Optimized initial down-scaled model design of RD-CRPT

D2.2.

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Executive Summary

This report describes how the optimised initial down-scaled model design of the Rim-Driven Counter-Rotating Pump-Turbine (RD-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 manual optimisation process was conducted in an iterative manner. The goal with the optimisation procedure was to increase the efficiency for a wide range of operating conditions in both pump and turbine modes. The process has resulted in a design of a down-scaled model with far better performance than the initial design. The RD-CRPT design is ready to be subjected to further numerical sensitivity analysis and multi-objective optimisation. It has been decided that it is the Shaft-Driven Counter-Rotating Pump-Turbine (SD-CRPT) that will be experimentally studied, rather than the RD-CPRT due to the fact that the Rim driven configuration cannot use “off the shelf” motor/generators components.

1. Introduction

ALPHEUS deliverables D2.1 (a parallel report) and D2.2 (the present 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 model scale design. Furthermore, 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 confirm the 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, with both a sensitivity analysis and a full multi-objective optimisation.

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 RD-CRPT at prototype scale 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 was used to derive the initial design of the Rim-Driven Counter-Rotating Pump-Turbine (RD-CRPT) in prototype scale. The software uses a meanline code for initial sizing and then a 3D inverse inviscid inverse design method to design the 3D blade shape for a given distribution of blade loading (proportional to static pressure jump across the blade). 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 use in the inverse design method with appropriate choice of blade loading to generate the first and second rotor 3D blade geometries.

1.2 Scaling Parameters

The prototype design was derived 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 laws assume 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

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

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

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

Here Q is the flow rate, Ω is the rotational speed of the runner, 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 Design Process

Starting from the scaled model of the initial prototype design, changes were made to the important input parameters of the inverse design method such as the spanwise $rV\theta$ distribution which equates to the swirl or work produced across the rotor, the streamwise blade loading and axial chord distribution across the span to create new designs for the contra-rotating stage. These new designs were then evaluated by CFD and the results of this manual "design optimization" process are described in this report.

2. Initial Design

The initial design of the RD-CPRT was derived with ADT's TURBOdesign suite [1] at prototype scale. The software uses a 3D inverse design method [2], [3] and [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. Figure 1 shows the geometry of the initial contra-rotating stage designed by TURBOdesign Suite. Note that Rotor1 has 8 runner blades and Rotor2 has 7 runner blades.

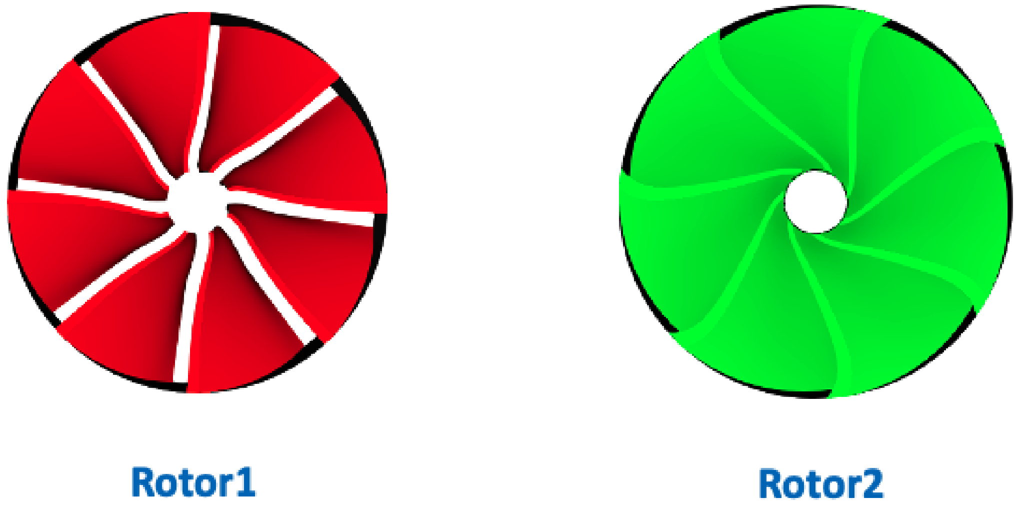


Figure 1 : Geometry of Initial RD-CRPT designed at Prototype scale

Table 1 summarises the dimensions and design parameters used to create the initial design of RD-CRPT by using TURBOdesign Suite. As in the case of the shaft driven design the design flow rate is 130 m^3/s in pump mode and the two runners rotate in opposite directions. The two runners have the same rotational speed of 50 RPM and a constant thickness of 60 mm is used for both runners.

