

Modernization of vertical Pelton turbines with the help of CFD and model testing

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Abstract. The modernization of water turbines bears a high potential of increasing the already installed hydropower capacity. In many projects the existing waterways allow a substantial increase of the available flow capacity and with it the energy output. But also the upgrading onto a state of the art hydraulic, mechanical and electrical design will increase the available power considerably after the rehabilitation. The two phase nature of the flow in Pelton turbines requires for the hydraulic refurbishment special care in the application of the available design methods. Where the flow in the high pressure section of the turbine is mainly of one phase nature, CFD has been used as a standard tool for many years. Also the jet quality, and with it the exploration of the source of flow disturbances that cause poor free surface quality can be investigated with CFD. The interaction of the jet with the buckets of the runner is also examined by means of CFD. However, its accuracy with respect to hydraulic efficiency is, because of the two phase flow and the transient flow process, in very few cases good enough for a reliable and accurate prediction of absolute numbers. The optimization of hydraulic bucket profiles is therefore always checked with measurements in homologous scaled model turbines. A similar situation exists for the housing flow after the water is discharged from the runner. Here also CFD techniques are available to explore the general mechanisms. However, due to the two phase flow nature, where only a very small space is filled with moving water, the experimental setup in a model turbine is always the final proof for optimizations of housing inserts and modifications. The hydraulic design of a modernization project for a power station equipped with vertical Pelton turbines of two different designs is described in the proposed paper. It will be shown, how CFD is applied to determine the losses in the high pressure section and how these results are combined with the model tests carried out in the hydraulic laboratory. Finally a comparison is made in between the achieved model turbine results with measurements carried out in the prototype.

1. Introduction

Of all turbine types, Pelton turbines are the most complex concerning the involved physics that describe the flow. While the single phase flow in reaction turbines can be reasonably reduced to a stationary flow situation, many of the flow phenomena that determine the performance of a Pelton turbine need to be treated as fully transient. In addition, downstream of the nozzles the flow in a Pelton turbine is no longer single phase, making it necessary to apply physical models for the prediction and reconstruction of the free surface interface. As a consequence the



computational requirements for a Pelton flow simulation are far higher than for a typical Francis runner simulation and the accuracy of the prediction is not as high.

Fifteen years ago the hydraulic design of Pelton turbines was based on model tests only. Lazzaro et. al. [1] and Brivio [2] showed typical examples of model test based developments. The design was to a certain extent based on trial and error strongly supported by flow visualizations in the model turbine.

The first step into the computer supported hydraulic design of Pelton runners was done by Kubota [3] with his animated cartoon method. Janetzky [4] showed the first transient simulations of the flow through a Pelton bucket based on a Finite Element formulation of the Navier Stokes equations. The modelling of the two phase nature of the flow was already done with today's well known Volume of Fluid (VOF) method.

The first application of CFD for Pelton turbines in an industrial environment was presented by Mack et. al. [5]. The validation of the flow prediction through the buckets was described by Mack in [6] following a description of the complete potential of computational fluid dynamics for Pelton turbines in [7]. Today the numerical prediction of the fluid flow is part of the hydraulic design as Rohne et. al. has shown in [8]. However, whenever absolute numbers on the efficiency are required or the housing flow plays a dominant role in the design, model tests are indispensable.

A typical example for this situation is the modernization of vertical Pelton turbines, like the machines installed in the powerstation LUENERSEE in the Austrian Alps. Built from 1954 to 1958 the pump storage powerstation LUENERSEE serves the primary grid control. It is equipped with five ternary units, each consisting of a motor generator, a four jet Pelton turbine, a torque converter and a five stage storage pump. The Pelton turbines installed in LUENERSEE are of two different designs. Their main technical data before the modernization of the turbines are listed in Table 1.

