Optimized initial down-scaled model design of SD-CRPT

D2.1.

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Executive Summary

This report describes how the optimised initial down-scaled model design of the Shaft-Driven Counter-Rotating Pump-Turbine (SD-CRPT) in the ALPHEUS project was derived. A first design was made in prototype scale, according to the expected requirements of the final machines. The design was scaled to an appropriate model scale with respect to requirements from the experimental test facility at TU Braunschweig. In model scale a numerical sensitivity analysis was conducted on the initial design to narrow down the number of parameters for the optimisation process. With a better understanding of the decisive parameters of the machine, a full optimisation process of the blade geometries were carried out. The goal with the optimisation procedure was to maximize the average net power considering the power input in pump mode and power output in turbine mode. In addition constraints were imposed on minimum head in pump mode in order to meet the test facility requirements at TU Braunschweig and constraints on risk of cavitation. The latter has a positive effect on fish friendliness, but also may extend the lifetime of the machine. The process has designed an optimised initial down-scaled model that is ready to be used for further numerical studies, as well as to start the design and manufacturing of the experimental set-up. It has been decided by the consortium that it is the SD-CRPT that will be experimentally studied, rather than the Rim-Driven Counter-Rotating Pump-Turbine (RD-CRPT).

1. Introduction

ALPHEUS deliverables D2.1 (the present report) and D2.2 (a parallel report) should describe the work leading to optimised initial model scale designs of the Shaft-Driven Counter-Rotating Pump-Turbine (SD-CRPT) and the Rim-Driven Counter-Rotating Pump-Turbine (RD-CRPT), respectively. At the time of the deliverables it should also have been decided which of those designs to proceed with for the experimental validation studies at TU Braunschweig. This decision was made earlier in the project due to feasibility limitations. The RD-CRPT generators/motors would require developments of components that are not available “off the shelf”, and for which the developments in ALPHEUS are planned in other WPs at a later stage. Already the initial design of the SD-CRPT had very promising properties, which would also improve during the optimisation process. It is thus a relevant candidate for the final design. Further, the purpose of the experiments is to validate the CFD simulations, and the similarities of the machines will make it possible to use the validation for one of the two designs to also validate the CFD results for the other design. The SD-CRPT type was hence chosen to be tested experimentally at TU Braunschweig.

Due to the early decision that the experimental tests will be made on the Shaft-Driven Counter-Rotating Pump-Turbine, more focus has been put on the optimisation of this alternative for the M9 deliverables (D2.1 and D2.2). This way the experimental validation can be made on a model design that is as good as possible at this stage. The studies of operation of the machine and the concept of counter-rotating pump-turbines will be more relevant for a model that has gone through a more thorough optimisation process. A rigorous optimisation study has been performed for the SD-CRPT, which considers both changes in flow path and chord distribution as well as changes to the 3D blade shape.

The initial design of the SD-CRPT has already been investigated under unsteady and even transient conditions in model scale to ensure the performance of the machine under more realistic conditions. The unsteady simulations have provided invaluable information on how the experiments should be conducted. The unsteady results also suggested designs of the parts of the machine that were not included in the optimisation process (hub, support struts, draft tubes, etc.), and where and how the measurements should be made.

The rim driven alternative has until now been subjected to a manual optimisation procedure with a number of design iterations. A rigorous multi-objective optimisation of the prototype scale RD-CRPT will be done in a similar fashion as that of the SD-CRPT, which will be part of deliverable D2.7.

1.1 Initial Design

ADT’s TURBOdesign suite [1] was used to create the initial design of the Shaft-Driven Counter-Rotating Pump-Turbine (SD-CRPT) in prototype scale. The software uses a 3D inverse design method [2], [3], [4] to design the blade shape for a given distribution of blade loading (pressure jump across the blade). It can be used for axial, mixed flow or centrifugal configurations and can easily handle contra-rotating stages. The target design for TURBOdesign was a machine with a power of 10 MW in turbine mode and a head of 9 m at the Best Efficiency Point (BEP). Initially a meanline sizing code TURBOdesign Pre was used to generate the initial flow path (hub and shroud diameter and chord distribution of the first and second rotor) based on specification of flow rate, rpm and head of 9 m. The resulting flow path was then used in the inverse design method with appropriate choice of blade loading to generate the first and second rotor 3D blade geometries. The performance of the initial stage was then verified by full 3D steady RANS computations which confirmed that the design meets the required head and power levels at reasonably high level of efficiency.