Meanline Design Details	Rim Driven
Flow rate [m3/s]	130
Rotor1 speed [rev/min]	50
Rotor2 speed [rev/min]	-50
Hub Diameter [mm]	1000
Shroud Diameter [mm]	5939
Maximum rotor1 axial span [mm]	983

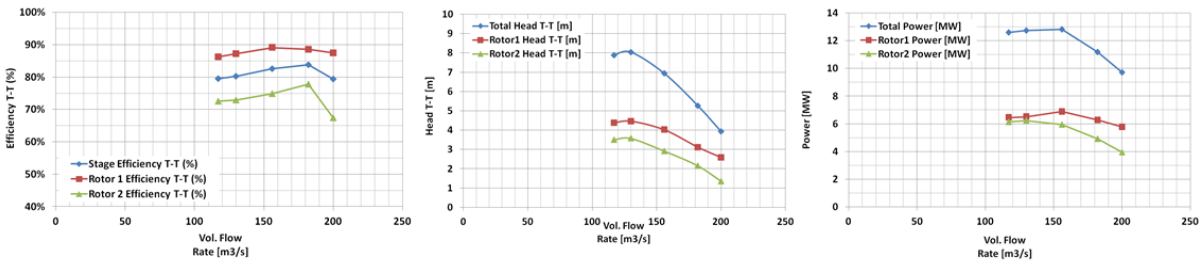
Maximum rotor2 axial span [mm]	875
Minimum axial gap between rotors [mm]	245
Rotor1 blade count	8
Rotor2 blade count	7

Table 1 : Main design parameters for the RD-CRPT Stage

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 operation points to evaluate the operation range of the machine. Commercial CFD code ANSYS CFX was used for all computations [5]. The CFD model utilises a full 360° domain with the complete runners and a stage interface approach between the rotating runners. The k-ω SST eddy viscosity model was used to account for turbulence, and the computation mesh contain roughly 29×10^6cells. In pump-mode the flow rate was specified at the inlet, and in turbine-mode the total pressure was used at the inlet.

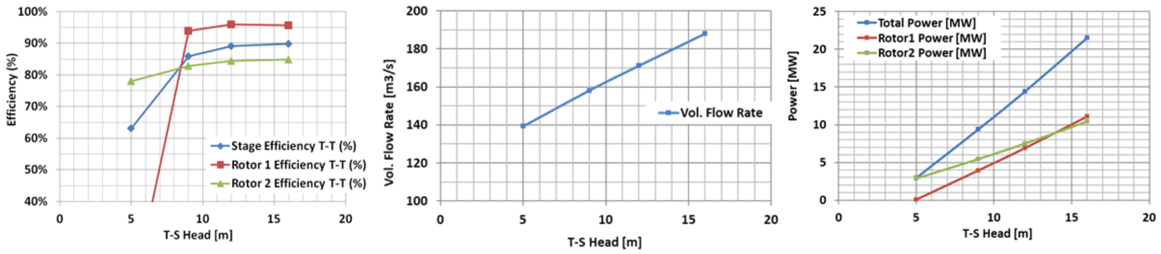
Figure 2 shows the computed efficiency, head, and power in pump mode as a function of flow rate for the initial design in prototype scale. The results depicted in Figure 2 (a) shows that the efficiency has a flat characteristic and is fairly equal across the operational range. The peak efficiency in pump mode for the machine is almost 84 %. The head and power from Figure 2 (b) and (c) shows 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 : Performance of initial design for CR-RDPT - Pump mode

The results for turbine mode are shown in Figure 3. The results for turbine are shown as a function of Total-static head. The results indicate that the machine has a fairly constant efficiency of around 90% between 9-16 m head but the efficiency drops rapidly below 9 m, see Figure 3(a). The volumetric flow rate and power, depicted in Figure 3 (b) and (c), show that both the power and flow rate increase linearly with the increase of head.



a) Efficiency b) Flow rate c) Power

Figure 3 : Performance of initial design for CR-RDPT – Turbine mode

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) analysis 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. 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 shroud for both the runners, and the maximum value is 154 MPA. This means that the safety factor to the yield strength is above 6.6, 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 % 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 % 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 was 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 m. The head includes both the height elevation of 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 3.1 for pump mode in a), and turbine mode in b).

