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The Influence of the Profile of the Blades of the Centrifugal Pump on its Energy Parameters

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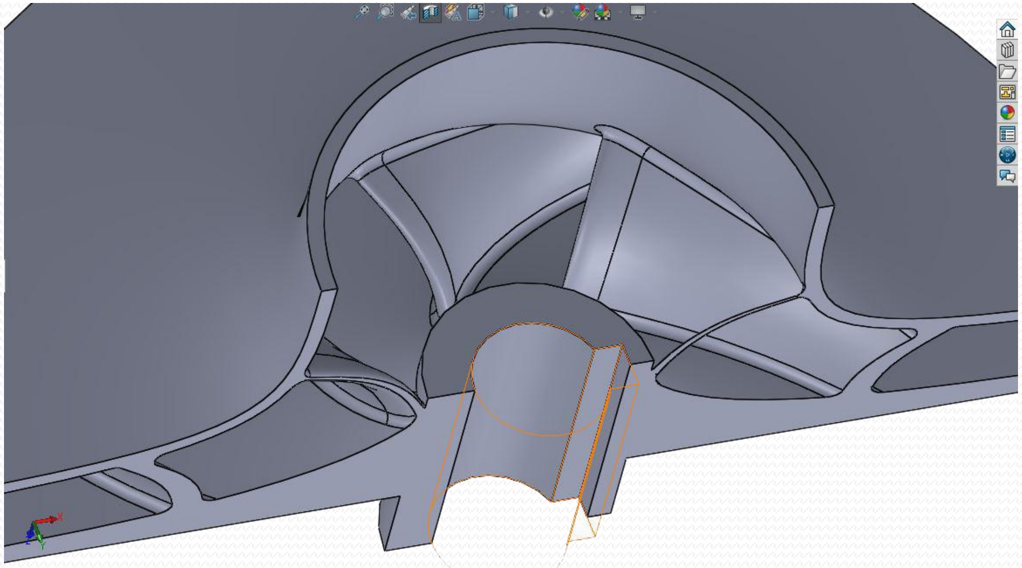
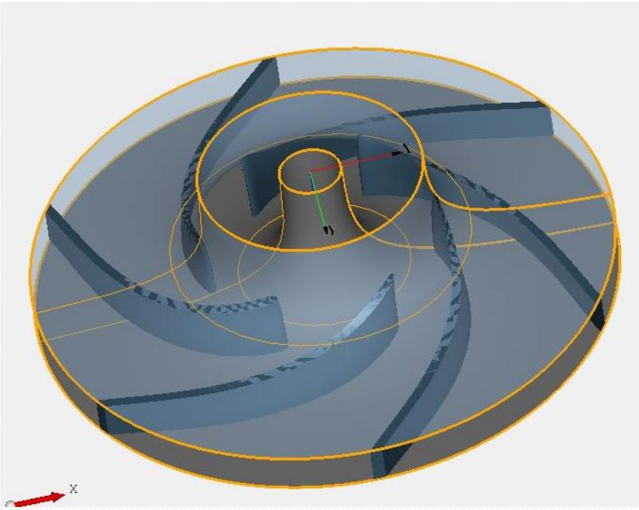
- The purpose of this paper has been the study of the influence of the blade profile of the centrifugal pump on its energy parameters. For this we have built a mathematical model that describes the passage of the fluid in the pump rotor.





- Our mathematical model consists of:

Bernoulli's equation:

$$p_1 + \frac{1}{2} \cdot \rho \cdot v_1^2 + \rho \cdot g \cdot h_1 = p_2 + \frac{1}{2} \cdot \rho \cdot v_2^2 + \rho \cdot g \cdot h_2 \qquad [1]$$

To describes the conservation of energy along fluid flow.

Continuity equation:

$$A_1 \cdot v_1 = A_2 \cdot v_2 \qquad [2]$$

Moment of Force Equation (Euler's Theorem for pumps):

$$T = \dot{m} \cdot r_2 \cdot (v_{u2} - v_{u1}) \qquad [4]$$

Momentum equations (Navier-Stokes):

For the x direction:

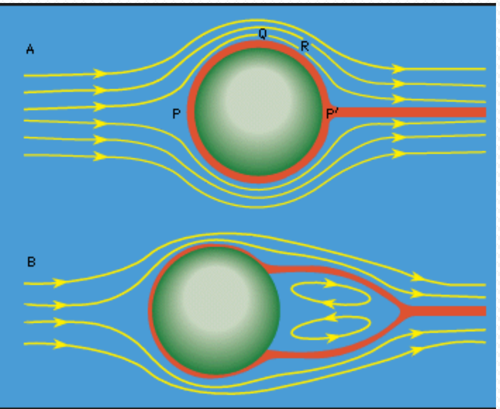
$$\rho\left(\frac{\partial u}{\partial t}+u\cdot\frac{\partial u}{\partial x}+v\cdot\frac{\partial u}{\partial y}+w\cdot\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)+f_x \qquad [4]$$

For the y direction:

$$\rho\left(\frac{\partial u}{\partial t}+u\cdot\frac{\partial u}{\partial x}+v\cdot\frac{\partial u}{\partial y}+w\cdot\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)+f_y \qquad [5]$$

For the z direction:

$$\rho\left(\frac{\partial u}{\partial t}+u\cdot\frac{\partial u}{\partial x}+v\cdot\frac{\partial u}{\partial y}+w\cdot\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)+f_z \qquad [6]$$



$$\underbrace{\rho}_{\text{Mass}}\cdot\underbrace{\left(\frac{\partial\vec{V}}{\partial t}+(\vec{V}\cdot\nabla)\vec{V}\right)}_{\text{Acceleration}}=\underbrace{\rho\vec{g}}_{\text{Body Force}}\underbrace{-\nabla p+\mu\cdot\nabla^2\vec{V}}_{\text{Surface Force}}$$

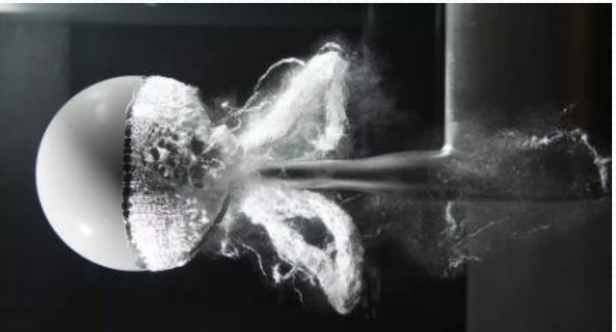
Energy Equation for Centrifugal Pumps:

$$H = \frac{U_2^2 - U_1^2}{2g} \qquad [7]$$

Specific Speed Relation:

$$n_s = n \cdot \sqrt{\frac{Q}{H^{\frac{3}{4}}}} \qquad [8]$$

The minimum NPSH value for the pump to operate without cavitation relation:

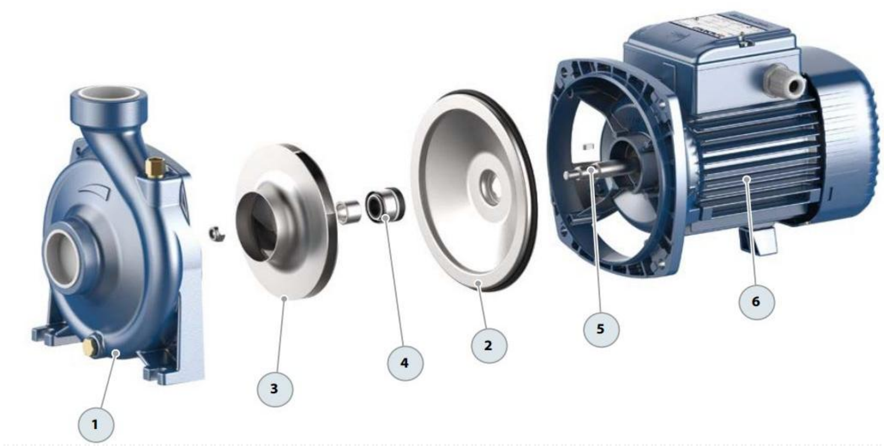


$$NPSH_r = \frac{(P_s - P_v)}{\rho \cdot g} \qquad [9]$$

$$NPSH_a = \frac{(P_{atm} - P_{loss}) + \rho \cdot g \cdot h_s - P_v}{\rho \cdot g} \qquad [10]$$

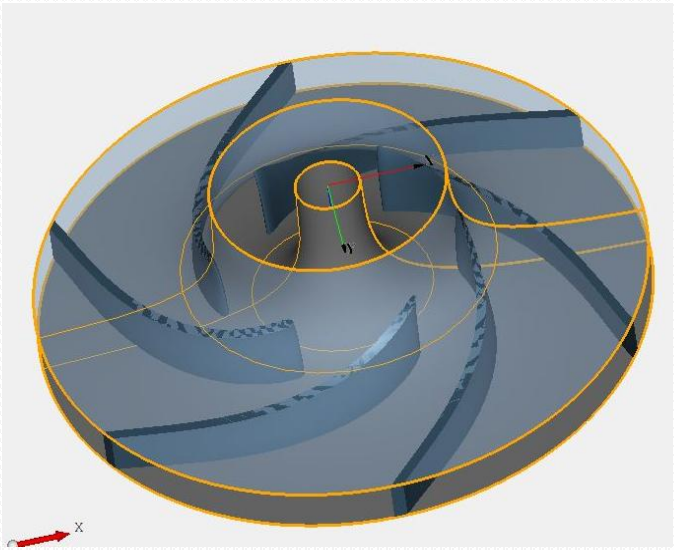
Modeling and Simulation

Through the program CFTurbo and SolidWorks we have remodeled the pump taken in the study



The pump taken into consideration is a ***Pedrollo centrifugal water pump*** of the ***HFm 5AM*** type with the maximum flow ***Q=500 l/min*** and Total ***manometric head H = 22.5 m***.