1.2 Scaling Parameters

The prototype stage was designed in prototype scale to ensure the required performance according to the ALPHEUS description, and to guarantee the strength of the blades at full scale. To account for limitations at the lab an appropriate model scale was derived based on scaling laws for hydraulic machines. The scaling law assumes that the flow coefficient, work coefficient, and power coefficient are constant between the prototype and model scales. This leads to relationships between the model and prototype scales as

$$\frac{Q_{p,i}}{Q_{m,i}} = (\frac{D_{p,i}}{D_{m,i}})^3 \left(\frac{\Omega_{p,i}}{\Omega_{m,i}}\right) \qquad (1.1)$$

$$\frac{H_{p,i}}{H_{m,i}} = (\frac{D_{p,i}}{D_{m,i}})^2 (\frac{\Omega_{p,i}}{\Omega_{m,i}})^2 \qquad (1.2)$$

$$\frac{P_{p,i}}{P_{m,i}} = (\frac{D_{p,i}}{D_{m,i}})^5 (\frac{\Omega_{p,i}}{\Omega_{m,i}})^3 \qquad (1.3)$$

Here Q is the flow rate, Ω is the rotational speed of the runner, D is runner diameter, H is the head, and P is the power. Subscripts p and m represent prototype and model scales, respectively, and i indicates runner 1 or 2, since there are two runners.

1.3 Optimisation

An optimisation procedure may look very different depending on what is meant with the optimisation, and what the goal is. Common for all optimisation procedures are that different quantities are compared and either maximised or minimised to make the overall performance of the product better.

The present optimisation procedure includes a sensitivity analysis to reduce the number of input parameters for the full-scale multi-objective optimisation procedure.

The sensitivity analysis included in total 21 design parameters. They were distributed across the parameter space with a Latin Hypercube Sampling into 35 simulated design points. The sensitivity analysis was carried out at multiple operation points, and for both pump and turbine modes.

Based on the results from the sensitivity analysis a narrower number of design parameters was chosen for the full multi objective optimisation procedure. The goal of the optimisation was to maximise the average power output of the turbine and minimise the power required for the pump, see section 1.3.1, and reduce the risk of cavitation. The latter is because it is important to reduce cavitation for fish mortality, and also extend the lifetime of the machine.

1.3.1 Pump and Turbine Objectives

The optimisation procedure was carried out at multiple operating points for both pump and turbine modes. The main goal was to maximise the average power output of the turbine and minimise the average power input required for the pump. The aim was to achieve a design that maximises the round trip power output of the pump-turbine.

2. Initial Design

The initial design of the SD-CPRT was derived with ADT's TURBOdesign suite at prototype scale. Figure **1** shows the two runners 3D geometry. Note that Rotor1 has 8 runner blades and Rotor2 has 7 runner blades.

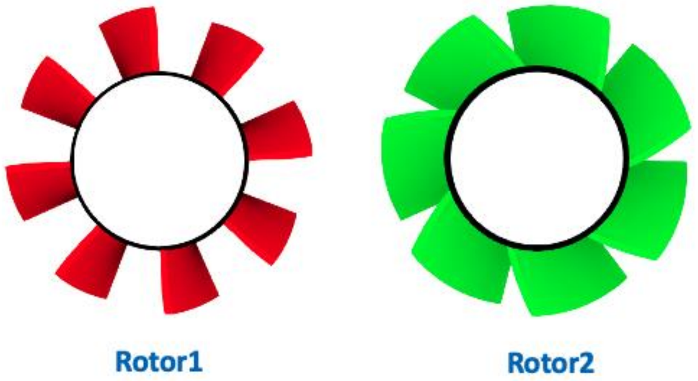


Figure 1 : Geometry of Rotor 1 and Rotor 2

Table **1**summarises the input data to TURBOdesign in order to generate the blade profiles depicted at Figure **1**. A constant work coefficient is specified at the trailing and leading edges of the runners, and a constant thickness of 60 mm is used.

As can been seen in Table **1**, the design flow rate is 130 m^3/s in pump mode and the two runners rotate in opposite directions. The second runner rotates at 90 % of the speed of the first runner.