Pump Mode		
	R1	**R2**
D_{shroud}[mm]	276	276
N_m[rev/min]	1155.55	1155.55
Q_m[m3/s]	0.3615	0.3615
P_m[kW]	18.392	15.888
H_m[m]	4.634	3.366

a) Pump Mode

Turbine Mode		
	R1	**R2**
D_{shroud}[mm]	276	276
N_m[rev/min]	776.4	776.4
Q_m[m3/s]	0.266	0.266
P_m[kW]	5.58	6.09
$H_{m\ total\text{-}total}$[m]	2.23	2.77

b) Turbine Mode

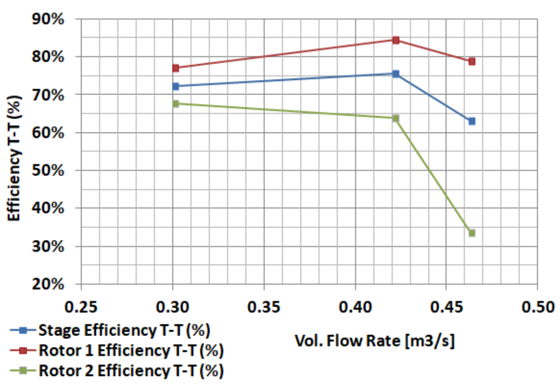
Table 2 : scale parameters of the initial RD-CRPT. Note that R1 and R2 denotes Rotor1 and Rotor2.

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 1156 RPM in pump mode, Table 2 (a), and 776 RPM in turbine mode, Table 2 (b).

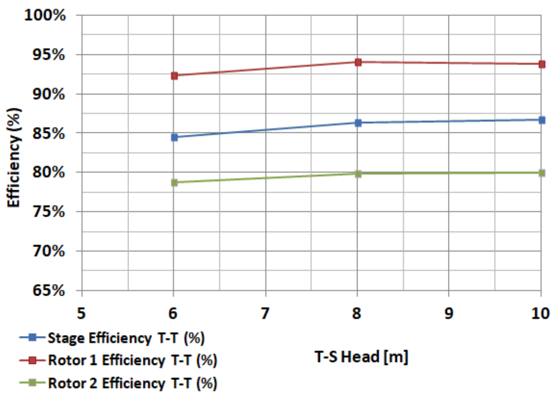
3.1 Performance in model scale

Before the manual design 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 total-total 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 (a), and in model scale, Figure 4 (a), it is found that the performance has been affected by the scaling. A significantly lower peak efficiency of 76% is predicted in model scale, versus 84% in prototype scale. The scaling effects are less drastic in turbine mode, comparing Figure 3 (a) and 4 (b). The characteristics in model scale are recognised in turbine mode, but a decrease of nearly 5 percentage points in efficiency compared to prototype scale is noted for the whole operation range.



a) Pump Mode



b) Turbine Mode

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

4. Manual design optimisation

The RD-CRPT was optimised in an iterative manner to increase the efficiency of the machine. No less than five full blade designs were carefully evaluated in both pump and turbine modes of operation for 3 design points covering a wide range. The blade geometries were generated in series with ADT's TURBOdesign suite, successive blade design decisions were based on the findings from the CFD results of the previous designs. Significant improvements in efficiency could be attained by this optimization process. A few of the notable iterations are discussed below.

4.1 Some Notable Designs

Design03: First main design modification performed is on the meridional geometry where it is made symmetric and the hub chord is reduced for both the rotors. The rotor work coefficient and blade loading are adjusted to keep the diffusion ratio (ratio of maximum velocity over the blade surface to the trailing edge velocity for different streamlines) from hub to shroud below 2.

These changes helped in improving the stage total to total efficiency in both pump and turbine modes as shown in the efficiency plot comparison in Figure 5. It can be seen that in pump mode the stage efficiency increased by ~2% points across the flow range, but the second rotor is performing worse than the baseline. In turbine mode a single operating point is simulated and it is seen that the Rotor2 performs better whereas the Rotor1 shows a drop in efficiency; the stage efficiency is improved by around 2% points. From the head comparison in Figure 5, it is seen that the Rotor2 is generating much less head. Hence the next design iteration is performed to generate more work from the second rotor.