Table 1. Technical data before the modernization

	unit 1 + 2	unit 3 - 5	
jet circle diameter D_1	1630	1600	<i>mm</i>
runner outer diameter D_a	2040	1972	<i>mm</i>
number of buckets z_2	23	23	—
bucket width b	388	360	<i>mm</i>
number of jets z_0	4	4	—
rated power of turbine	46.2		<i>MW</i>
rated power of motorgenerator	56		<i>MVA</i>
flow per turbine	5.52		$\frac{m^3}{s}$
rated head	894		<i>m</i>
rotational speed	750		$\frac{1}{min}$

In 2009 Voith Hydro St. Poelten got the contract for the modernization of all five Pelton turbines installed at LUENERSEE. The scope of the modernization project with respect to the hydraulic design of the Pelton turbines was to deliver five new runners with increased efficiency and output, the modification and repair of the installed nozzles and the hydraulic design including the model test for both turbine designs.

One requirement for the modernization was, that the new runners had to be exchangeable between the different turbine makes, without modifying the jet circle diameter. In addition, the maximum output of the turbines got increased to 56 MW.

2. Hydraulic Development

The hydraulic development for the modernization of the two Pelton turbine designs was a combined effort of model testing and numerical prediction. This way, the homology in the model test could be focused on the dominant components that require an accurate prediction on the performance, whereas the components that are reliably predicted with numerical methods were analysed in an economic way. In the following the different components of the machine together with the strategy for the homology is described. For easier reference the hydraulic design of the machines for unit 1 and 2 is denoted as Design A, whereas the design of the units 3 to 5 is named Design B.

2.1. Distributor line

Both distributor lines in the prototype are casted components as they were common during the time of their installation. Their hydraulic shape made use of acceleration bends without edges as they are typical for today's Tri-Cone welded solution. Since no modification was foreseen for the hydraulic shape in the distributors and a reliable prediction with numerical methods was available, a standard distributor design was used for the model test and the influence of the individual designs was predicted with computational fluid dynamics. All simulations were done in model size, to estimate the effort with respect to its behaviour on the model efficiency.

In Figure 1 a comparison of the CFD results for the two prototype designs and the chosen standard geometry is shown. In this analysis the velocity distribution in the symmetry plane is displayed. As one can see, the bifurcations in the casted designs show very little zones of separating flow. In addition the velocity field close to the walls is very homogenous compared to the standard design, which reflects a welded solution.

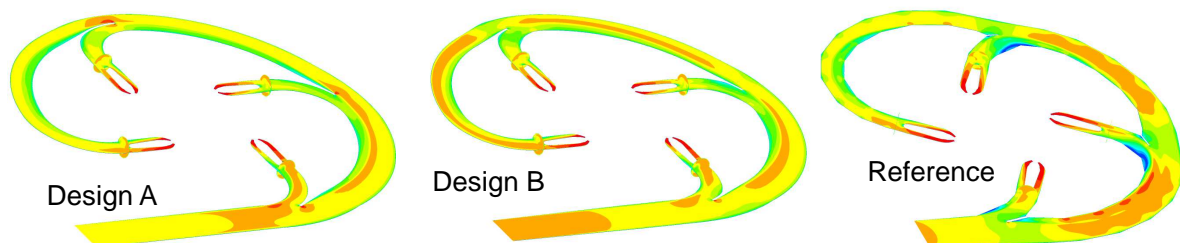


Figure 1. CFD results of the distributor simulation for Design A, Design B and Reference design. Shown in the figure is the distribution of the velocity in the symmetry plane.

The influence on the performance of the different distributor designs is given in Figure 2. In this graphic, the energy loss between the distributor inlet and the inlet into the nozzles is displayed as an efficiency difference of the two casted designs to the reference distributor design. As the simulations have been carried out for 4 different flows, the efficiency difference is displayed as a function of the bucket specific load. For comparison reasons, the efficiency differences are normalized with the standard distributor value at a $\Phi_B = 0.1$.

As the values in Figure 2 show, the energy loss in the casted designs is larger than in the reference design, even though the flow in the main part of the distributor is more homogeneous. The reason for this behaviour is the stronger redirection of the flow, which is in both casted designs 90° whereas the reference design is only redirecting the flow by 60° .

For the prediction of the effect of a homologous model distributor the values of Figure 2 are used as it will be explained in section 2.5.