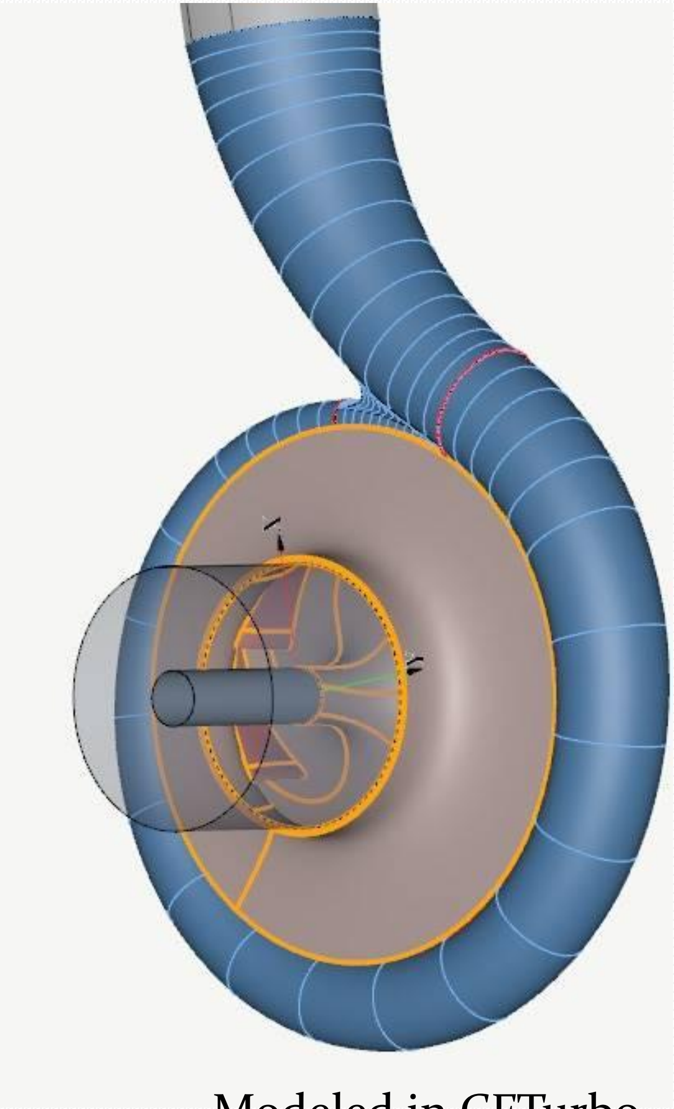
Original Rotor of pump



The same rotor modeled in CF Turbo

After the construction in CFturbo, we transferred the 3D model of the entire pump to the Symerix-PumpLinx program in order to perform the CFD analysis of the pump's performance with the current parameters.

Original Volute of PUMP

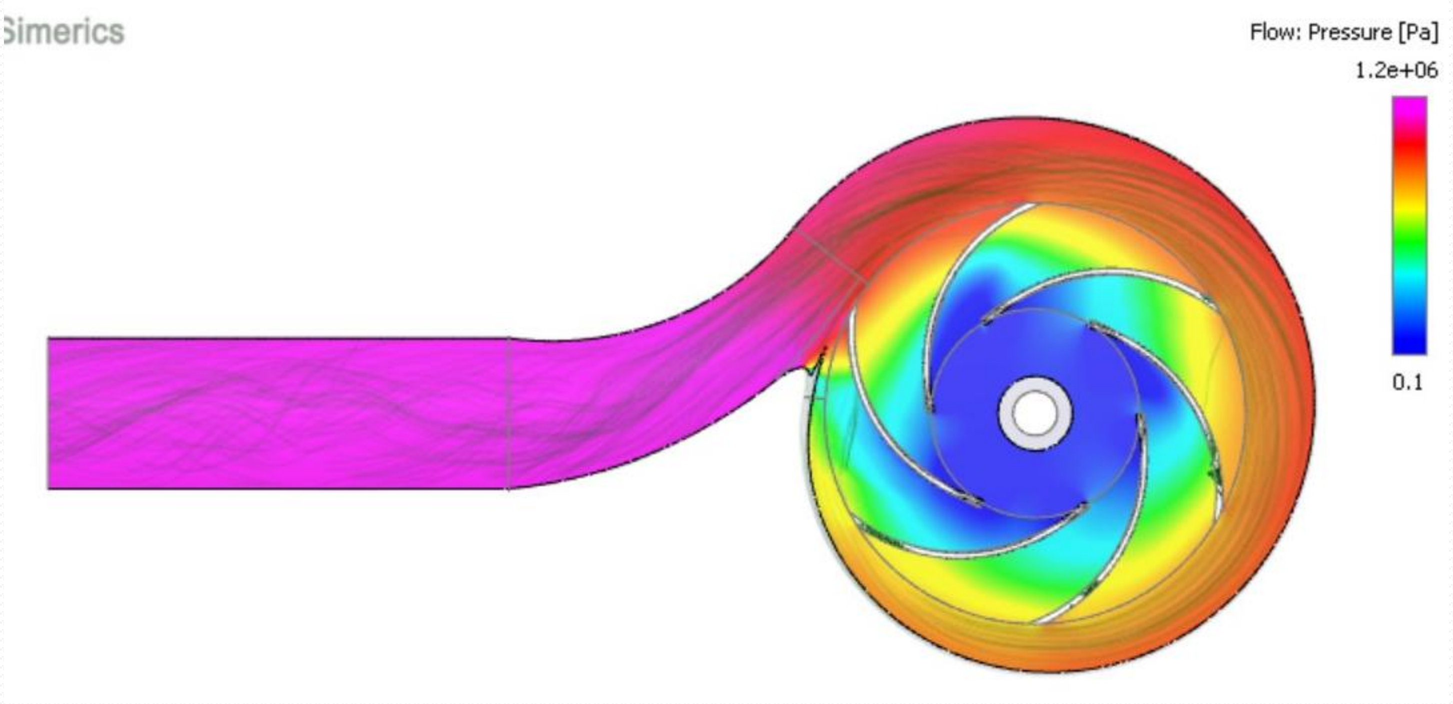


Modeled in CFTurbo

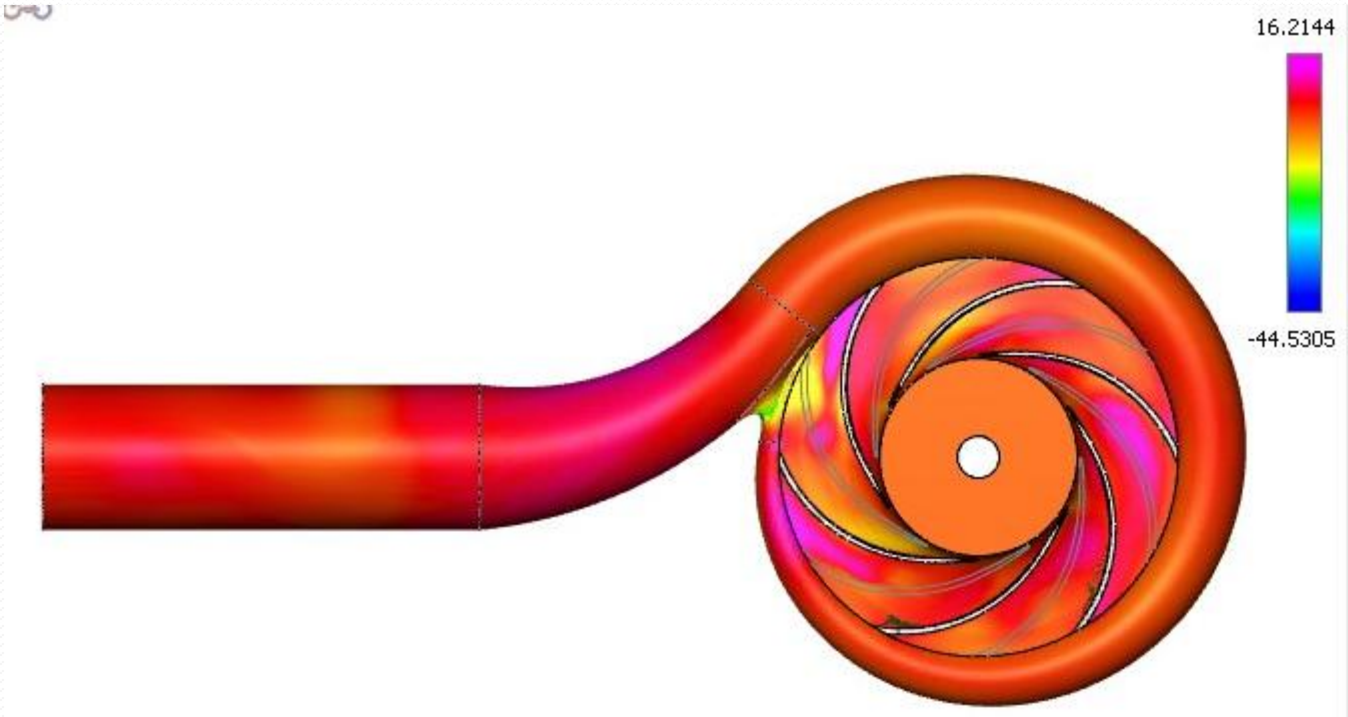
Simulation

To simulate the performance of the pump, we have used the Simerics PumpLinx Software.