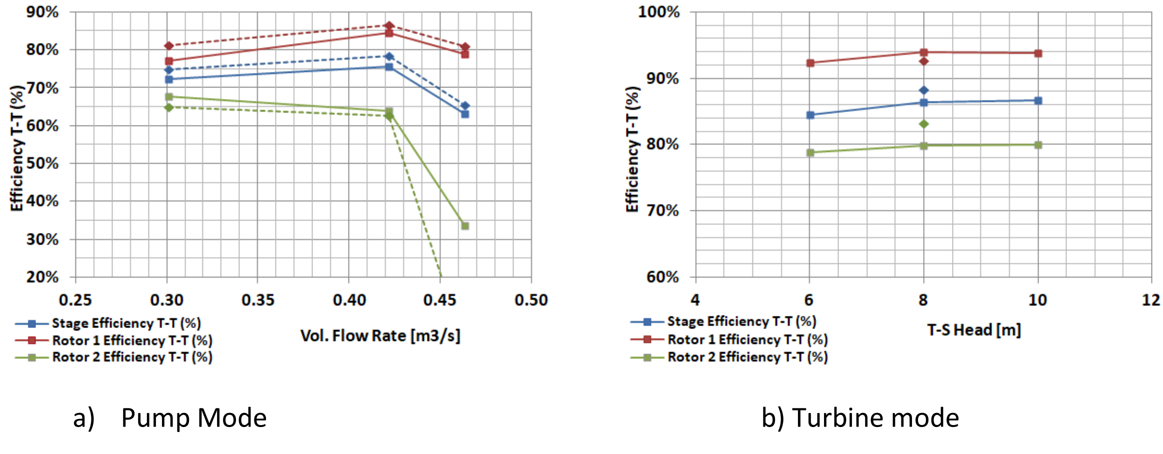


Figure 5 : Baseline (solid lines) vs Design03 (dotted lines, diamond marker sign) total to total Efficiency

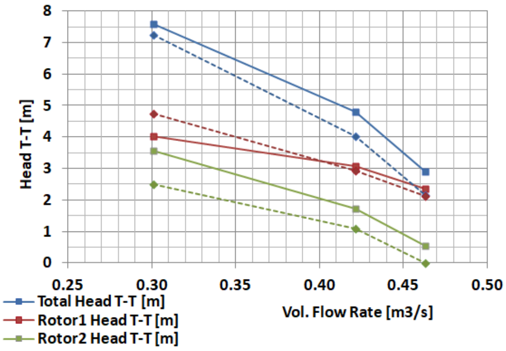


Figure 6 : Head in pump mode Baseline (solid lines) vs Design03 (dotted lines)

Design04: For this design, the Rotor2 chord is further modified. In addition, the non-dimensional work coefficient given by

$$rVt^* = \frac{W}{\omega R_{ref} U_{ref}}$$

where ω is the rotational speed, R_{ref} the reference radius, U_{ref} the reference velocity and, W represents the rotor specific work is adjusted for the second rotor to generate more head. Rotor 1 has a reduction in work coefficient of around 10% towards the shroud, while it remains the same from mid span towards the hub. As shown in Figure 7, these modifications helped in improving the pump stage efficiency by ~4% over the baseline and with Rotor2 performing better than baseline across the low flow points. Also in turbine mode a stage efficiency improvement of greater than 2% points is seen; Rotor1 of the new design is performing as well as the baseline.

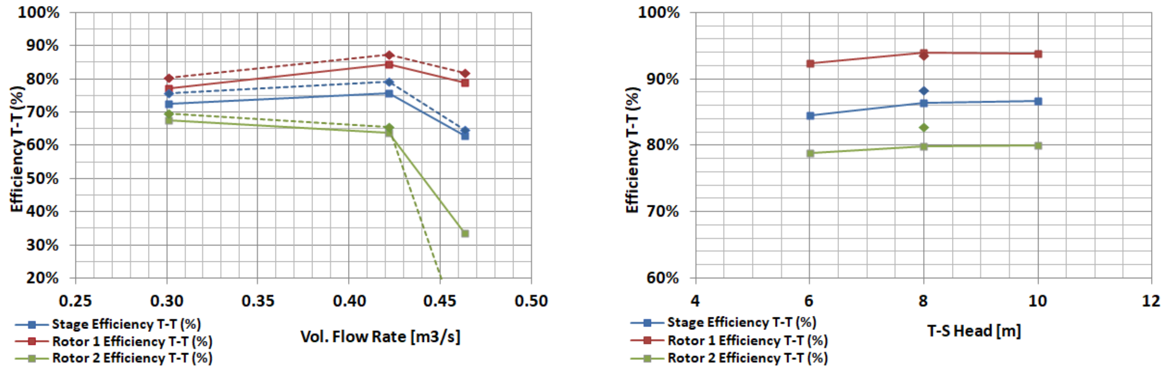


Figure 7 : Baseline (solid lines) vs Design03 (dotted lines, diamond marker sign) total to total Efficiency

4.2 Final Optimized Design

From the analysis of the previous design iterations, a good improvement in stage efficiency is seen in both modes of operation though the second rotor is limiting the performance especially at high flow in pump mode. The second rotor is modified to increase the throat area by ~11%. This helped in shifting the peak efficiency of the second rotor to a higher flow rate in line with the first rotor and has helped in improving overall stage performance. The detailed performance plot comparison of the final design with the baseline design is presented in Figure 8 and Figure 9. A significant improvement in performance is noted in both pump and turbine mode across the operating ranges analysed. Compared with the baseline, the second rotor contributed the most towards improvement in both modes of operation. First rotor showed improvement in pump mode over the baseline while the baseline design has slight advantage in turbine mode; nevertheless the efficiency is still at a very high level for the Rotor1 of the optimized design.