Meanline Design Details	Shaft Driven
Flow rate [m3/s]	130
Rotor1 speed [rev/min]	50
Rotor2 speed [rev/min]	45
Hub Diameter [mm]	3502
Shroud Diameter [mm]	6064
Maximum rotor1 axial span [mm]	773
Maximum rotor2 axial span [mm]	1220
Minimum axial gap between rotors [mm]	200

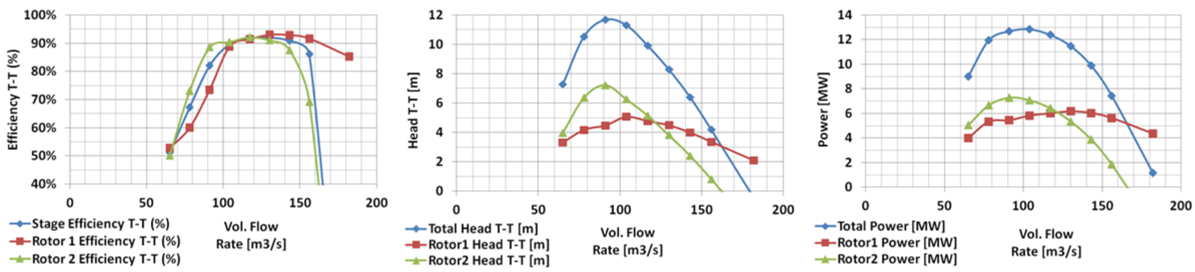
Table 1 : Main parameters from TURBOdesign Pre

2.1 Performance, CFD

The initial prototype design was evaluated with Computational Fluid Dynamics (CFD) to ensure that the performance is in accordance with the ALPHEUS goals. The simulations were based on steady-state computations at a number of operating points to evaluate the operation range of the machine. Commercial CFD code ANSYS CFX was used for all computations [5]. For steady state analyses with uniform inflow, the impeller/runner is typically modelled as a single blade channel bounded by periodic (repeating) boundaries in the circumferential direction [6]. This serves to substantially reduce the size of the model and to speed up the computational analyses, which is of utmost importance for design purposes. The two rotors are separated by a mixing plane interface responsible for transferring flow quantities between the two counter-rotating domains. The k-ω SST eddy viscosity model was used to account for turbulence, and the computational mesh contains roughly 7.3×10^6 cells. As for boundary conditions:

- In pump-mode the flow rate is specified at the inlet,
- In turbine-mode the total pressure is used at the inlet.

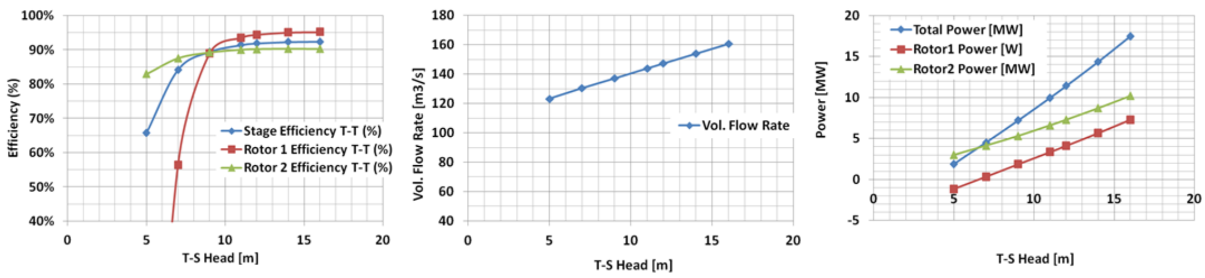
Figure **2** shows the computed efficiency, head, and power in pump mode as a function of flow rate for the initial design in prototype scale. The results depicted in Figure **2** shows that the efficiency is about 90% for a wide range of operating conditions. The head and power results show that the initial design is in the right range according to the ALPHEUS requirements already before any optimisation is made.



a) Efficiency b) Head c) Power

Figure 2 : Efficiency, head, and power as a function of volumetric flow rate in pump mode for the initial design of the SD-CRPT

Figure **3**shows the corresponding efficiency, flow rate, and power, in turbine mode displayed as a function of total-to-static head. The results indicate a relatively high efficiency over a head range of 8-16 m but efficiency drops very rapidly below 8 m head. The volumetric flow rate and power depicted show that both the power and flow rate increase linearly with the increase of head.



a) Efficiency b) Volumetric flow rate c) Power

Figure 3 : Efficiency, volumetric flow rate, and power as a function of total-to-static head in turbine mode for the initial design of the SD-CRPT

2.2 Performance, FEA

The CFD simulations only include the fluid flow, and to ensure that the machine can handle the loads a Finite Elements Analysis (FEA) was carried out at the best efficiency point in pump mode. The pressure field from CFD and the runner rotational speed is included in the FEA as the supplied loads. Based on the material properties of the bulb turbines at the Rance tidal power plant, the runners are assumed to be manufactured in conventional Stainless Steel, SS 17-4 H1075, with a yield strength of 1020 MPA, and ultimate strength of 1130 MPA.

The FEA showed that the maximum von-Mises stress occurs at the hub for both the runners, and the maximum value is 151 MPA. This means that the safety factor to the yield strength is above 6.7, and the risk of failure of the runners is thus low.