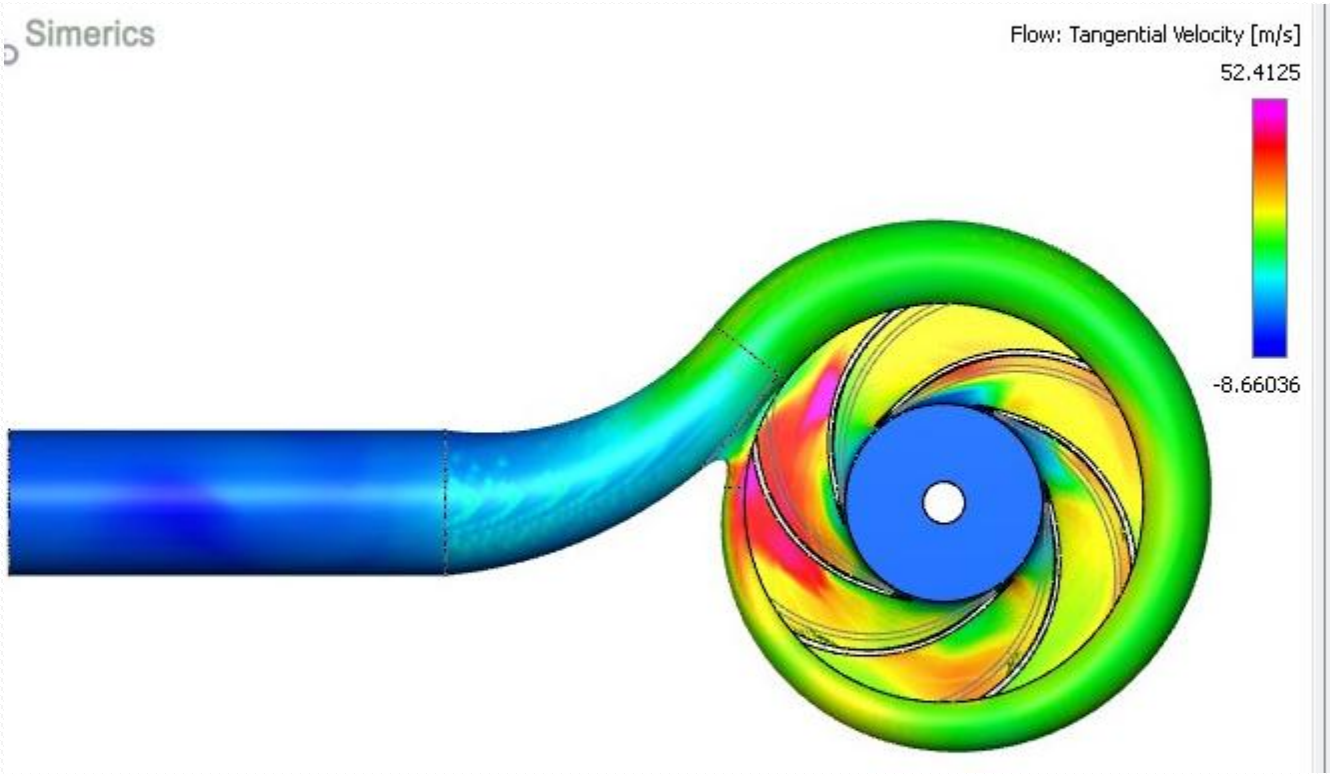
After we set the initial conditions (pressure inlet P=atm, temperature t=20 degrees Celsius, number of rotations 2900 rpm, Nr. of blades, direction of rotation as well and we determined the mesh grid for the CFD analysis, we got the results as below



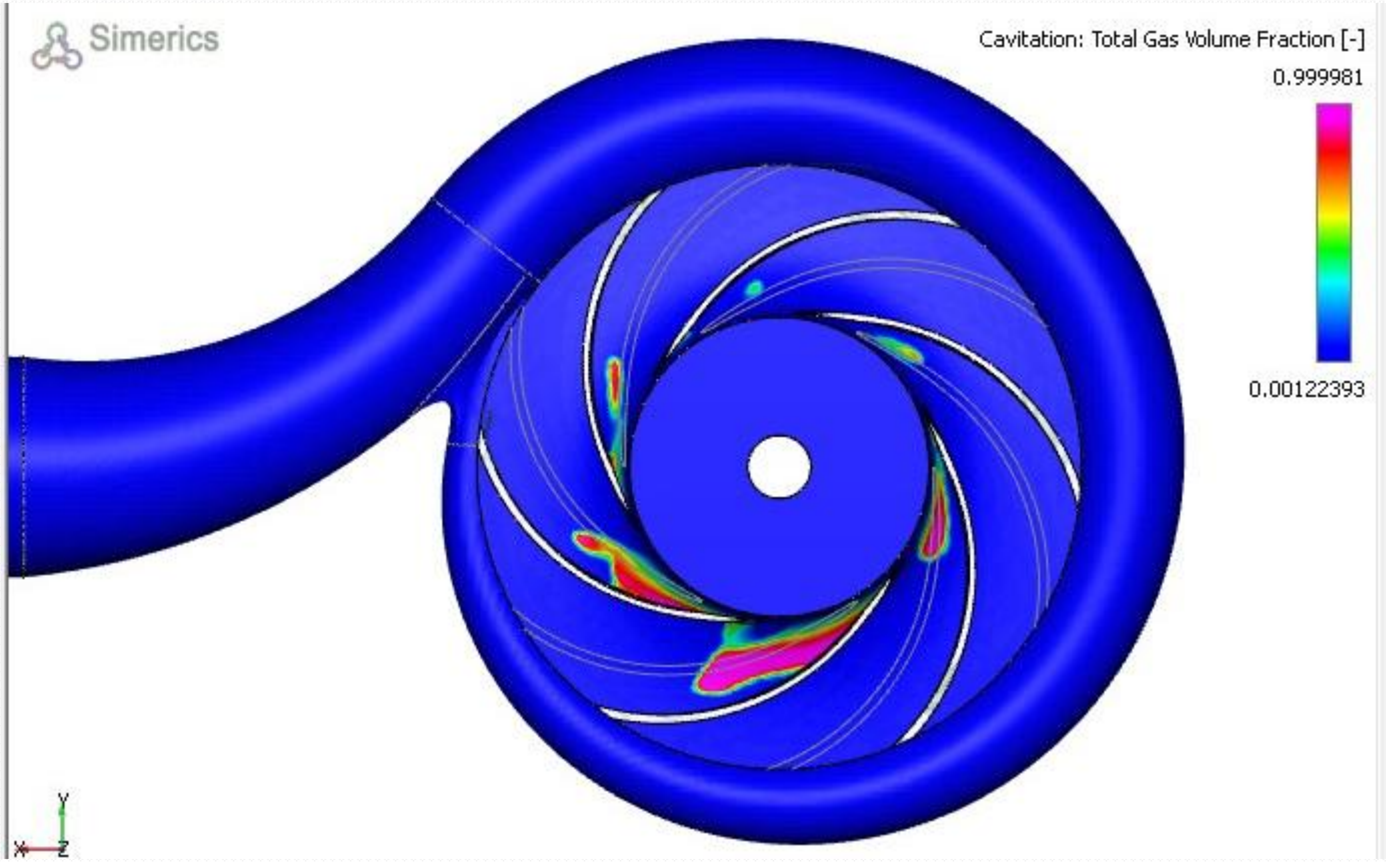
Pressure gradients for the original rotor of the pump



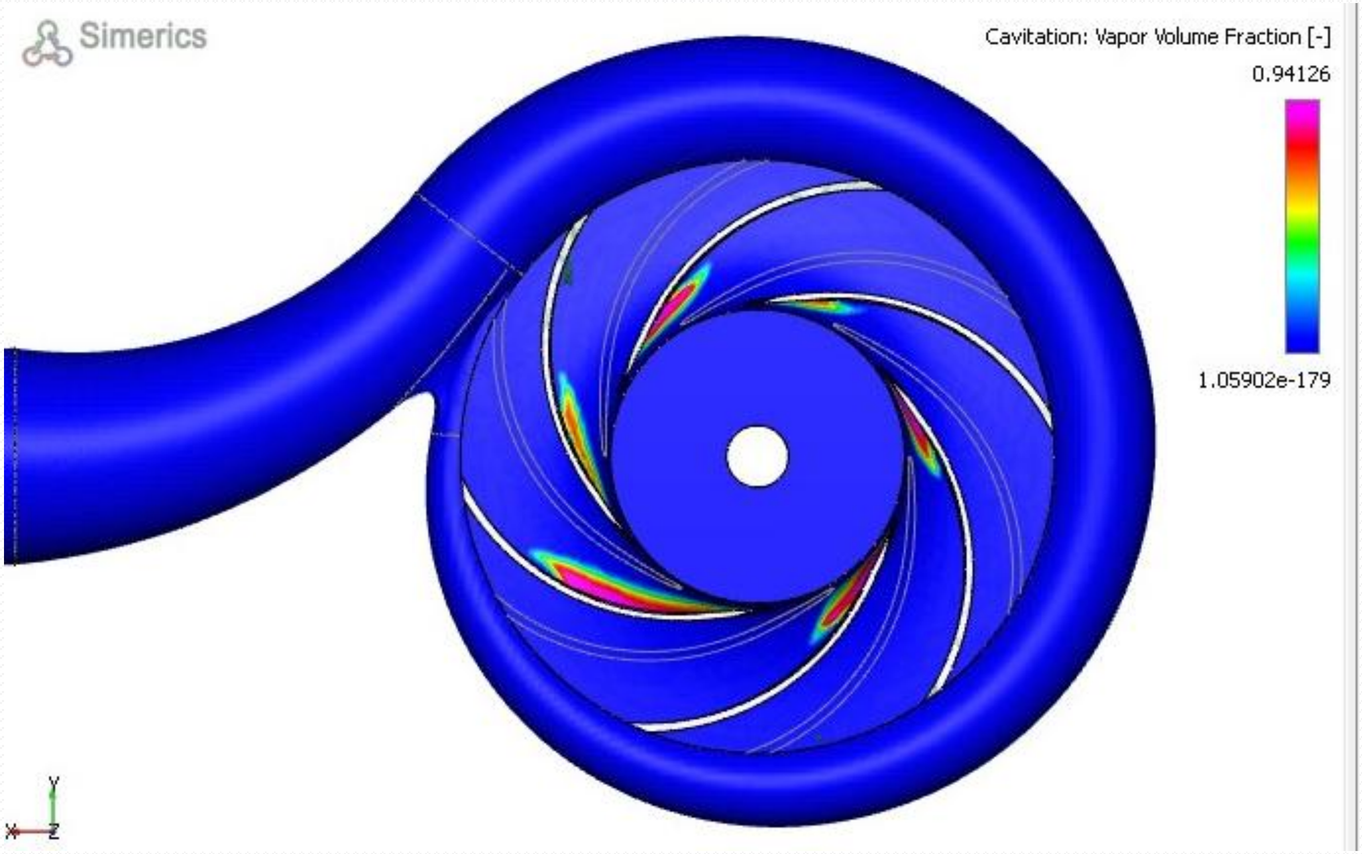
Radial Velocity gradients for original pump rotor