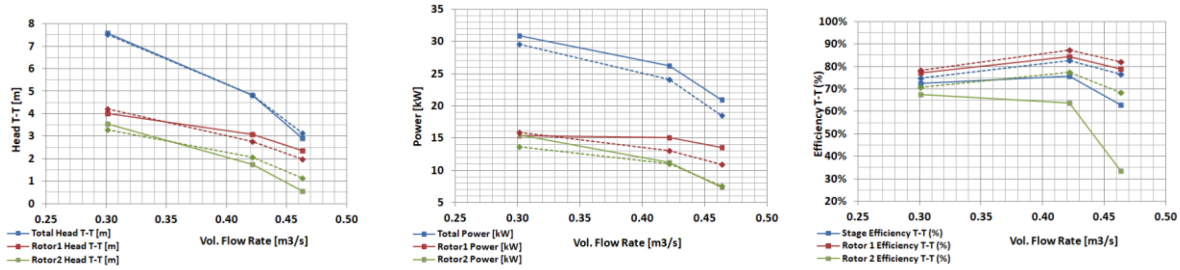


Figure 8 : Performance curves in pump mode - Baseline (solid lines) vs Optimized design (dotted curve)

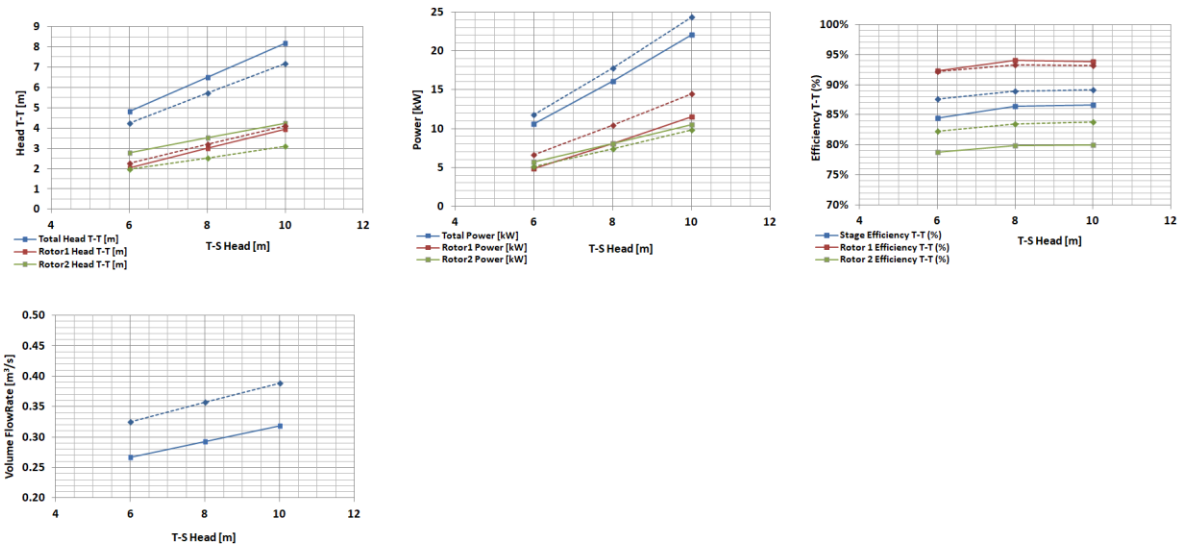


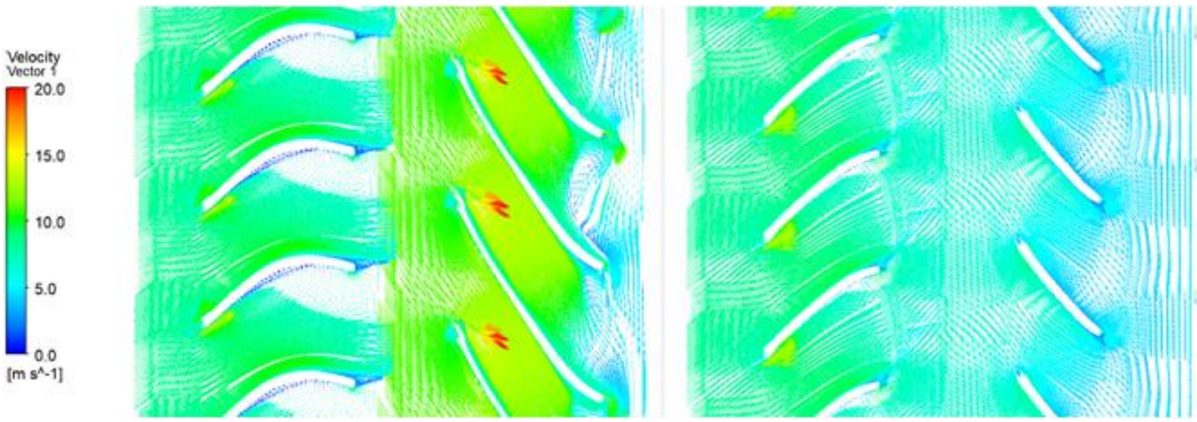
Figure 9 : Performance curves in turbine mode - Baseline (solid lines) vs Optimized design (dotted line)