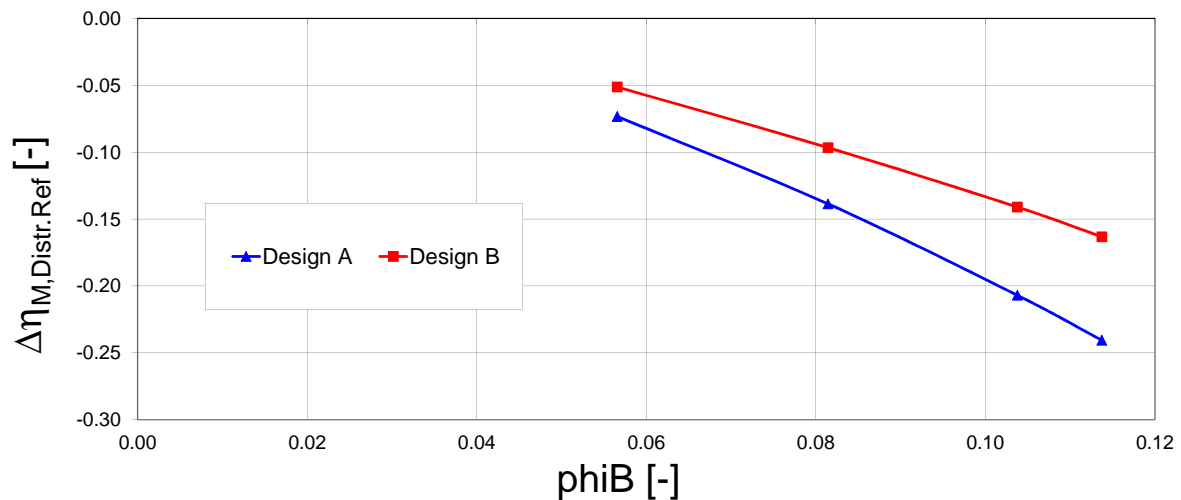


Figure 2. Influence of the distributor design on the efficiency. Shown in the two graphs are the efficiency differences of the individual designs compared to the reference distributor used in the model test.

2.2. Nozzle

For the hydraulic design the original installed nozzles had to be adjusted in order to cope with the higher output and to improve the hydraulic performance. The comparison of the two hydraulic designs, however, shows significant differences. While both nozzles are controlled by an outside servomotor, the meridian cross section and the number of ribs that support the needle rod are different. In Figure 3 the two designs are shown in comparison.

While Design B uses a conical housing shape for a constant acceleration the Design A is only accelerating in the nozzle mouth region up to jet velocity. Since both nozzle inlet diameters are around the same size, the average velocity inside the Design B will be higher than in Design A. The second main difference between the two designs is the number of ribs that support the needle rod. Design A uses eight ribs, whereas Design B is only using three ribs. In addition, the nozzles of Design B use a flow straightener that reduces the secondary flow. The effect of the flow straightener is shown in Figure 4 with the results of a comparative simulation. While the three ribs alone are not able to eliminate the secondary flow, the resulting jet surface shows a strong deformation which results in less favorable conditions for the jet bucket interaction and possibly even cavitation damages. The use of the flow straightener, however, leads to a compact jet as the secondary flow induced by the upstream bifurcations has been significantly reduced.

Since the jet quality and its influence onto the jet bucket interaction has a higher prediction quality in the experiment the influence of the homologous nozzle was tested also by model test. However, in order to save experimental effort for the model tests, the hydraulic prediction of the nozzle influence was carried out with single jet comparison tests. This single jet comparison test was carried out by exchanging the nozzle at position 4 in one test with a standard design and in the comparative tests with the homologous scaled nozzle designs. These tests were run only with the nozzle at position 4 in operation. By keeping the rest of the model turbine the same, the only influencing part that is causing the measurement differences are the effects that are induced in the nozzle, including the jet quality and its effects on the jet bucket interaction.

The extract of the single jet comparison tests is shown in Figure 5. As for the distributor analysis, the efficiency values for the two designs are shown here as well as a difference to the test results of the standard nozzle design again normalized with the values of the standard nozzle at $\Phi_B = 0.1$.