Tangential Velocity gradients for original pump rotor

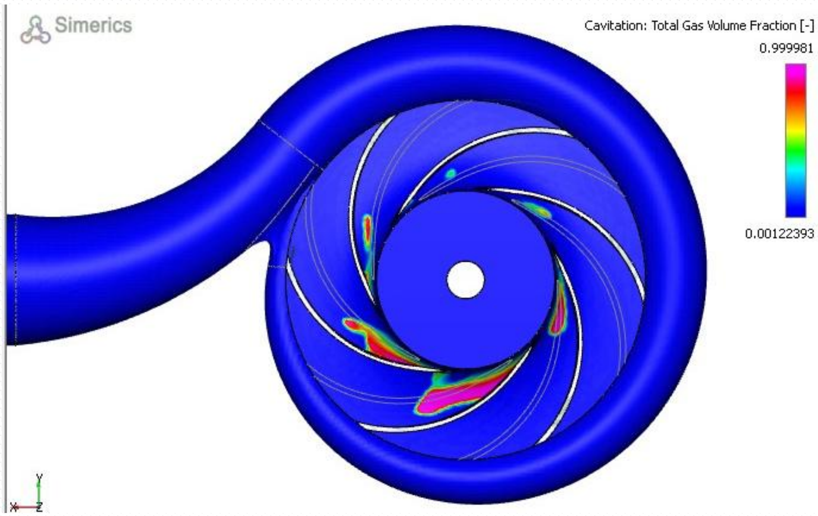


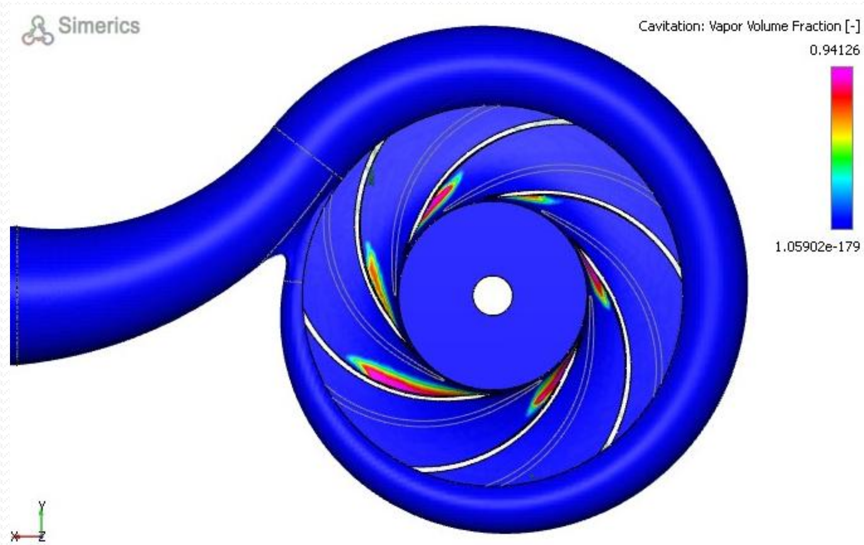
Cavitation: Total Gas Volume Fraction gradients , in the transitory state when the pump starts working



Cavitation: Vapor Volume Fraction gradients , in the transitory state when the pump starts working

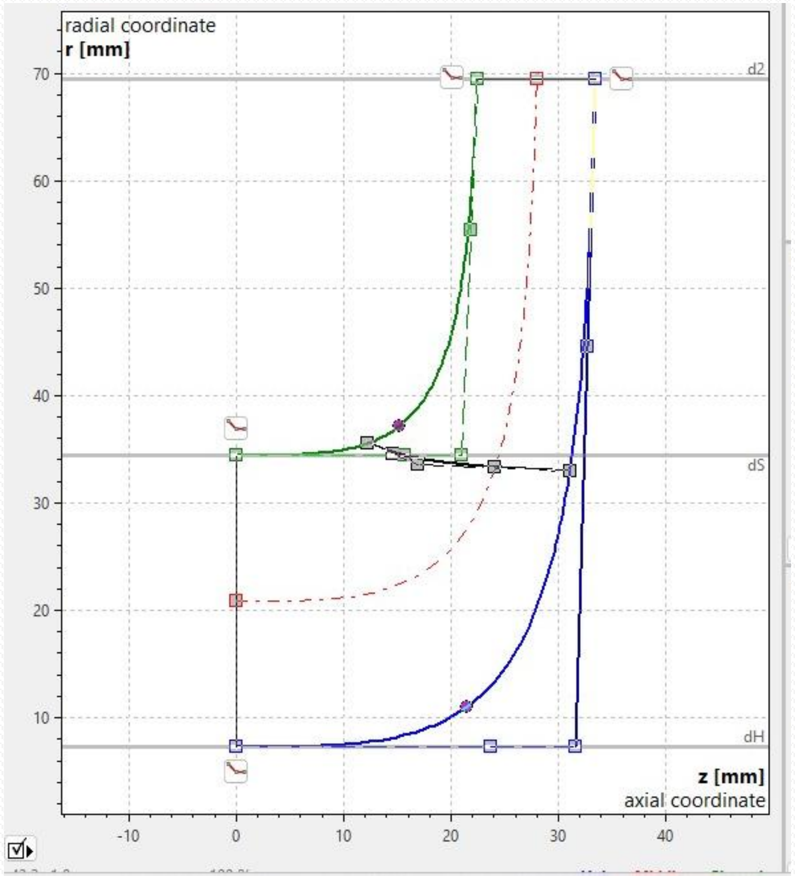
As can be seen from the computer simulations, in the transitory areas of the pump work, we have the creation of cavitation bubbles.

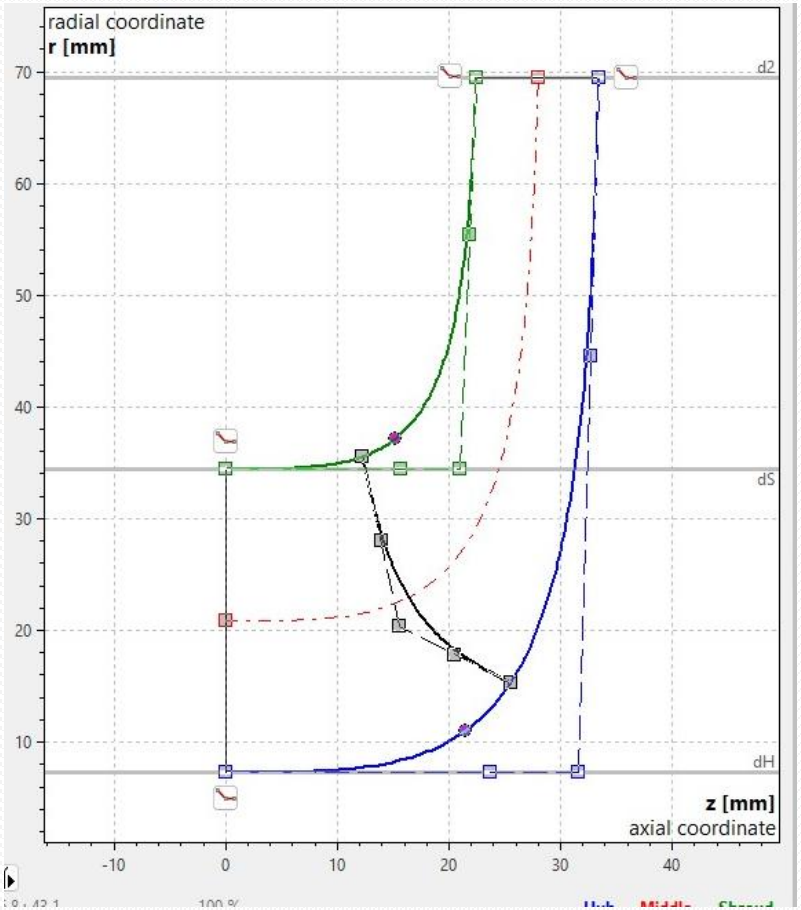




Cavitation: Total Gas Volume Fraction and Vapor Volume Fraction gradients , in the transitory state when the pump starts working

To avoid these phenomena and to increase the performance of the pump, we have intervened in the profile of the pump blade. changing the relative length of the meridian of the blade from 40% to 25%, thus increasing the length of the blade in the direction of the entrance and cross section fo the blades.