3. Scaling to model scale

The scaling between the prototype and model scales was made at a point 120% Volume Flow Rate (VFR) of the BEP in both pump and turbine modes. This is because in pump mode the machine must overcome the head losses, and in turbine mode the losses need to be subtracted, in the experimental test facility. In the derivation of the prototype design no head losses are assumed, hence to ensure the machine has sufficient available power it was scaled at 120 % VFR of BEP, which also is regarded as the most extreme scenario that the lab can reproduce. Equations 1.1 - 1.3 were utilised to get the size and operation point in model scale. The constraints for the scaling were a maximum flow rate of 500 l/s, a rotational speed of the runners that are maximum 1500 RPM, and the minimum head in pump mode is 8.0. m. At the time of scaling, the minimum head target of 8.0 m was based on available information in terms of the proposed test set up at TU Braunschweig. The head includes both the height elevation between the upper and lower basins in the lab and the head losses from the pipes. It was assumed that the power and rotational speed-ratio are the same in prototype and model scale. The parameters in model scale are presented in Table **2** for pump mode in a), and turbine mode in b).

Pump Mode		
	R1	**R2**
D_{shroud}[mm]	276	276
N_m[rev/min]	1501.9	1351.7
Q_m[m3/s]	0.442	0.442
P_m[kW]	29.67	9.66
H_m[m]	6.45	1.55

Turbine Mode		
	R1	**R2**
D_{shroud}[mm]	276	276
N_m[rev/min]	842.4	758.1
Q_m[m3/s]	0.234	0.234
P_m[kW]	3.86	6.8
$H_{m\ total\text{-}total}$[m]	1.78	3.22

Table 2 : Scaled parameters of the initial SD-CRPT. Note that R1 and R2 denote Rotor1 and Rotor2.

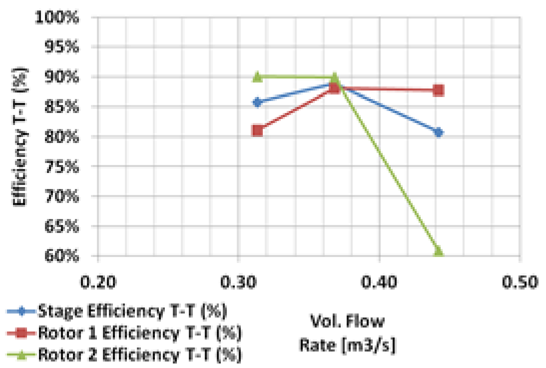
According to the scaling laws the appropriate model scale is a machine with a runner diameter of 0.276 m, compared with 6 m in prototype scale. The rotational speed has also increased drastically, from 50 RPM in prototype scale, up to 1502 RPM for Rotor1 in pump mode, see Table **2**for **R1**.

3.1 Performance in model scale

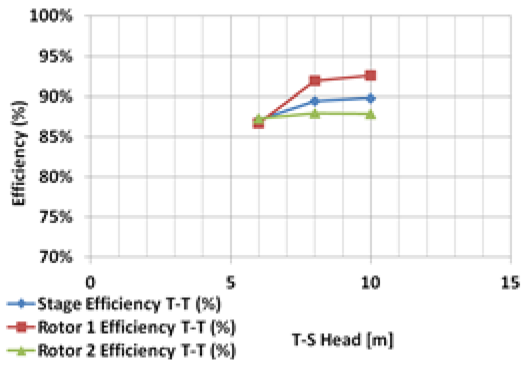
Before the optimisation process was started the performance was also evaluated in model scale. This is important since it is necessary to know the performance before the optimisation, to be used as reference. The evaluation in model scale also shows how well the scaling has worked, and how sensitive the machine is to scaling, since not all parameters can be included in a geometrical scaling.

The efficiencies in model scale are shown in Figure 4 for both pump mode in a), and turbine mode in b). By comparing the efficiencies in pump mode between prototype scale, Figure 2, and in model scale, Figure 4 (a), it is found that the performance has been affected by the scaling. A lower efficiency is predicted in model scale by the CFD simulations. The scaling effects are less drastic in turbine mode, comparing

Figure **3** and Figure **4** (b). The characteristics in model scale are recognised in turbine mode, with an efficiency of above 90 % as the head is increased.



a) Pump Mode



b) Turbine Mode

Figure 4 : Model scale efficiency as a function of flow rate in pump mode and head in turbine mode.

4. Sensitivity analysis

For optimization it was decided to use both meridional geometry parameters such as hub and shroud diameter and chord distribution as well blade loading parameters for inverse design that control the 3D blade geometry of each. Hence, before full surrogate model based optimization is run it's important to do a sensitivity analysis to find the most important design parameters to use for detailed optimization. The aim with the sensitivity analysis was to reduce the number of parameters for the full optimisation.

