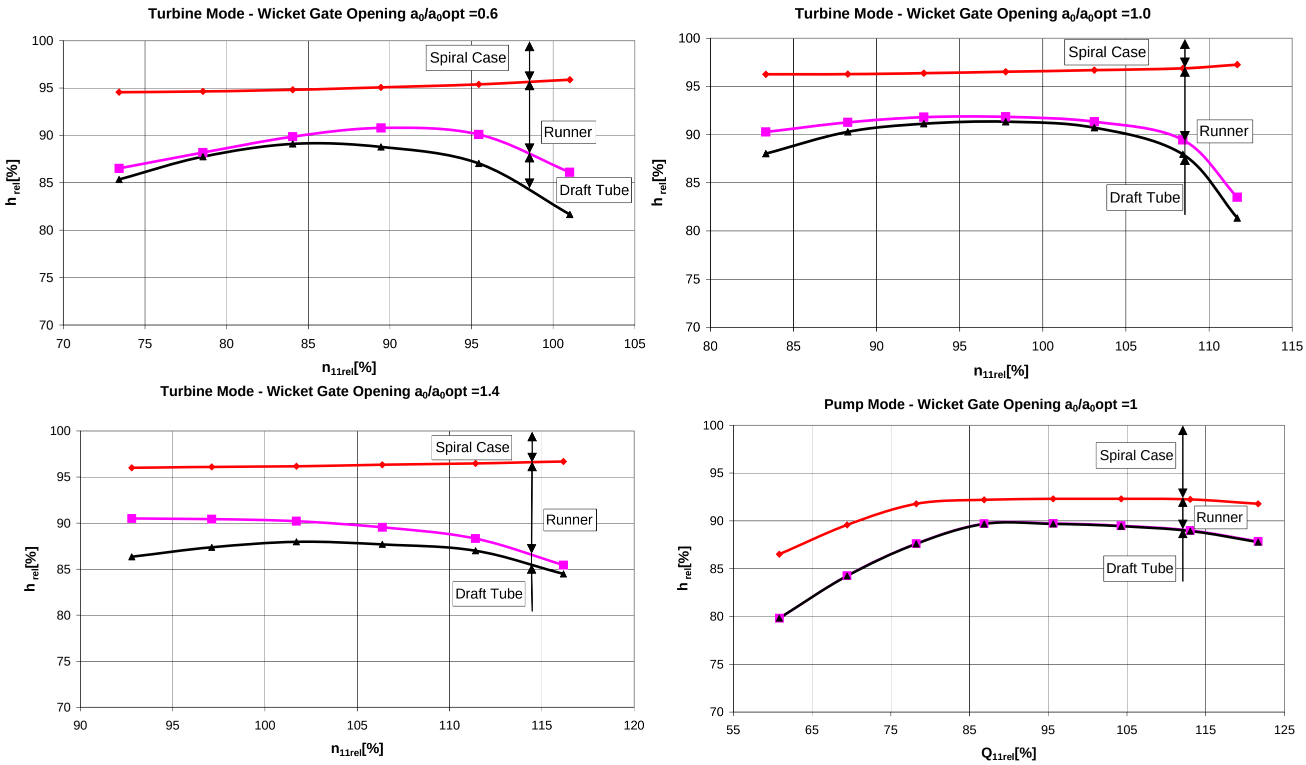


Fig. 10 Hydraulic losses in pump turbine

The portion of energy losses in complete spiral case and turbine mode is maximal for small relative wicket gate opening (0.6) and it is around 5% of energy. Minimal losses in the runner for this wicket gate opening correspond with 90% of the relative unit speed. Energy losses in draft tube rises with relative unit speed, see Fig. 10. For optimal wicket gate opening in turbine mode energy losses in spiral case are minimal (about 4%). Losses in the runner are for this opening relative low in a wide operation range. Losses in draft tube are small due to optimized velocity profiles at the runner outlet. For the maximal wicket gate opening losses in spiral case are around 4.5%. Losses in the runner rise with relative unit speed from 6% to 11% at maximal relative unit speed. Losses in draft tube decrease with relative unit speed from 5% at minimal to 1% at maximal unit speed.

In the pump mode energy losses in spiral case are higher than in turbine mode. It is a consequence of the deceleration flow in spiral case for pump mode. On the other hand losses in the draft tube in case of pump mode are very low and it could be neglected. Distribution of energy losses in the runner has not strong minimum but, for discharges smaller than 85% of the relative unit discharge, energy losses rise considerably. In the pump mode the turbine operates practically in one operation point which corresponds with 100% of the relative unit discharge and efficiency of the machine in pump mode is maximal.

