

# UNIVERSITY OF WEST ATTICA SCHOOL OF ENGINEERING DEPARTMENT OF MECHANICAL ENGINEERING

# TURBOPUMP DESIGN FOR CRYOGENIC ROCKET ENGINE



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## ΤΡΗΜΕΛΗΣ ΕΞΕΤΑΣΤΙΚΗ ΕΠΙΤΡΟΠΗ

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«Είμαι συγγραφέας αυτής της διπλωματικής εργασίας και ότι κάθε βοήθεια την οποία είχα για την προετοιμασία της είναι πλήρως αναγνωρισμένη και αναφέρεται στην εργασία. Επίσης, οι όποιες πηγές από τις οποίες έκανα χρήση δεδομένων, ιδεών ή λέξεων, είτε ακριβώς είτε παραφρασμένες, αναφέρονται στο σύνολό τους, με πλήρη αναφορά στους συγγραφείς, τον εκδοτικό οίκο ή το περιοδικό, συμπεριλαμβανομένων και των πηγών που ενδεχομένως χρησιμοποιήθηκαν από το διαδίκτυο. Επίσης, βεβαιώνω ότι αυτή η εργασία έχει συγγραφεί από μένα αποκλειστικά και αποτελεί προϊόν πνευματικής ιδιοκτησίας τόσο δικής μου, όσο και του Ιδρύματος.

Παράβαση της ανωτέρω ακαδημαϊκής μου ευθύνης αποτελεί ουσιώδη λόγο για την ανάκληση του πτυχίου μου».

Ο Δηλών



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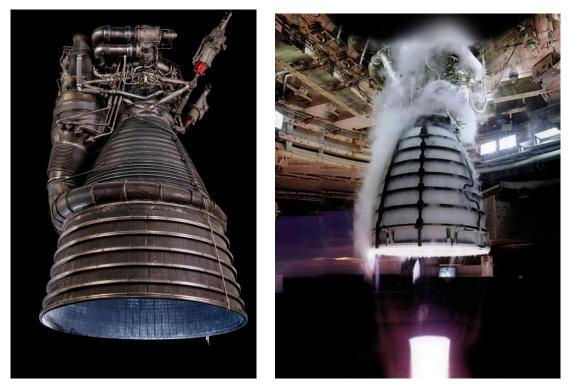
## ABSTRACT

In this thesis, a general report is made on the cryogenic rocket engines of spaceships using liquid fuels and a detailed description on the technical characteristics of a turbopump. Simultaneously, studies of various turbopumps that were used in known rockets are reported and the new current developments are analyzed. Next, all the laws of fluid mechanics that apply to a turbopump are described, as well as all the coefficients that determine its geometric characteristics and its durability. Subsequently, the design methodology of a turbopump for pumping liquid hydrogen and oxygen in detail using the CFturbo program. Finally, the CFD analysis for each component of the turbopump is conducted using the Simscale program. The results are evaluated, new suggestions for potential efficiency enhancements are proposed and general conclusions are drawn.

### Unit 1: Introduction to liquid rocket engine

#### 1.1 Liquid fuel spacecraft

As known, most spacecraft space missions are implemented using liquid fuels as they present a much higher degree of specific impulse Isp compared to those of solid fuels and hybrids (liquid oxidizer and solid fuel) [55]. In case of cryogenic rocket engine, the oxidizer and fuel are stored in liquid form under low temperature conditions. In this way, a greater amount of fuel is stored and volume is saved [55]. The liquid fuel and oxidizer are pumped and pressurized at high pressure into the combustion chamber of the rocket engine[55]. Before the fuel enters the combustion chamber it first flows through passages at high speed inside the walls of the rocket engine and cools it. After the combustion is done, the exhaust gases are expanded and exit the nozzle at high pressure and speed. With this method, the rocket engine is protected from thermal stress as very high temperatures develop inside the combustion chamber. A classic case is the cryogenic rocket engine that burns liquid hydrogen fuel with liquid oxygen. [1]. Also, there are liquid fuels which aren't in cryogenic state. For example, hydrazine and kerosene (RP-1) [55]. In this case the fuel cools the rocket engine but it isn't cryogenic.



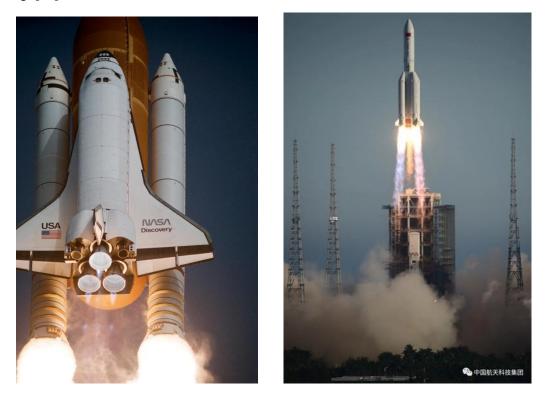
**Picture 1**: The F-1 engine, with 1.5 million pounds of thrust, was the powerplant for the first stage of the 363-foot long Saturn V launch vehicle that took astronauts to the Moon for six successful landing missions between 1969 and 1972 in the Project Apollo program [2].