4.1. Initial number of parameters

The blade design is determined by a number of input parameters in ADT's TURBOdesign suite. For the sensitivity analysis 21 design parameters were chosen and distributed in the design space with the Latin Hypercube Sampling. In total 35 designs were compared, at three operation points and for both pump and turbine modes, with a linear response surface model [7] to find the most influential parameters on the performance of the machine. The parameters are categorised in different sections with eight parameters correlated to the meridional geometry, ten parameters to the blade loading and rotor work, and three parameters related to the blade stacking and RPM ratio. The shroud diameter was maintained unchanged to 0.276 m through the sensitivity analysis and all the parameters are varied with the initial design as the reference for the specific parameters. The responses from the sensitivity analysis are efficiency, power, and head.

4.1.1 Important parameters

The outcome of the sensitivity analysis suggests that 11 out of the 21 parameters have more influential on the key performance parameters such as efficiency, power and head and the remaining are less influential according to the response surface model on stage total to total efficiency. The influential parameters are presented in order of importance in Table **3**. The rotor work coefficient is the most important parameter, followed by the speed ratio of the runners. It is promising that the speed ratio is an important parameter since the final machine will be controlled by changing the rotational speed and speed ratio of the two runners. From the parameters, six originate from the blade loading and rotor work, four from the meridional geometry, and one for the RPM ratio. These more sensitive parameters will be used in the next stage to run a full Design of Experiments (DoE) and optimise the design using a surrogate model.

Order of importance	Parameter Name	Parameter Description
1	R1 RVTMEAN_TE	Rotor work coefficient
2	R2_R1_RPM_Ratio	Rotor 2 to Rotor 1 rotational speed ratio
3	Hub_Tip_Ratio1	Hub radius to shroud radius ratio for Rotor 1 leading edge and Rotor 2 trailing edge
4	R1_TE_TIPtoMID_rvtRatio	Rotor shroud to mid span work ratio
5	Hub_Tip_Ratio2	Hub radius to shroud radius ratio for Rotor 1 trailing edge and Rotor 2 leading edge
6	R1_Bow	Rotor 1 bow shape parameter at leading edge to control mid span chord
7	R2_Bow	Rotor 1 bow shape parameter at leading edge to control mid span chord
8	R1 DRVT_LES	Rotor 1 incidence parameter at shroud
9	R2 DRVT_LES	Rotor 2 incidence parameter at shroud
10	R1 DRVT_LEH	Rotor 1 incidence parameter at hub
11	R2 DRVT_LEH	Rotor 2 incidence parameter at hub

Table 3 : The most important parameters from the sensitivity analysis, presented in the order of impact.

5. Optimization

The most influential parameters from the sensitivity analysis were chosen as input parameters for optimization. Optimization is done by using a surrogate model based process in which a design matrix is generated by using the DoE method. All geometries in the design matrix are then run in Ansys CFX using the CFD strategy described in 2.1 Performance, CFD for multiple operating points in both pump mode and turbine mode. The details of the optimization process based on a surrogate model and use of the 3D inverse design method is described in more details in [8]. The resulting performance parameters are added to the design matrix and then a surrogate model based on second order quadratic regression response surface method [8] is used to correlate the performance parameters with the design parameters. Finally a multi-objective Genetic algorithm (MOGA) [9] is used on the Surrogate model to find the trade off between performance parameters in pump mode and turbine mode subject to constraints.

5.1 Design of Experiments

The most influential parameters from the sensitivity analysis were chosen as input parameters and varied for the detailed design of experiments (DoE) for optimisation. The 10 less sensitive parameters are fixed at their values of the baseline design. The 11 input design parameters were distributed with the Latin Hypercube Sampling (LHS) technique across the design space to form a design matrix of 100 design points. For each design in DoE matrix, computations would be performed for three operating points in both pump and turbine mode to ensure that a wide range of operation is covered within the optimisation process. This was crucial since the final machine should have a broad range of high efficiencies, rather than a distinct peak efficiency at a certain operation point. The output responses measured included power, head, efficiency and cavitation parameters (NPSHr computed for each rotor at different operating points based on 3% drop in power/torque in pump mode by using the single phase CFD results). Out of 100 design points in the design matrix, 95 converged designs were generated using TURBOdesign1 and were simulated with CFD for three operating points in each mode. To summarise, the data produced from the DoE consisted of a matrix of size $95_{design\ points} \times 6\ (i + o)$, where i and o are the number of input parameters and output responses respectively for each design analyzed.