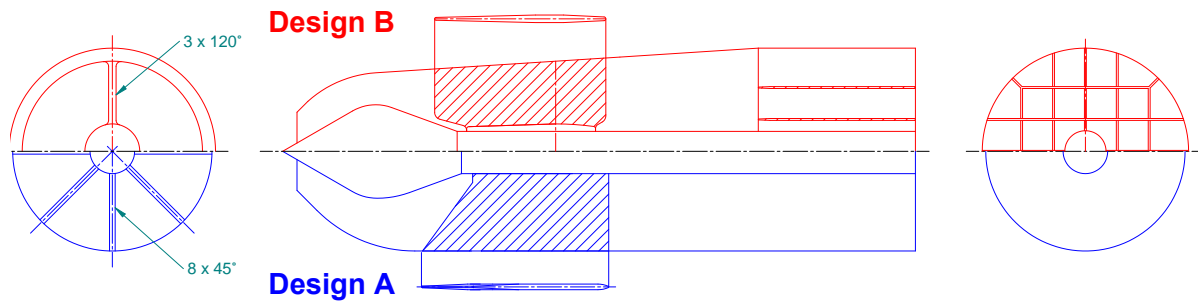


Figure 3. Meridian profile of the two nozzle designs installed in LUENERSEE. While Design B uses a conical housing to maintain a constant acceleration throughout the nozzle Design A is accelerating only in the nozzle mouth part. Design B is using a flow straightener to compensate the reduced number of ribs.

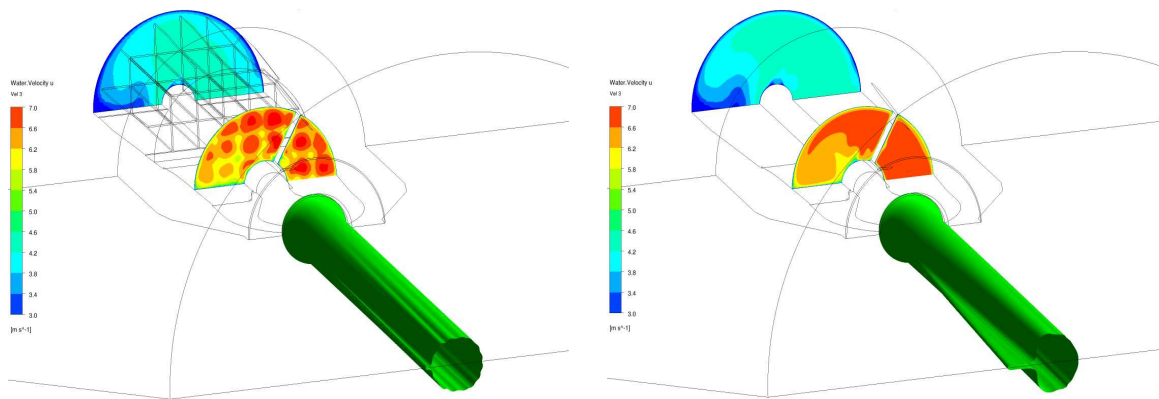


Figure 4. Influence of the flow straightener installed in Design B. The three ribs only will allow the remaining secondary flow to deform the jet after it leaves the nozzle mouth. With the flow straightener the secondary flow is reduced and the resulting free jet has higher quality.

Remarkable on the test results displayed in Figure 5 is the fact, that Design B shows clearly higher performance at bucket specific loadings smaller than $\Phi_B = 0.08$, whereas at higher flows the performance is decreasing stronger than Design B. This behaviour can be explained with the existence of the flow straightener. While at lower flow the increased jet quality helps the jet bucket interaction, the friction loss within the flow straightener plays a more dominant role at higher flow.

2.3. Runner

The design of the new runners was faced with the two prerequisites, that on one side the output had to be increased to result in 56 MW and the fact, that the runners had to be interchangeable in between the different turbine designs. Since the jet circle diameter of the two designs is different ($D_{1,Dsg.B} = 1600\text{mm}$, $D_{1,Dsg.A} = 1630\text{mm}$), special attention had to be given on its main dimension and the interaction with the stationary parts.

The final dimensions of the new runner were found in the model tests. The bucket width was increased as a consequence of the increased output. With this value the necessary margining to the full load operating points was achieved, keeping the optimum at the required level.