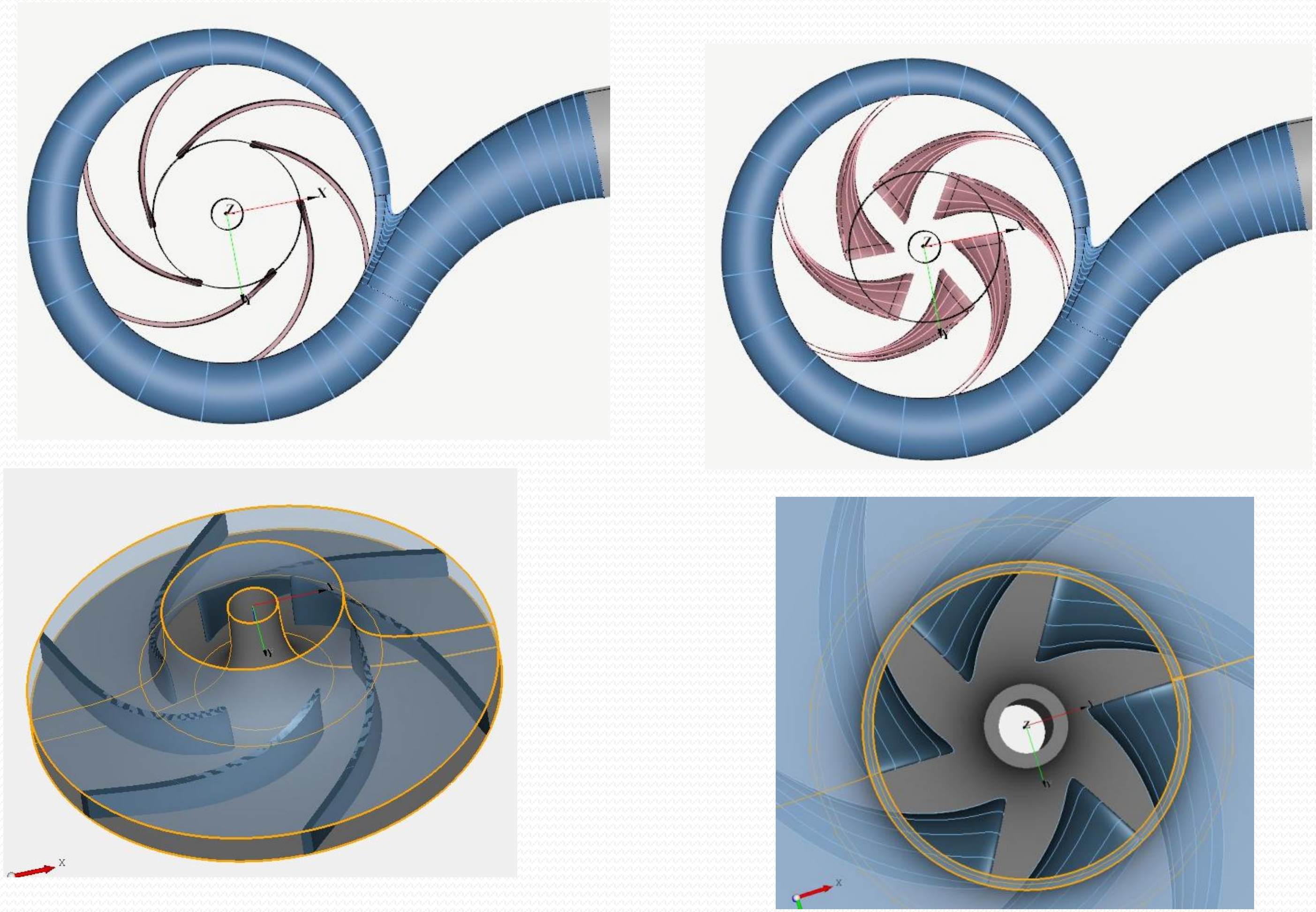
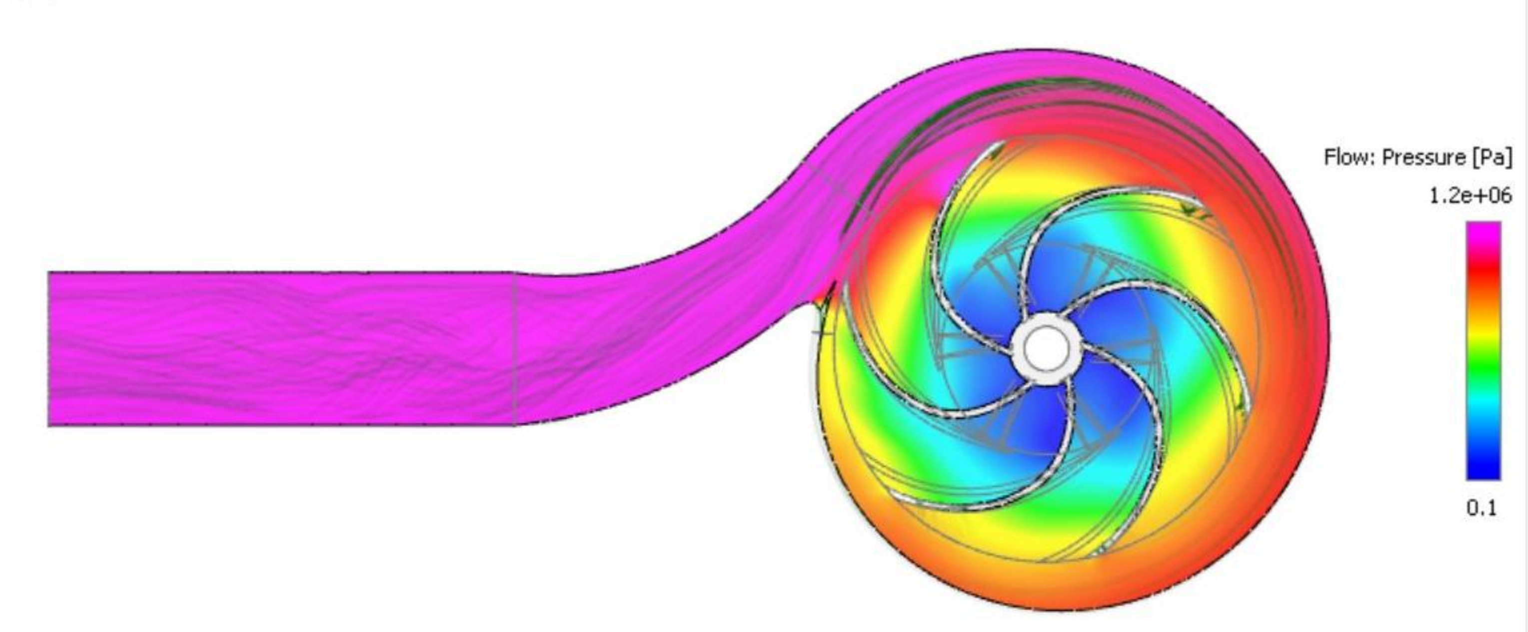
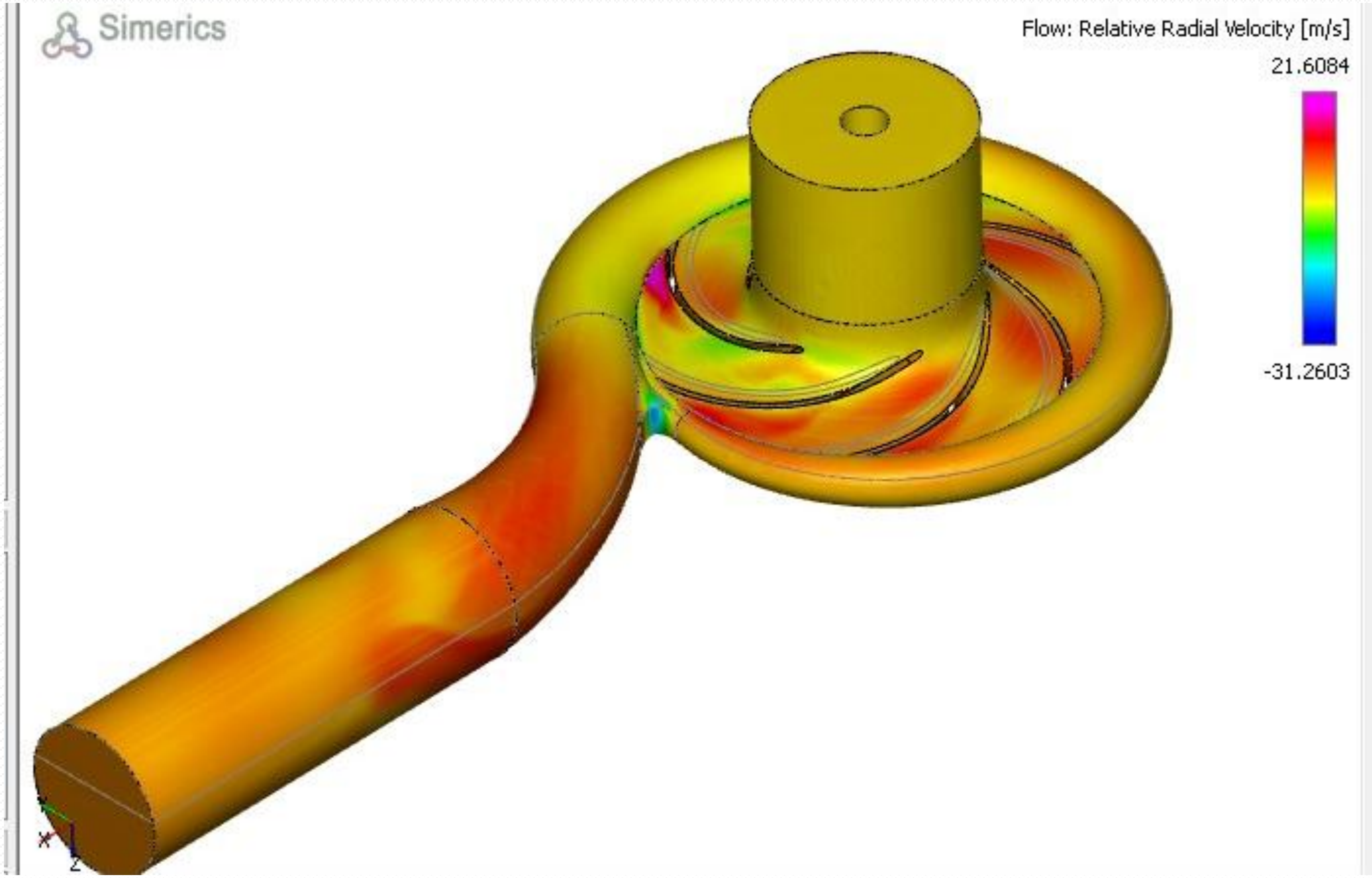


Figure a) Pump rotor modeled in CFTurbo with original parameters,
b) Pump rotor modified by changing the impeller with relative meridian length from 40 to 25.