5.2 Surrogate Model and Optimisation

With the data from the DoE the optimisation process is commenced. First, the response surface model is used to create a surrogate model that correlates the various performance parameters in pump mode and turbine mode to the design parameters [8]. In this study, a design exploration tool by Dassault Systemes called ISIGHT is used to perform the optimization, as well as Design of Experiment and response surface model. Input parameters and output responses associated with all the design points (95 designs from DOE) are arranged in a table format. Initially a Surrogate Model approximation is performed on the data using linear regression response surface model. Then optimization method based on a Non-dominated Sorting Genetic Algorithm – II (NSGA-II) [9], with 100 X 100 (population size X generations) is used to explore the design space, using the surrogate model only. A few best design candidates from optimization are chosen from among Pareto points and are simulated in CFD to assess performance improvements.

5.3 Performance Parameters

The main aim of this optimisation is to improve the overall efficiency of the CRPT while maintaining the required head. The efficiencies are to be maximized in each mode of operation, but more importantly the combined pump-turbine mode of operation should guarantee a very good efficiency. To achieve this, the power consumption in pump mode of operation must be minimized and the power produced in turbine mode of operation must be maximized. Hence it is logical to use power in each mode of operation across the range of operation as optimisation objectives. So the objectives used for the optimization are as stated below:

- ➢ Maximize power produced in turbine mode for all the 3 operating points
- ➢ Minimize power consumed in pump mode across all the 3 operating points

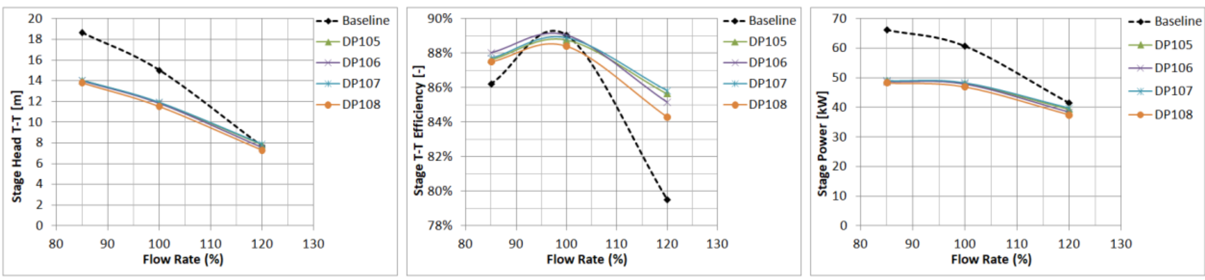
It is to be ensured that the designs satisfy the minimum head requirements in pump mode for the lab test. Considering the final details available for the test facility at TU Braunschweig, the pump is required to deliver a minimum head of 7 [m] to cover for the basin level differences and the losses in the pipes. Hence the minimum head requirements are specified as a constraint. Also the cavitation performance as assessed by the NPSHr parameter is to be better for the new designs compared to the baseline. Hence NPSHr is mentioned as a constraint. The constraints used in the optimization are as stated below:

- ➢ NPSHr, for both Rotor1 and Rotor2, at 3 different flow rates in pump mode with an upper limit same as that of baseline design
- ➢ A minimum head at each operating point in pump mode

Optimization process in the end produces an optimized candidate in addition to a set of points representing the Pareto front of optimization. A CFD simulation is carried out for 4 of the chosen designs, which are selected from the Pareto front.

5.4 Optimised Design

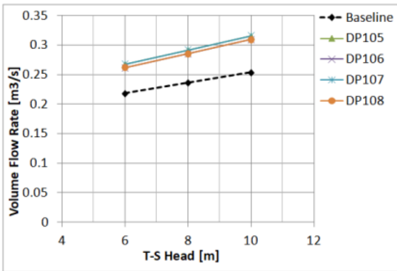
The performance comparison of the optimized designs selected from the optimisation process with baseline design in pump mode is presented below, in Figure **5**. The designs shown as DP105, DP106, DP107 and DP108 represent designs obtained from the optimisation process. The average stage efficiency between the 3 operating points improved by up to 2.6% points for the best performing candidate. As a result of the power minimization objective, the power is dropped considerably for the optimized designs compared to the scaled baseline (Section 3. Scaling to model scale) at lower flow rates. The minimum head requirement of 7 [m] is met at the 120% flow rate by all the designs. NPSHr values computed also showed improvements over the baseline.