With the increased bucket width also the number of buckets was reduced. Detailed

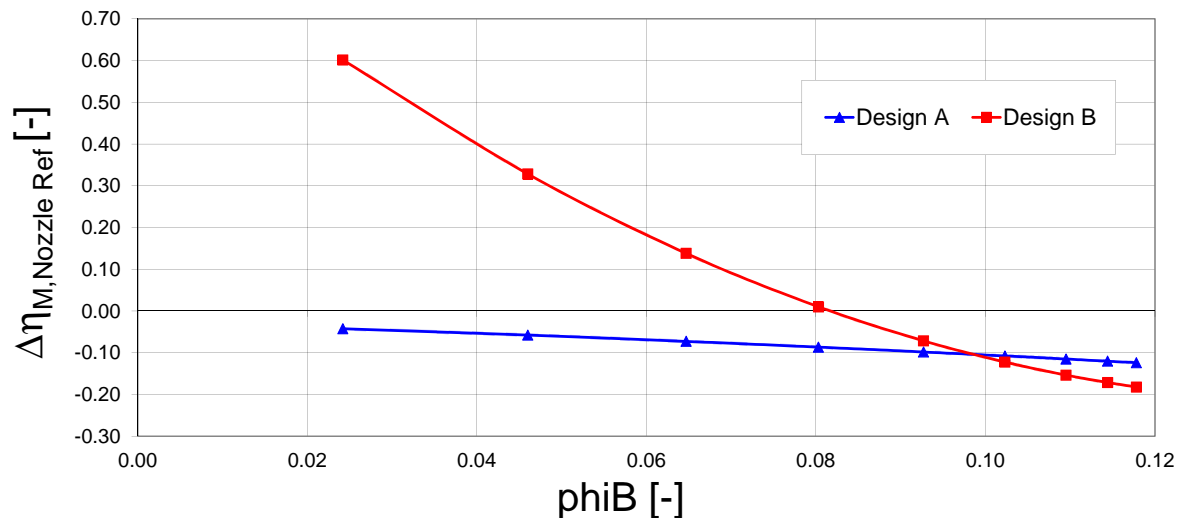


Figure 5. Influence of the nozzle designs on the efficiency. Shown in the two graphs are the tested efficiency differences of the individual designs compared to the standard nozzle.

investigations on the shaft line showed, that the reduced number of buckets will not lead to any excitation that is causing harm to the components connected to the primary and secondary shaft line.

The runner was homologous in its hydraulic active surfaces tested in the model tests. In Figure 7 the runner, build out of single buckets is shown installed in the model machine. This way the component mainly responsible for the performance was tested with the highest level of accuracy in terms of absolute efficiency.

2.4. Housing

Both housing designs serve the 4 jet distributor design with an eight corner housing. The general shape of the plan view and the cross sections, however, is completely different. While Design B is utilizing a regular octagonal shape for the ground floor, Design A uses a square with chamfered corners. A 3D view of the two different designs is shown in Figure 6 for visual comparison.

Compared to modern designs, the main dimensions of the installed housings are small. Therefore the model tests had to consider the shape and dimensions homologous scaled from the prototype. In Figure 7 the inner side of the two model machines for Design A and for Design B are shown. Transparent windows were included in the mechanical design of the housings to allow flow observations during the tests.

Included in the model machine were also the deflectors, homologous scaled together with the rods going through the ceiling of the housings. This way, any positive or negative influence of the deflectors on backflowing water was considered during the measurements. Part of the model machine was also the central shaft cover, which protects the primary shaft that is connected to the hydraulic torque converter. The housing cone above the runner was scaled homologous as well to get any influence during the dewatering of the buckets.