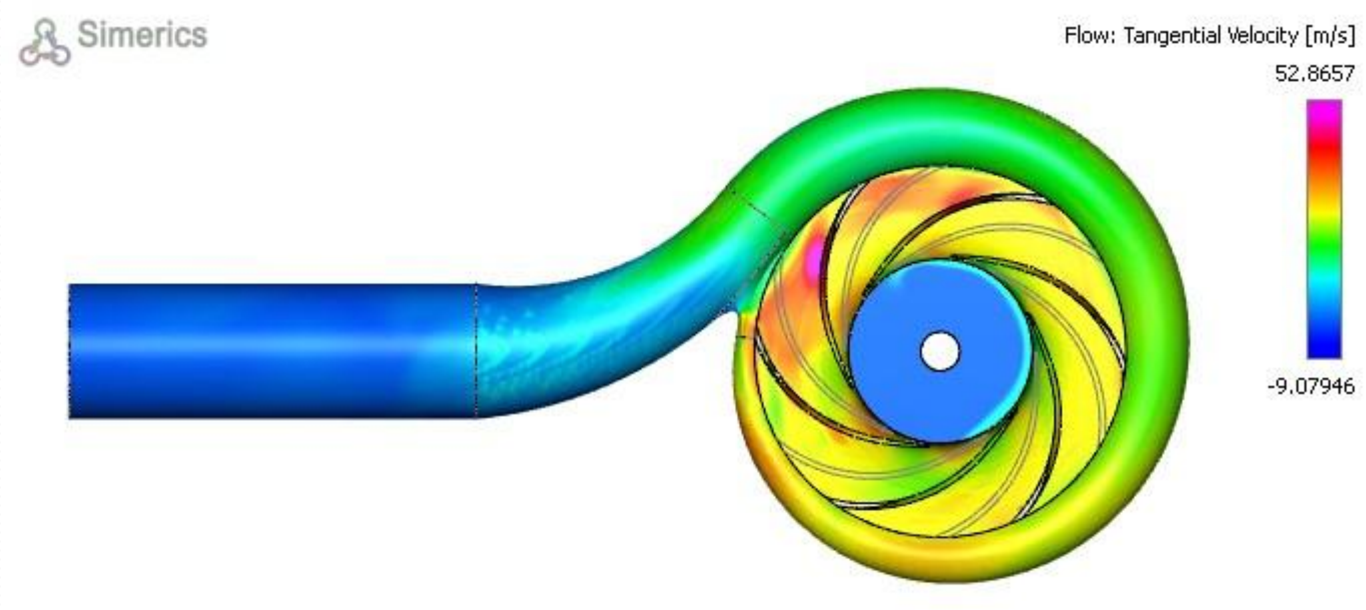
After the modifications made, we did other computer simulations on the new model and got the following results



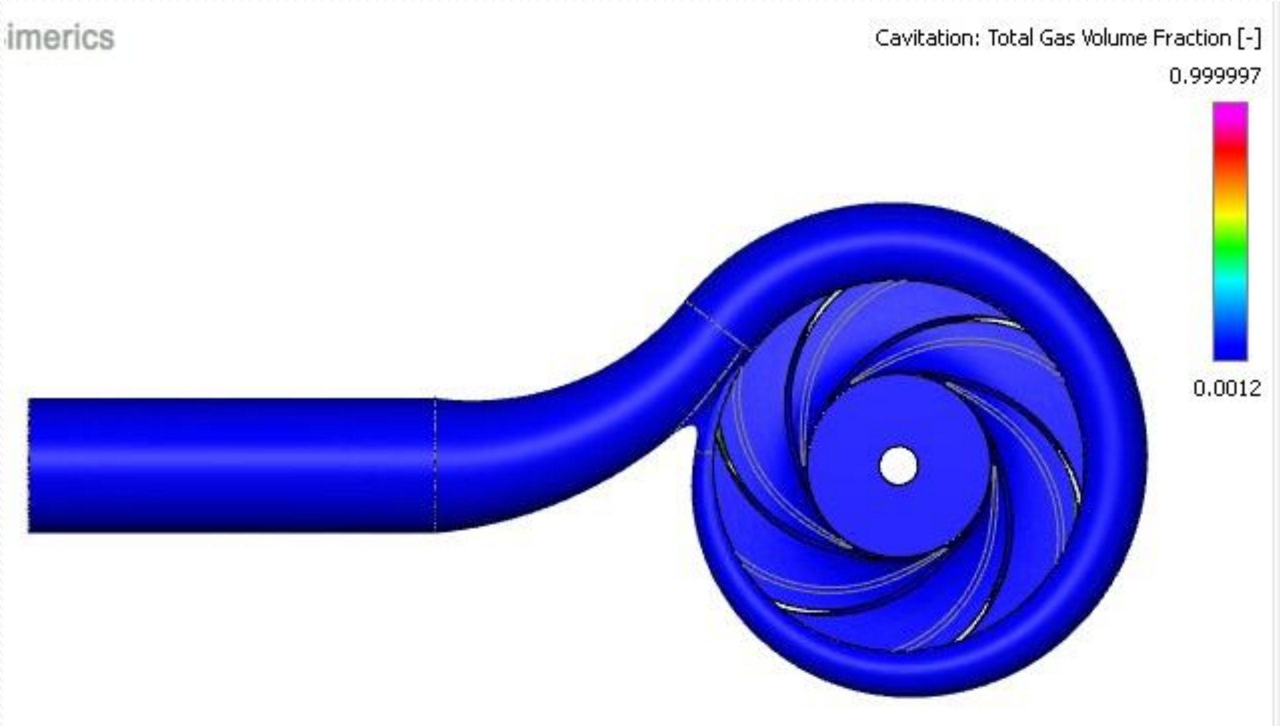
Pressure gradient for modified pump rotor



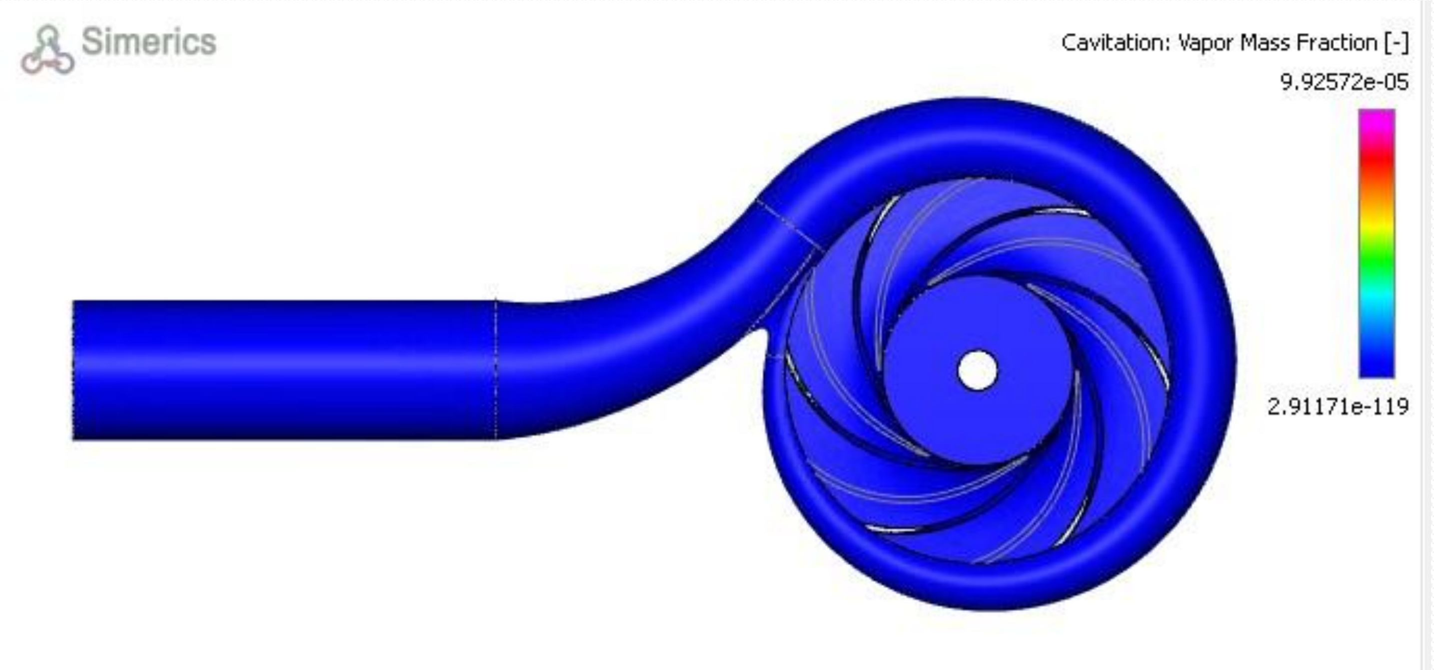
Radial Velocity gradients for modified pump rotor



Tangential Velocity gradients for modified pump rotor

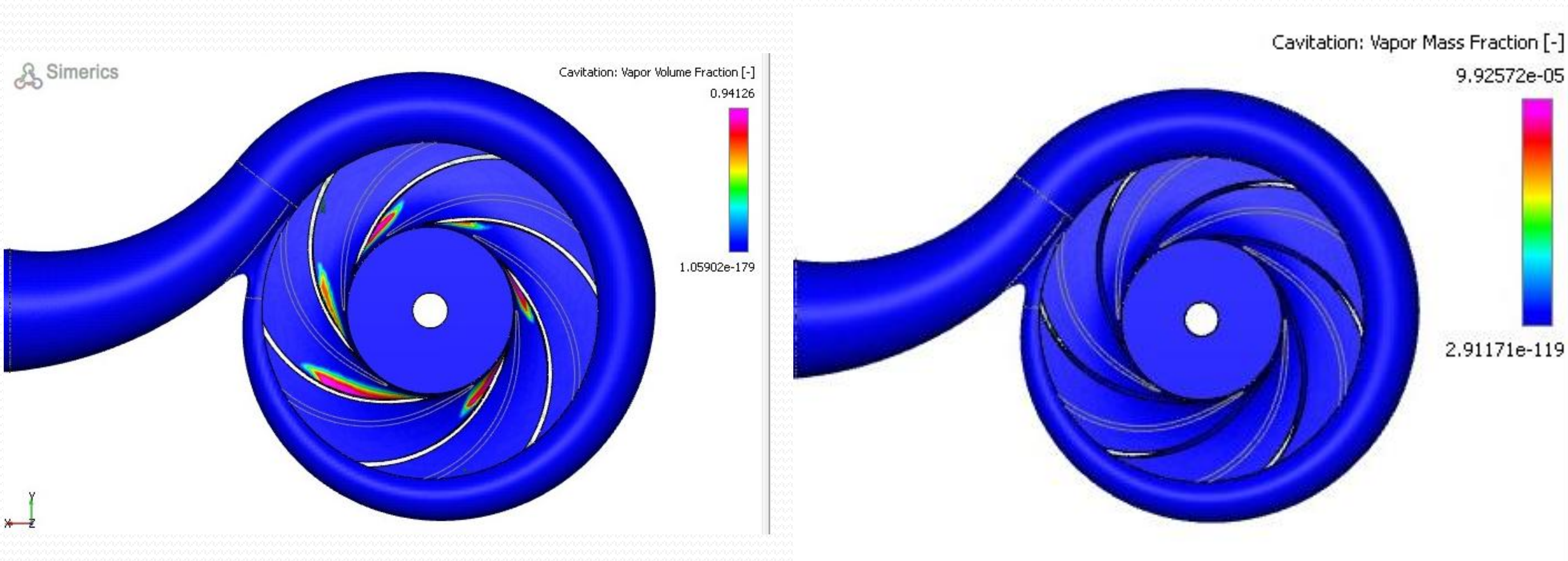


Cavitation: Total Gas Volume Fraction gradients , in the transitory state when the pump starts working for modified pump