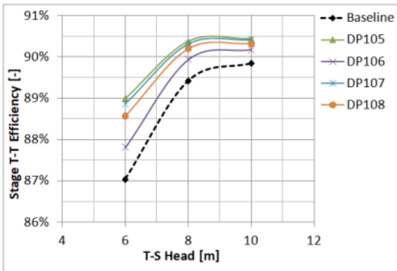
a) Stage Head b) Stage Efficiency c) Stage Power

Figure 5 : Comparison of the Optimised Designs versus the baseline design - Pump Mode

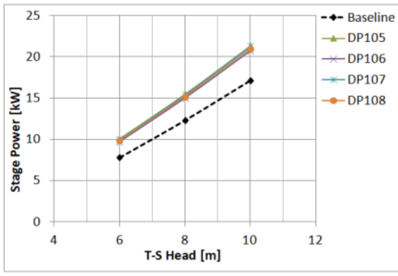
The performance comparison in turbine mode is presented below in Figure **6**. The average stage efficiency between the 3 operating points improved by up to 1.1% points for the best performing candidate. As a result of the power maximization objective, the power is increased considerably for optimized designs compared to the baseline. Though the total head remained the same, the increased flow rate helped in increasing the power output.



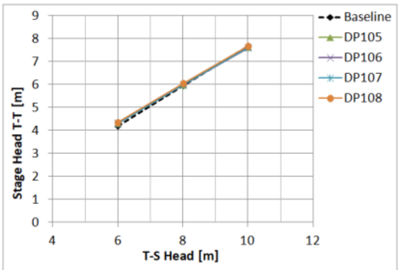
a) Volume flow rate



b) Stage Efficiency



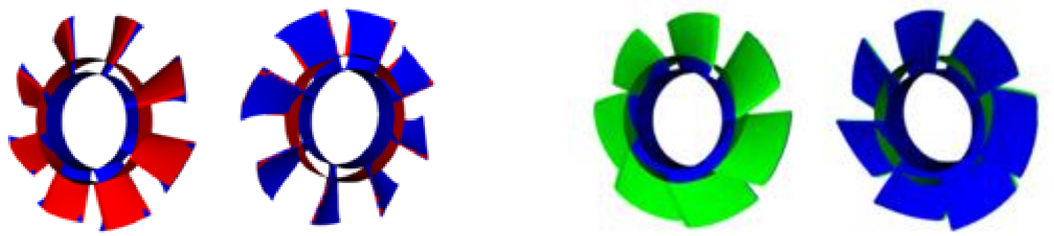
c) Stage Power



d) Stage total head

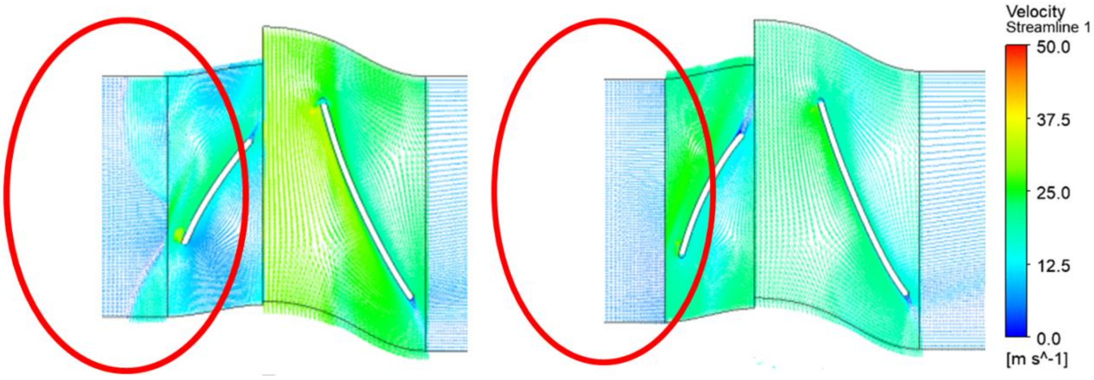
Figure 6 : Comparison of the Optimised Designs versus the baseline design - Turbine Mode

Considering both modes of operation, the design DP107 gave the most efficiency improvements with ~2.6% points increase in average pump mode efficiency (average of 3 operating points) and ~1.1% points increase in average turbine mode efficiency. A comparison of geometries, flow field and other plots are given below, where evidence of improvements can be seen. As seen in Figure 7, DP107 design has a lower hub to tip ratio. Figure 8 shows the velocity vector plot comparison of both designs at 85% flow rate in pump mode; here it can be clearly seen that the inlet flow recirculation region is eliminated for the new design. From the surface pressure plot comparison at 95% span at 100% flow rate in pump mode in Figure 9, it can be inferred that the low pressure regions are having a higher pressure for DP107 than baseline design and hence a lower risk of cavitation. The flow angle plot comparison in turbine mode at 8 [m] total to static head in Figure **10** shows that the exit flow angle from the contra-rotating stage in turbine mode is close to zero for the DP107 design and hence more work is extracted from the optimised stage in comparison to baseline design.



a) Rotor1 b) Rotor 2

Figure 7 : Comparison of Design DP107 and baseline rotor geometries (DP107 shown in blue).



a) Baseline b) DP107 Design

Figure 8 : Vector plots at 95% span location at 85% flow rate.