2.5. Efficiency step up

The prediction of the efficiency derived directly from the model test results to the individual prototype values is done in three steps, which are illustrated in the Figure 8. The model tests, carried out with the standard distributor, the standard nozzle, the homologous runner and the homologous housing is in the first step corrected by the results of the distributor CFD, as

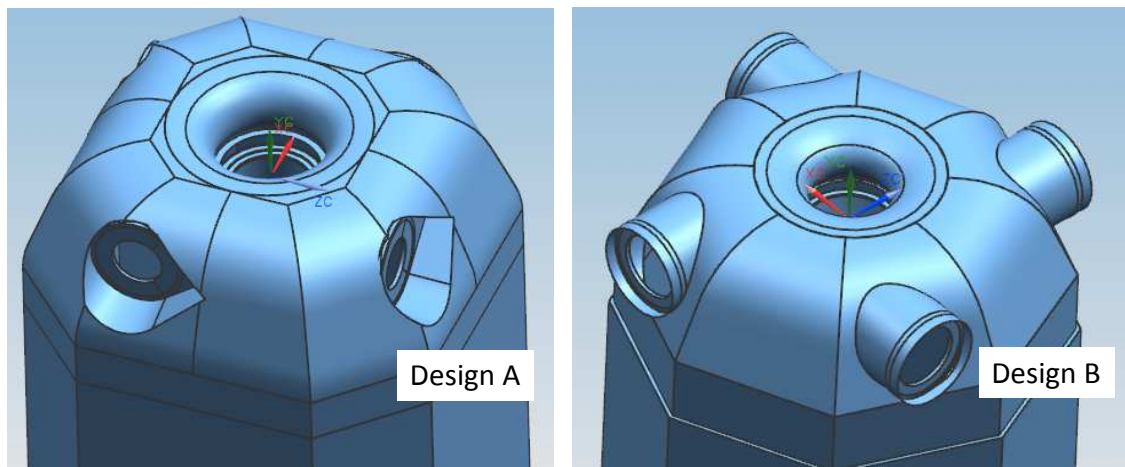


Figure 6. 3D illustration of the two housing designs installed in LUENERSEE. While Design B is based on an octagonal floor plan uses Design A a square with chamfered corners. Both designs have been homologous tested in the model turbine.

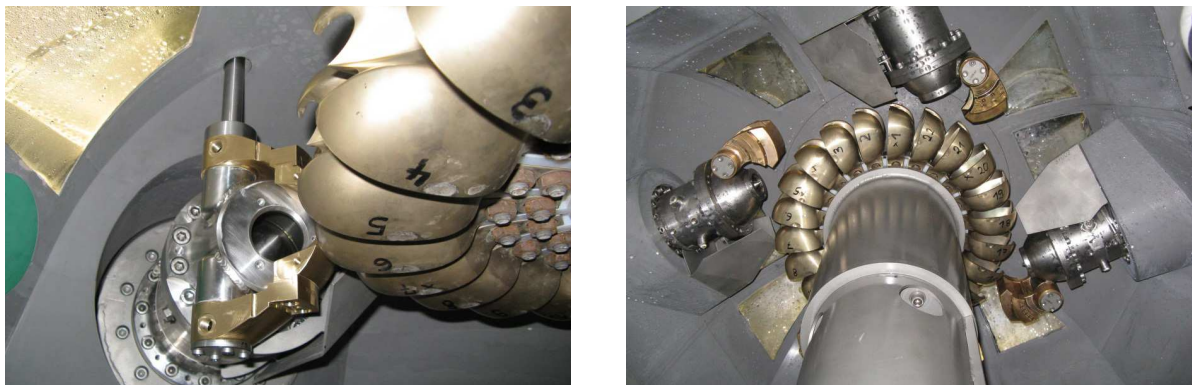


Figure 7. View into the model turbines. Both housing designs have been homologous scaled reproduced using transparent windows for flow visualization. The influence of the central shaft cover which protects the shaft to the hydraulic torque converter was also investigated for both housing designs during the model tests.

explained in section 2.1. The so corrected performance values reflect a model machine which is only semi homologous with respect to the nozzles.

In the second step the model test results treated by the results of the distributor simulations are corrected by the results of the single jet model tests for standard and homologous nozzles as explained in section 2.2. The resulting performance characteristics do reflect the situation of a fully homologous model test.

The so derived model test characteristics are scaled to prototype using the efficiency correction as described in IEC 60193, Annex K to arrive at the performance characteristic of the prototypes. In Figure 9 the comparison of the guarantees with the model test based derived values are shown. One can see, that over the complete output range the guarantees were met. The strong full load behaviour of the optimized hydraulic design is a consequence of the high weighting value of the guarantee points at high output.