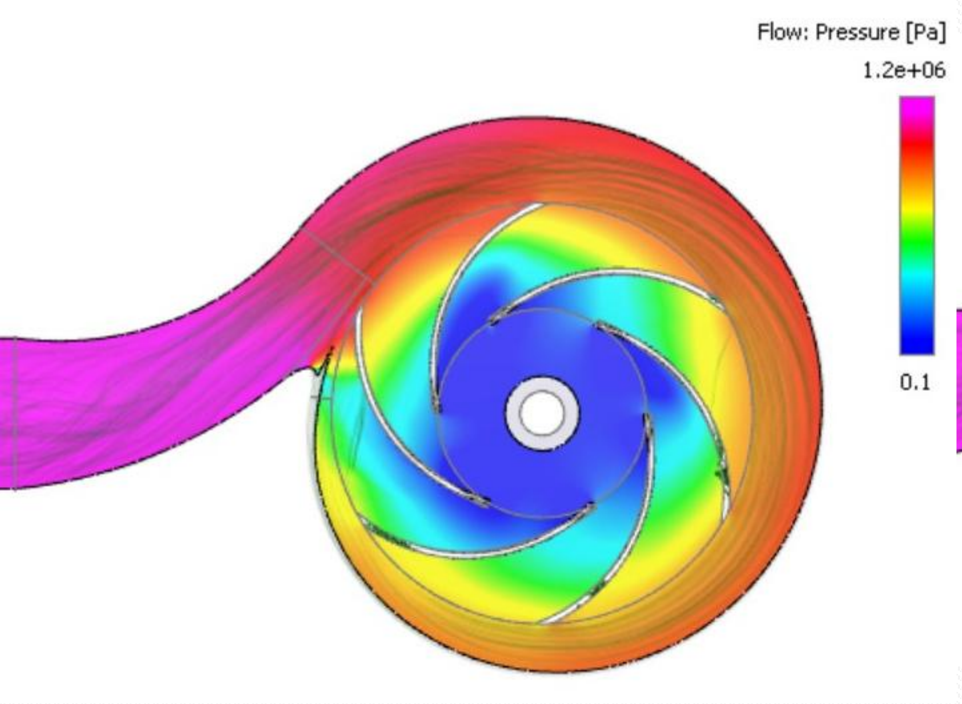


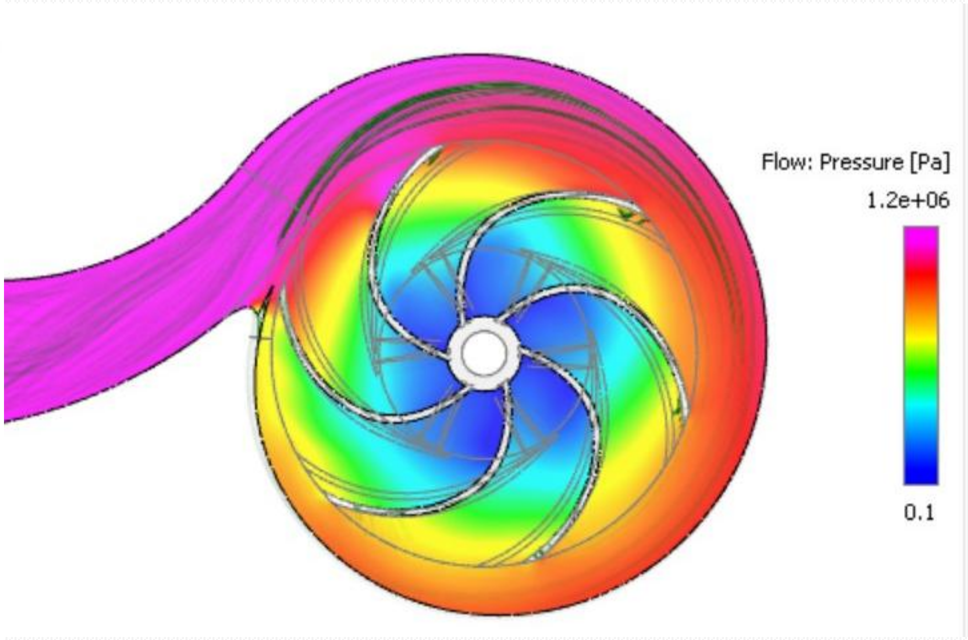
Cavitation: Vapor Volume Fraction gradients , in the transitory state when the pump starts working for modified pump

As can be seen from the comparison of the simulations made on the original pump and the modified one.
we have the elimination of steam bubbles in the transitory phase of the pump's work

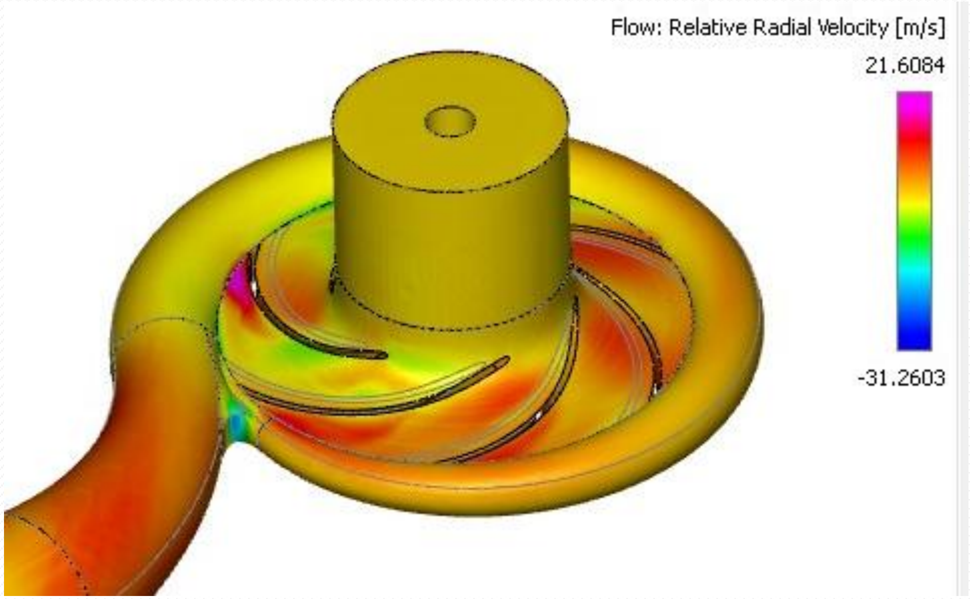


Cavitation, Vapor mass Fraction Left original pump right modified pump



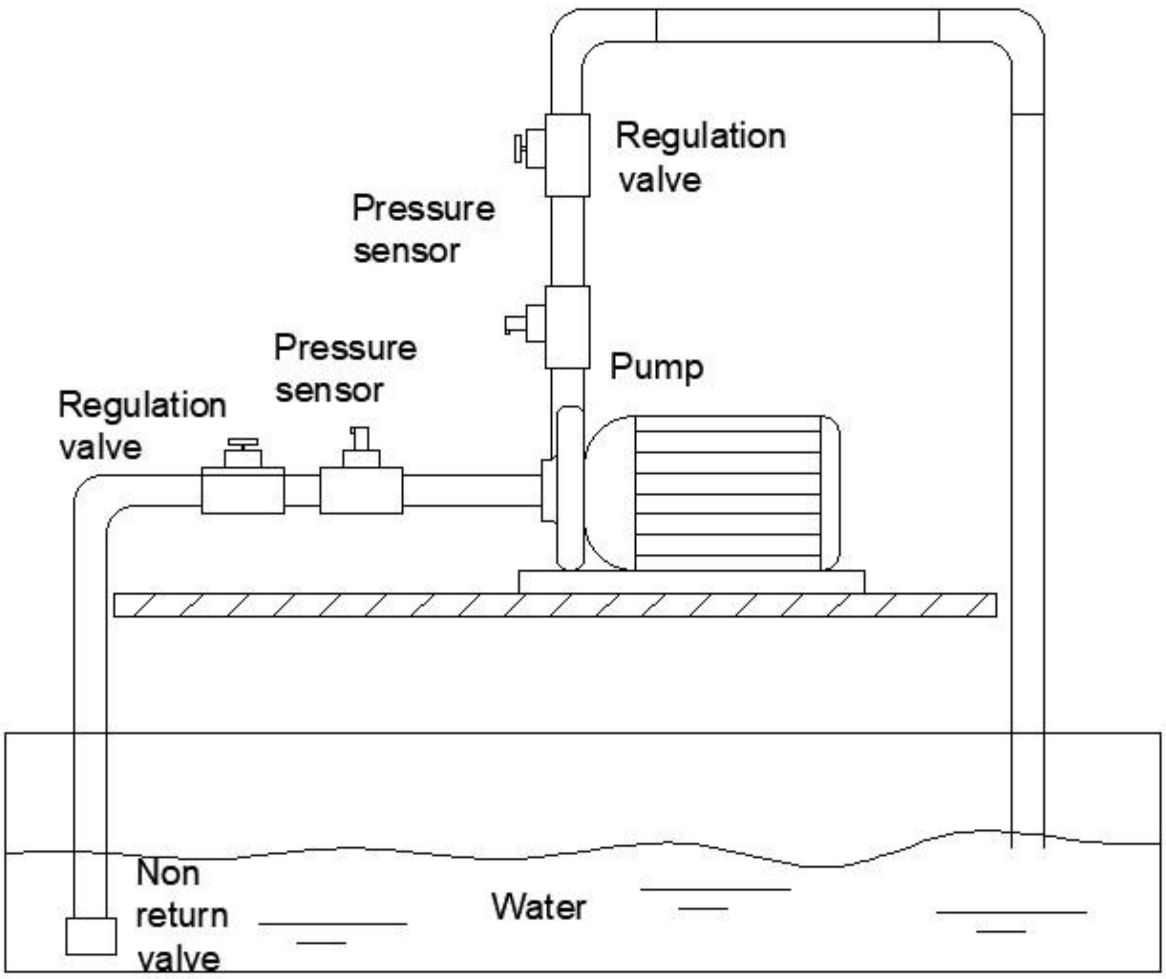


Even from the comparison of pressures, an increase in pressure values can be seen in the modified pump, both at the outlet and at the inlet