3.2.4 Pressure pulsations analyses

Pressure pulsations were evaluated at surfaces S0, S1 and S2. Surface S0 corresponds with inlet to the spiral case, S1 is in the middle of the wane less area and S2 is at the draft tube inlet, see Fig 11. At surfaces S0 and S2 time path of the static pressure as an integral value was carried out and at surface S1 some representative point for describing of the static pressure behaviour were chosen. There is provided comparison of the homogeneous and heterogeneous blade passage in optimal operating point in pump as well as in turbine operation mode. Results of these analyses are shown in the next figures. Horizontal axis shows ratio of frequency of static pressure and rotation frequency (f/fn) of the runner. Vertical axis shows amplitude (A-ps) of the pressure at appropriate frequency.

Results of the frequency analyses for turbine mode at surfaces S1 and S2 are shown in Fig. 12.

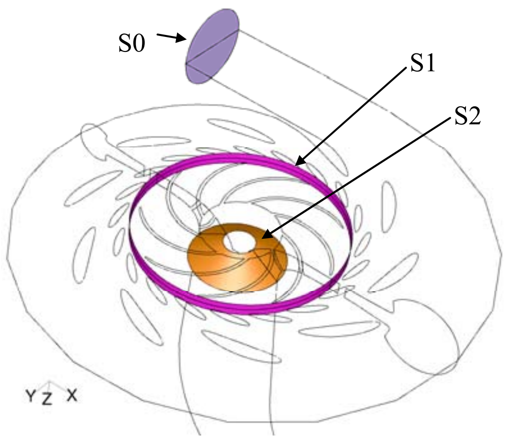


Fig. 11 Surfaces for pressure pulsation evaluation

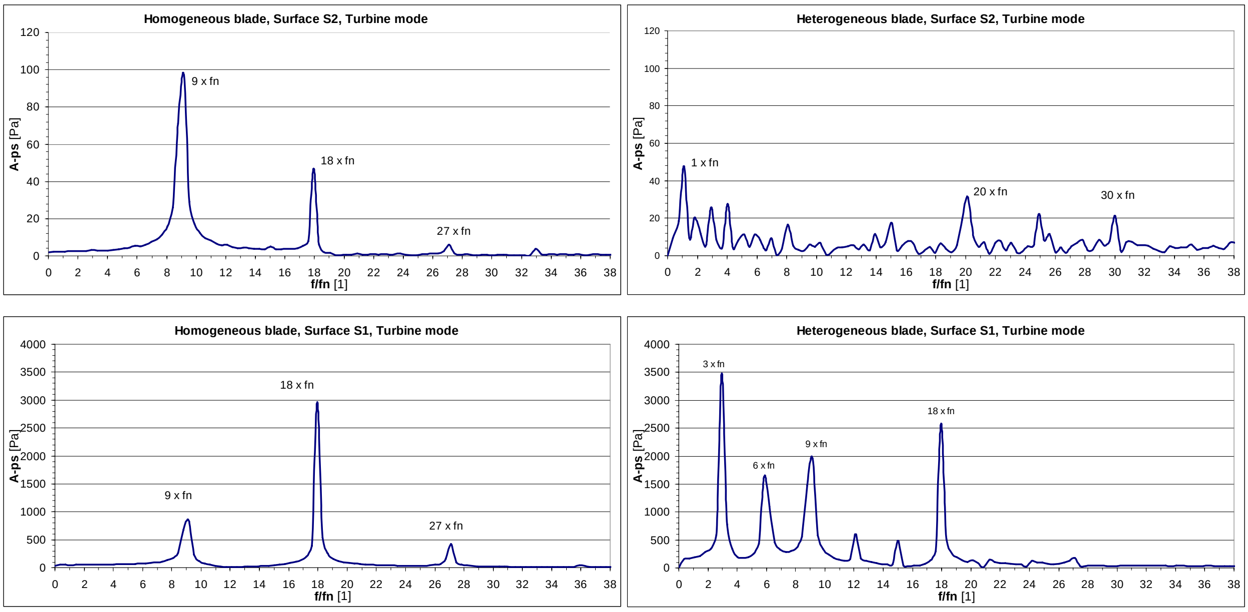


Fig. 12 Pressure pulsations in turbine mode

In turbine mode and homogeneous runner the behaviour of the pressure pulsations in wane less area (surface S1) and draft tube inlet (surface S2) corresponds with theoretical solutions, Lit. [7]. Different situation was found for heterogeneous runner. At surface S2 pressure pulsations completely are detuned. There were found maximal static pressure amplitudes for different frequencies than at the homogeneous runner, but the values of amplitudes of the static pressure are very low at this surface. In wane less area some additional frequencies f=3fn and 6fn were found. At higher frequencies f=18fn and 27fn the amplitude of the static pressure are declined. It means that the energy of the pressure field in vane less area was shifted to lower frequencies.