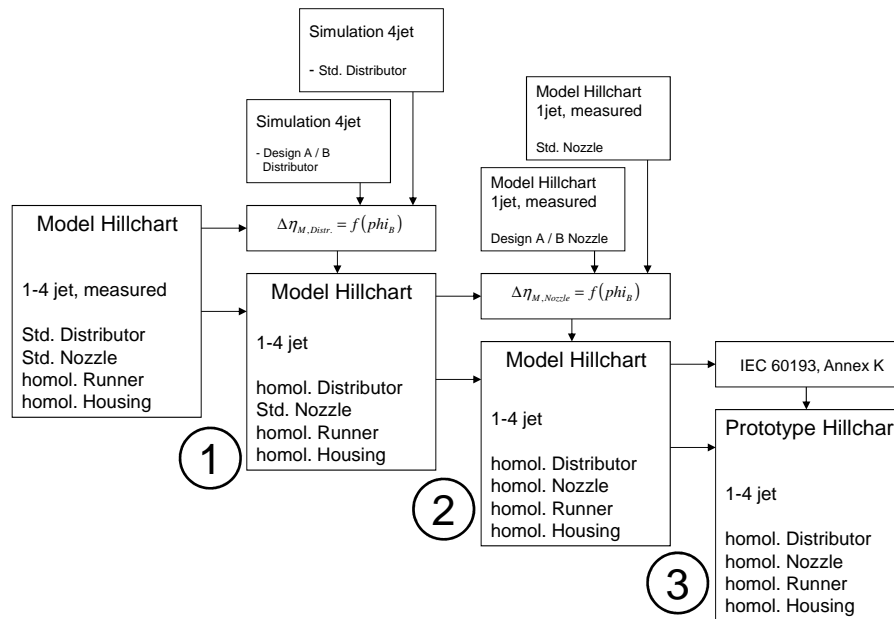


Figure 8. Graphical illustration of the determination of the prototype efficiency, based on the flow simulations for the distributor line, the 1 jet comparison tests for the nozzles and the multi jet model tests with homologous runner and housing.

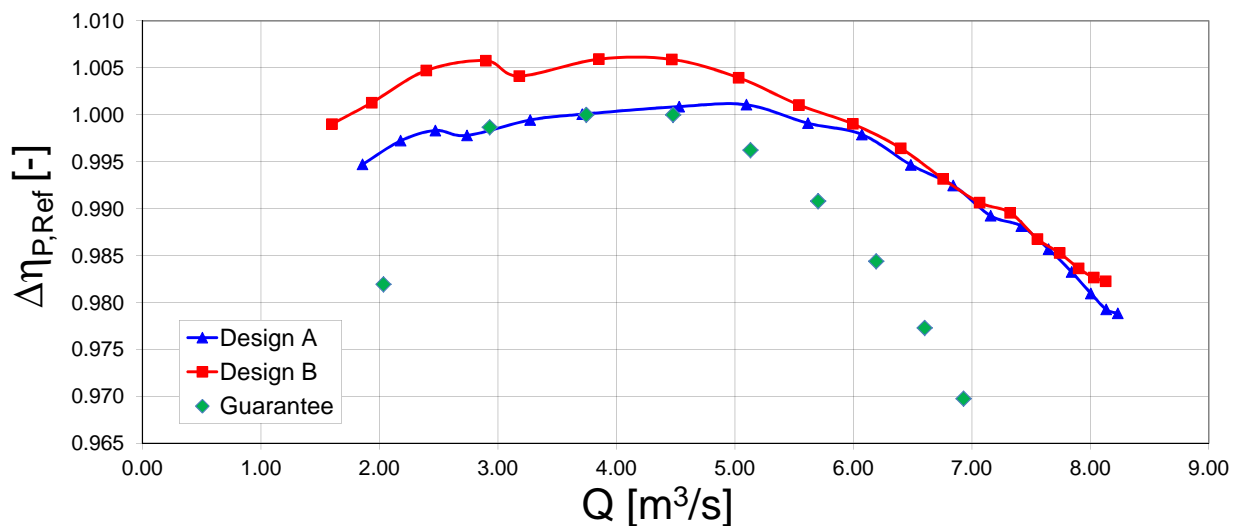


Figure 9. Comparison of the guarantees with the model test based predicted prototype values.