The stage efficiency is largely improved by the optimisation procedure. The average efficiency across the operational range is increased by more than 7.7 percentage points in pump mode, and 2.8 percentage points in turbine mode. The flatness of the efficiency curves also provide a better characteristics of the machine since one of the main goals of the ALPHEUS project is to produce a machine that has a broad operational range rather than a peak operation point. The efficiency curve in turbine mode is almost completely flat with an efficiency close to 90% for heads of 6 - 10 m. Pump mode also did not show much drop from the peak efficiency across a wide range.

A comparison of the optimized design geometry with baseline is shown in Figure 10.

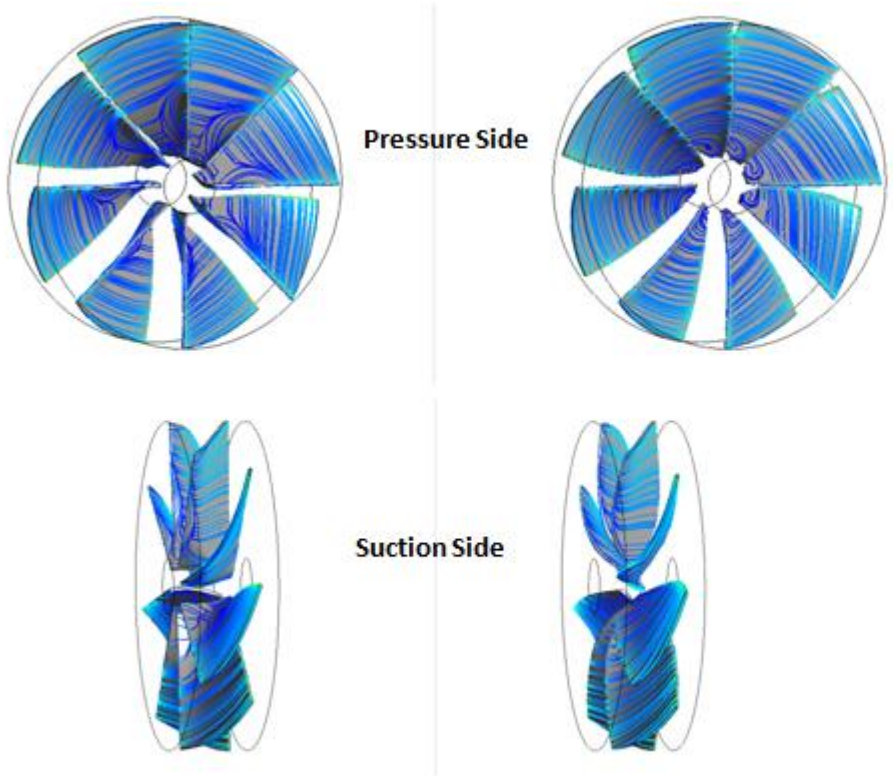
Figure 10 : Geometry comparison - Optimized Design (blue color) vs Baseline

A comparison of velocity vector plot at 15% span at 420 kg/s in pump mode in Figure 11 shows that separations and recirculations downstream of each rotor is considerably reduced. At higher span locations as well optimized design has a better flow incidence compared to baseline. Surface streamline plots in Figure 12and Figure 13for Rotor1 and Rotor2 shows that the streamlines are well aligned for the optimized design and the influence of flow recirculations at lower span is much lower compared to baseline. The suction side separation on the Baseline Rotor1 at mid locations - indicated by the white, low velocity region in Figure 11(a) - is also absent in the new design. For the Rotor2, the metal surface area is also much lower which helps in reducing profile losses.