From the comparison of the radial speeds, an increase in the speed values can be seen in the modified pump

To verify the results of the performed simulations, we have built a test bench for the centrifugal pump.



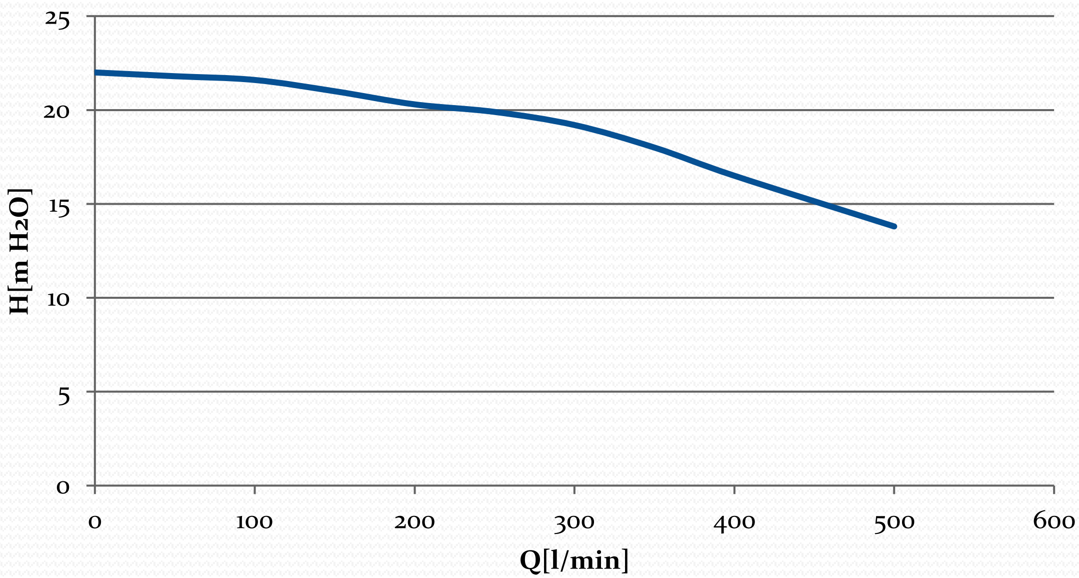
Test bench built

Through 3D printing we have reproduced the modified pump and we have performed experimental measurements with the aim of building the characteristics of the original pump and the modified one.

Through experimental measurements we have built the characteristics of the pumps shown in the graphs below

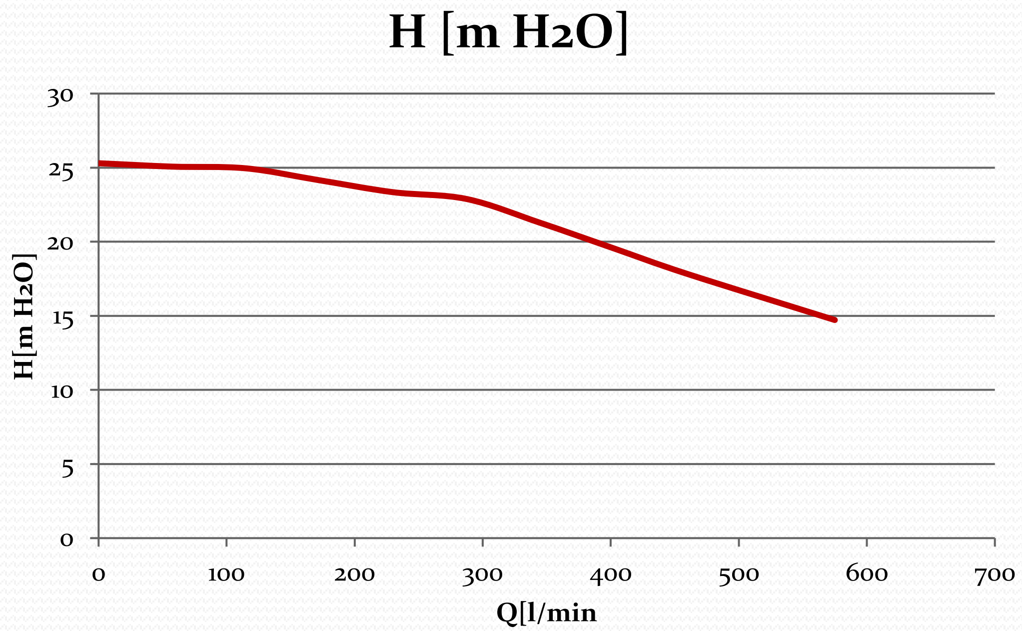
Table.1. Experimental measurements on the original pump

Q [l/min]	0	50	100	150	200	250	300	350	400	500
H [m H2O]	22	21.8	21.6	21	20.3	19.9	19.2	18	16.5	13.8



Original Pump characteristics measured

Experimental measurements on the modified pump										
Q [l/min]	0	57.5	115	172.5	230	287.5	345	402.5	460	575
H [m H2O]	25.3	25.07	24.96	24.15	23.345	22.885	21.28	19.55	17.825	14.72



Modified Pump characteristics measured

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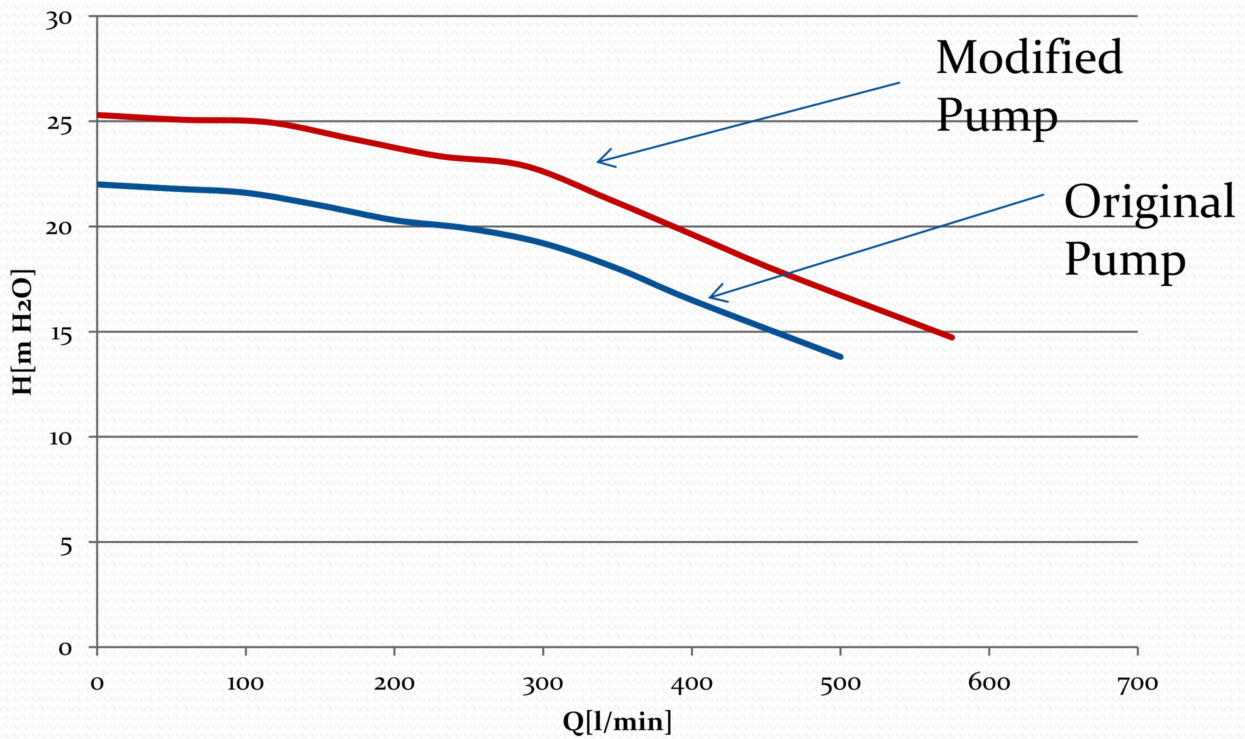

Superposition of the characteristics of the pumps

Conclusions

The change of the meridian length of the blade of the centrifugal pump positively affected the performance of the pump by eliminating the bubbles of water vapor created in transitory phases of the pump, and significantly increased its energy parameters. The increase in inlet pressure creates a larger reserve of NPSH, and this also proves the fact that steam bubbles are eliminated. these modifications are also supported by the mathematical model developed if we refer to Euler's equation for pumps. as well as speed fluctuations in centrifugal pumps where the increase in radial speed is visible in our simulation as the flow increases. a support for this argument is given by the experimental measurements carried out in the built bank test

THANK YOU FOR ATTENTION