3. Prototype Measurements

After the first machine in the powerhouse was modified and the new runner installed, thermodynamic tests were carried out. The result of these measurements is shown in Figure 10 as normalized values. The tested machine is of Design B.

One can see, that the trend of the curves and the distance to the guarantees is similar to the values determined by the model test measurement based predictions. Over the complete range of the load the guarantees are met. At full load the efficiencies are clearly exceeded.

As a side effect the noise level of the prototype machine was significantly reduced. Even at the new maximum output the noise level is below what it was at the original limit setting. This effect is a direct consequence of the increased efficiency after the modernization.

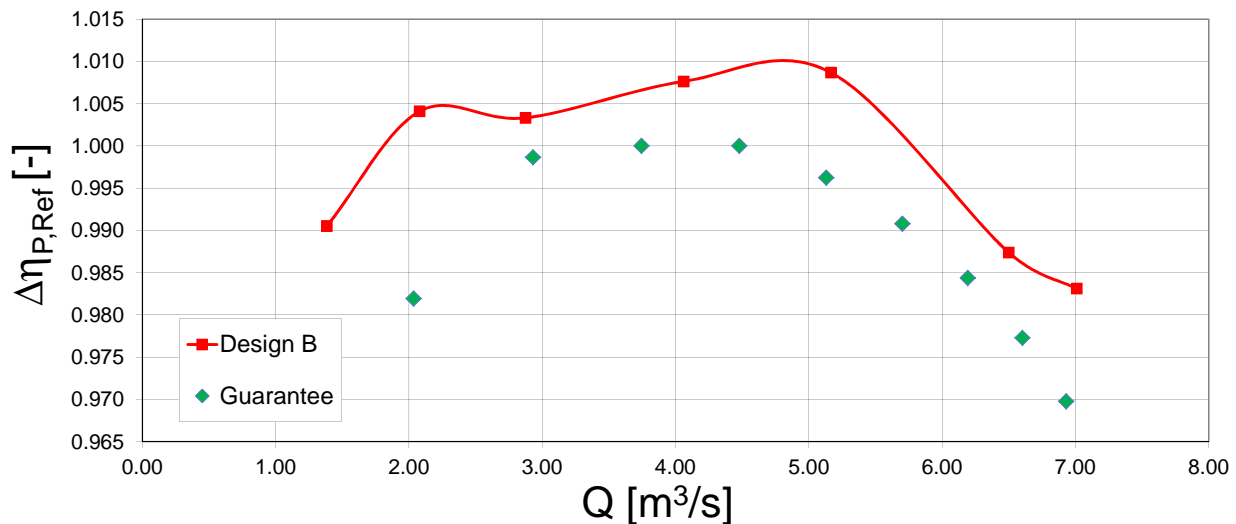


Figure 10. Comparison of the thermodynamic measurements in the prototype with the guarantees given for the modernization. The measurements have been carried out at a turbine with Desing B.

4. Conclusion

Modernization projects as described above are very common in today's market and bear a high potential for increasing the output with reasonable investments. The accurate design and a reliable prediction of the performance, however, has to consider the appropriate tools and a reasonable strategy in terms of model testing.

Because of the two designs installed at LUENERSEE a complete homologous model test measurement would not have been justifiable and therefore a combined CFD and physical model solution was chosen for the hydraulic optimization. During the development phase the numerical investigation of the individual distributor designs turned out to be economically advantageous and also an accurate prediction method to determine the hydraulic influence of these components. Likewise, the single jet comparison tests helped to reduce the effort in the hydraulic optimization, having the accuracy of the model test necessary to determine the jet quality and with it the influence on the jet bucket interaction. Since the housing dimensions were small compared to modern designs, a homologous scale down into the model test turned out to be the correct approach for this project.

The close comparison between the model test based prediction and the results of the thermodynamic tests has proven, that the strategy for the hydraulic optimization based on semi homologous model tests, single jet comparison measurements, and reasonably applied numerical simulations was a valid and cost effective approach for a successful execution of this modernization project.

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