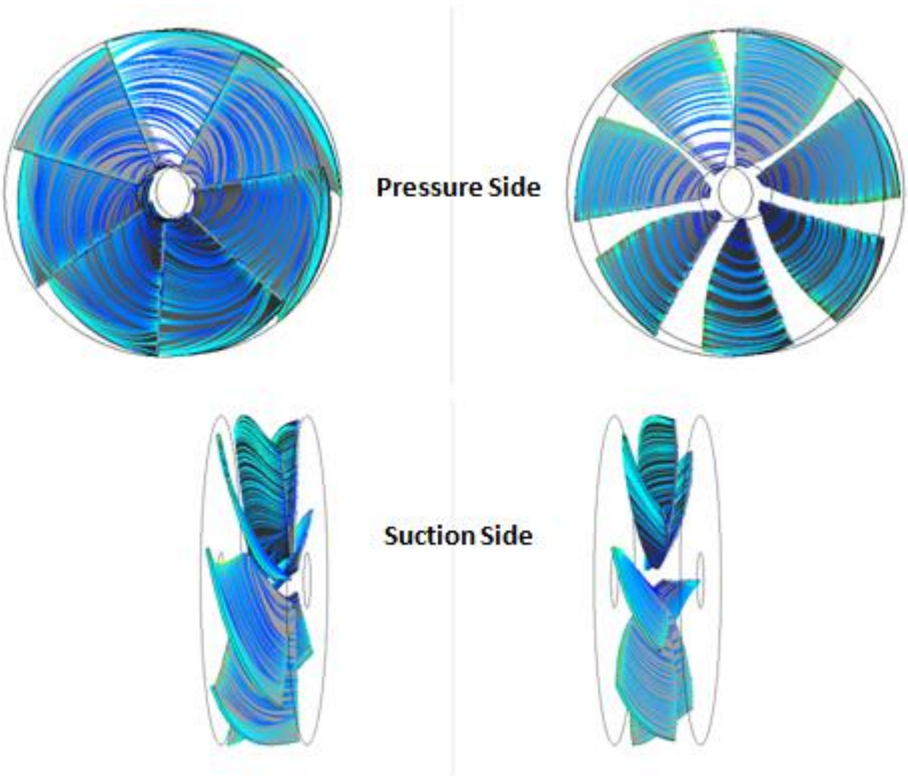
a) Baseline b) Optimised Design

Figure 11 : Velocity vector plot at 15% span (close to hub) at 420 [kg/s] in pump mode



a) Baseline b) Optimised Design

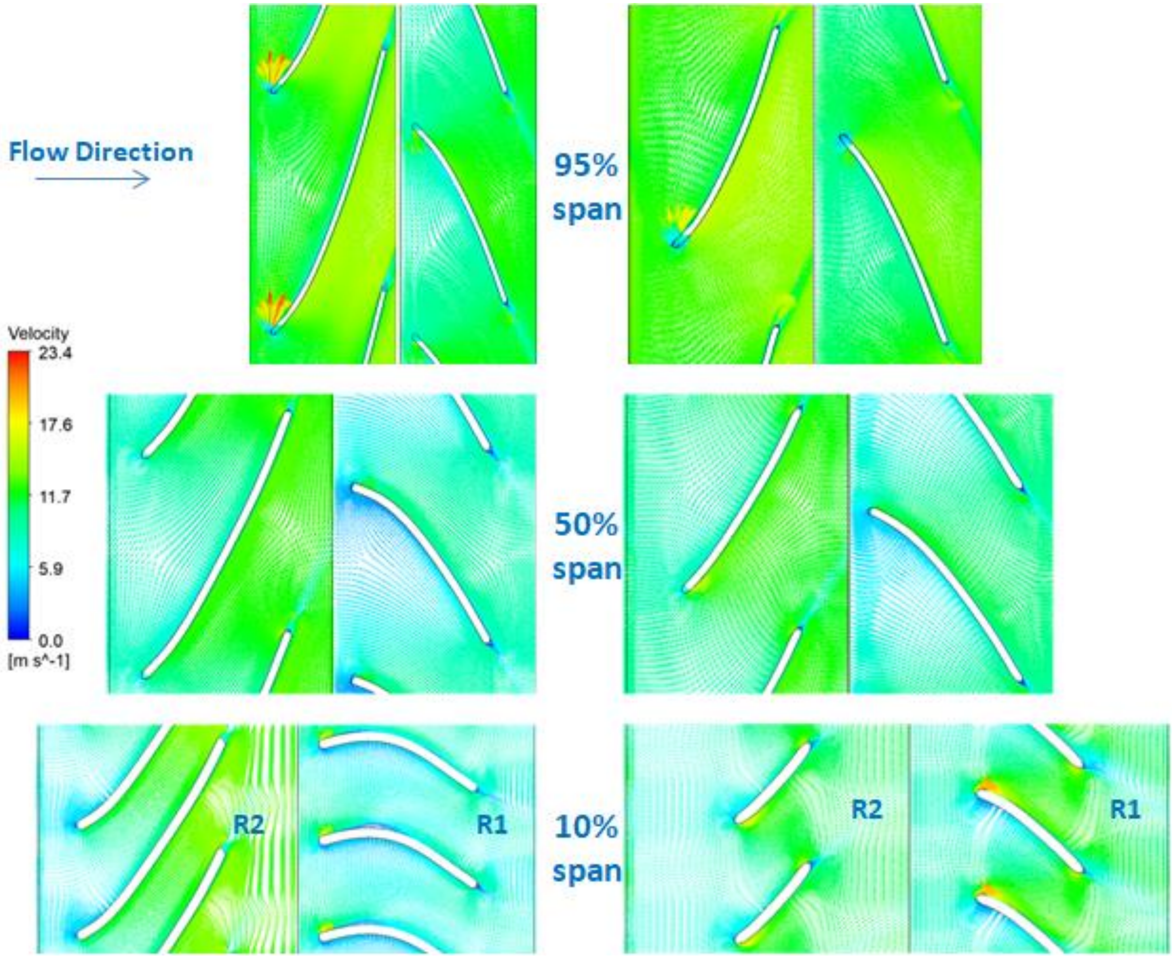
Figure 12 : Surface streamline plot at 420 [kg/s] in pump mode for Rotor1