Results of the frequency analyses for turbine mode at surfaces S0 and S1 are shown in Fig. 13. Surface S0 corresponds with spiral case inlet. At surface S0 the static pressure field is detuned and there was found relatively high pressure amplitude for frequency f=3fn. In wane less area (surface S1) for optimal operating point amplitudes of the static pressure for heterogeneous runner at the level 80% of the amplitudes are received from results of numerical simulation realized for homogeneous runner.

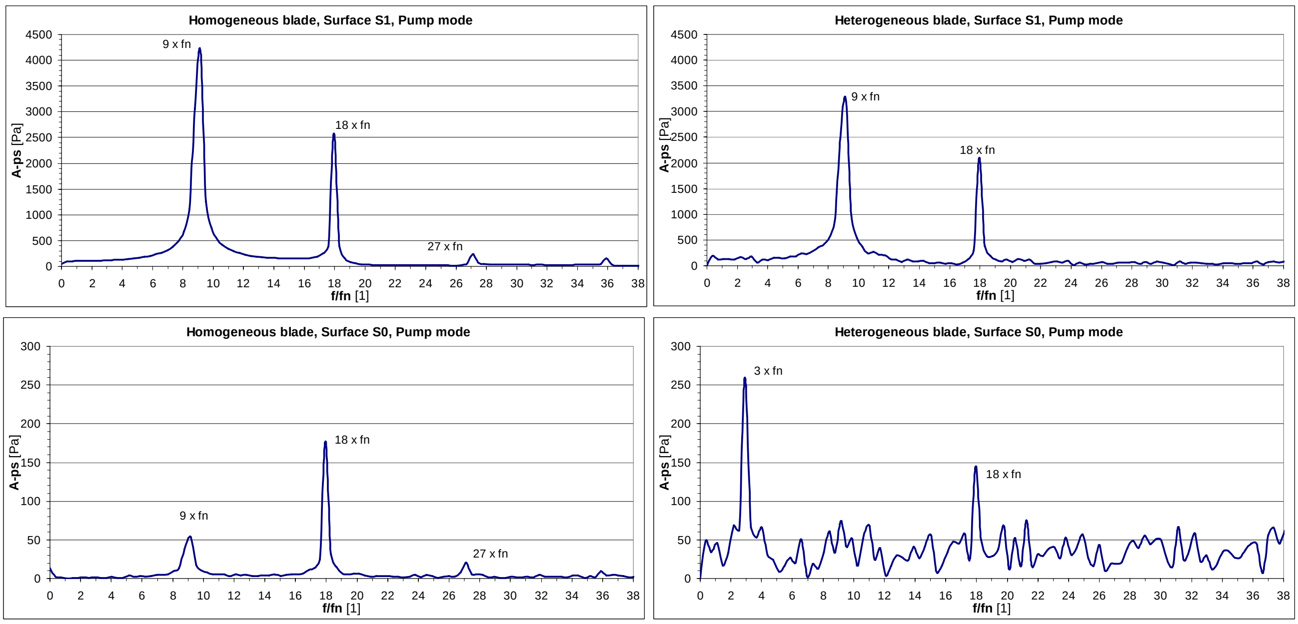


Fig. 13 Pressure pulsations in pump mode

Acknowledgments

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Conclusion

The heterogeneous blade passage concept of water turbine runners was applied on a low specific speed pump turbine runner. The results of this concept were received from numerical experiments. It was found out that the efficiency level is very high not only for the best efficiency point, but it is high in a wide operational range in the pump mode as well as in the turbine mode.

Basic results received from the numerical simulations of the heterogeneous pump turbine runner are as follows:

- Mechanical design is comparable to classical design with homogenous blade passages.
- Efficiency level of the runner is high.
- Pump turbine can be operated in wide operating range with higher efficiency level.
- Pressure pulsations in pump turbine are impressed.
- Pressure field at inlet and outlet of the turbine is detuned.
- Pressure field in wane less area is shifted to smaller frequencies
- The concept is suitable for pump turbines which operate with high unit speed

Hydraulic design of the heterogeneous blade passage has not been finished yet. During solution of the problem of heterogeneous blade passage a lot of addition questions appeared and these questions will be solved in the future.

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