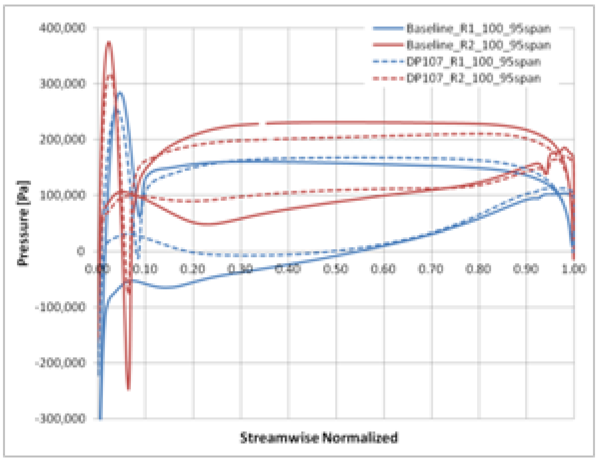


Figure 9 : Surface pressure plot at 95% span at 100% flow rate in pump mode (solid lines - Baseline, dotted lines - DP107; Blue - Rotor1, Red - Rotor2).

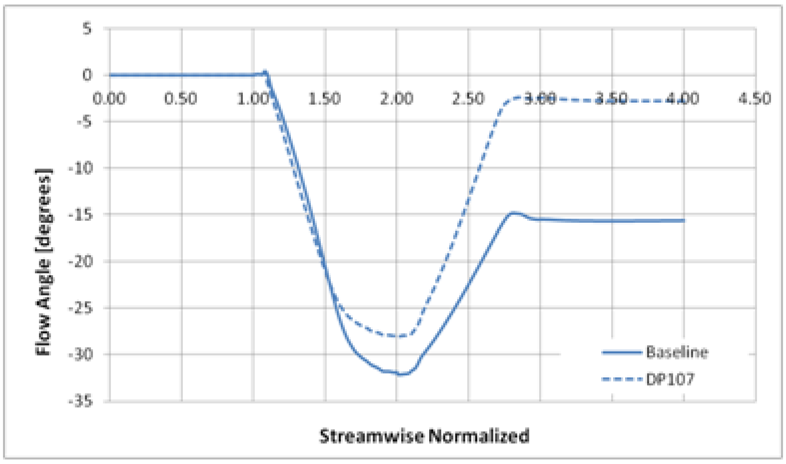


Figure 10 : Variation of Flow angle through the contra-rotating stage- Turbine mode 8m T-S head condition. (Solid line baseline, Dashed Line DP107).

6. Conclusions

The initial prototype design showed a great potential with a hydraulic efficiency of about 90% for a wide range of operating conditions on both pump and turbine modes. The results are very promising given that the initial design is made before any optimisation is applied to the blade geometry.

The scaling from prototype to model scale showed that the characteristics of the efficiency are maintained, but the values are slightly lower in model scale. The reason for this is that it is in practice very difficult to scale all parameters of interest to match between prototype and model scale.

Hence it was decided to carry out an optimization of the model scale pump/turbine to be used for model testing. A two stage optimization process was used. In the first stage, sensitivity analysis showed that out of 21 design parameters, 11 are deemed as more important. The 11 more important parameters were included in the full optimization.

The optimisation procedure is divided into two stages. The first stage is to generate a design of experiment (DoE) that fills the design space in an efficient manner. Hence 95 design points are evaluated at 3 operating points in pump and turbine modes respectively to create a trustworthy response surface. Multi-objective Genetic Algorithm was then run on the response surface to maximise turbine power and minimise pump power subject to appropriate constraints on cavitation performance and head requirement. The optimisation algorithms suggested a number of favourable designs. Out of these 4 designs are chosen and the performance is verified by steady CFD in both pump mode and turbine mode. . Finally the most suitable candidate (DP107) was chosen given its superior performance both in pump mode and turbine mode. This optimised design shows ~2.6% points increase in average pump mode efficiency and ~1.1% points increase in average turbine mode efficiency. In addition it shows improvement in cavitation performance. This design will be used for experimental validation at TU Braunschweig.

7. References

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