a) Baseline b) Optimised Design

Figure 13 : Surface streamline plot at 420 [kg/s] in pump mode for Rotor2

In turbine mode, flow is well attached for both rotors of the baseline as well as optimized geometry. Comparison of vector plots in turbine mode at a total to static head condition of 8 [m] is shown in Figure 14. Here flow incidence seems to be better especially for the rotor1 of the optimized design at lower span and rotor2 of the optimized design at higher span. The main contribution towards efficiency improvement comes from Rotor2 where the reduced wet surface area helped in reducing profile losses.



a) Baseline b) Optimised Design

Figure 14 : Velocity Vector plots at a total to static head of 8[m] in turbine mode

An investigation is also carried out for the optimized design with a hub wall of infinitesimal thickness (Figure 15) where fluid can pass through the hub to see its impact on controlling the recirculation in the centre. But as shown in comparison plots in Figure 16, efficiency is reduced for this configuration.

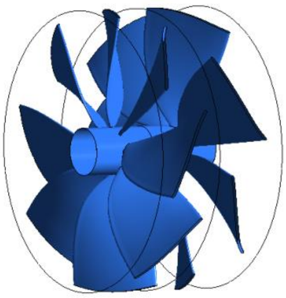
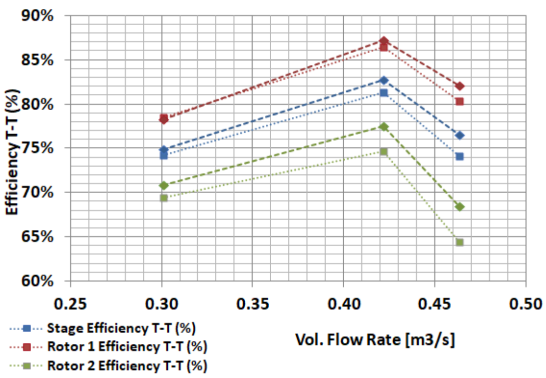
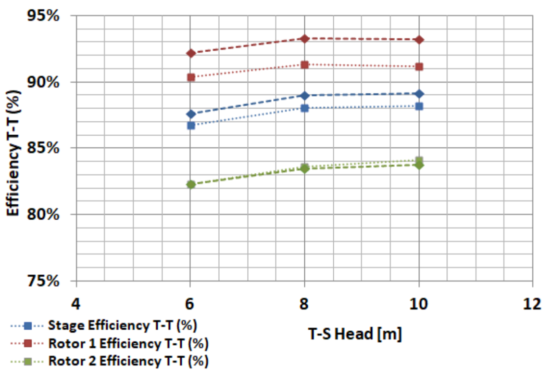


Figure 15 : Optimized Design with a hub wall





a) Pump Mode b) Turbine mode

Figure 16 : Optimized Design without a hub wall (dotted line, diamond marker sign) vs with hub wall (dotted lines, square marker sign) - total to total Efficiency

5. Conclusions

The initial prototype design showed promising potential with a peak hydraulic efficiency of almost 90% in turbine mode and 84% in pump mode for a wide range of operating conditions in both modes. The results are 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 is maintained, but the values are lower in model scale, significantly so for the pump mode. 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.

The optimisation procedure is conducted based on multiple blade design iterations performed in succession, a later design is based on findings from the CFD result analysis of the previous designs. The final design has a flatter efficiency characteristic than the initial design in both pump and turbine modes across the flow range analysed. The average efficiency across the whole operation range is increased by ~7.7 percentage points in pump mode and ~2.8 in turbine mode. This provides an excellent stand point to continue investigating of the RD-CRPT to find the most sensitive design parameters and perform full multi-objective optimisation to increase the round trip efficiency even further.

6. References

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