Picture 2 The RS-25 engine, which successfully powered the space shuttle, is being modified for America's next great rocket, the Space Launch System [3].

The table below lists the most well-known and widespread fuels and oxidizers used in liquid rocket engines [55].

Chemical	Liquid fuels		Liquid oxidizers		
properties	Hydrazine	Liquid natural gas	Liquid hydrogen	Liquid fluorine	Liquid oxygen
Melting point (°C)	2	-183	-259	-220	-222
Boiling point (°C)	104	-162	-253	-188	-183
Heat- producing energy (Kj/Kg)	28.000	55.000	120.000	-	-
Density (Kg/m <sup>3</sup> )	1010	422	71	1509	1142
Stability	Toxic & flammable	Toxic & flammable	flammable	Very toxic & flammable	good
storage	normal	cryogenic	cryogenic	cryogenic	cryogenic

Since liquid rocket engines, which are commonly used in launches, have a leading role in international space missions, they are still prone to failure, so the risk and cost remain high[54].



Picture 3 The space shuttle was NASA's space transportation system. It carried astronauts and cargo to and from Earth orbit. The first space shuttle flight took place April 12, 1981. The shuttle made its final landing July 21, 2011. During those 30 years, the space shuttle launched on 135 mission[4]

Picture 3 A long March 5B rocket lifts off from the Wenchang launch center May 5. Credit: CASC [5]

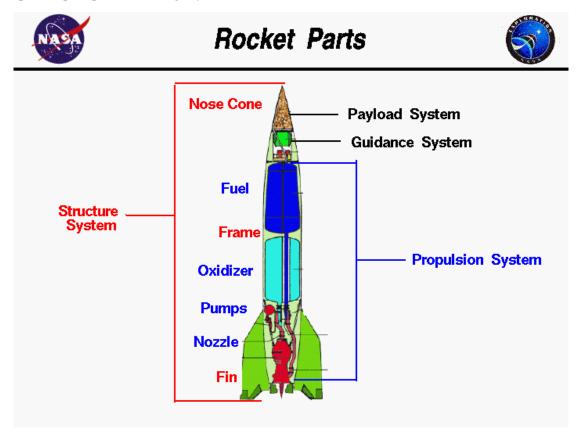
#### 1.2 Liquid fuel storage systems

Fuel and oxidizer are stored in separate tanks. They should be relatively light. When the fuel and oxidizer are in cryogenic conditions then they must be both durable and insulated [6]. They are usually made of aluminum alloys. In order to maintain low temperatures, polyurethane, mud or polystyrene coating is used as thermal insulation. Also, it should be noted that the choice of material for the tanks also depends on the fuel or oxidizer itself that will be stored. The table below show various materials that are conventional for two classic examples, liquid hydrogen and liquid oxygen.

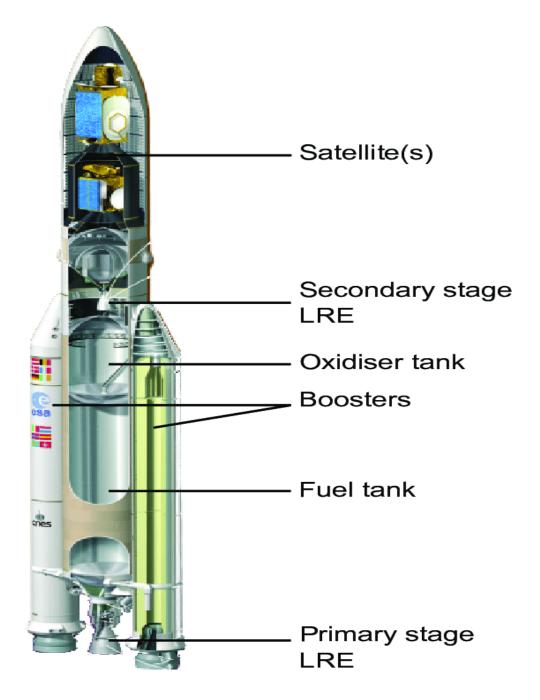
Liquid substance	Conventional materials of tanks		
Liquid hydrogen	Aluminum allows, Nikel allows, stainless steel		
Liquid oxygen	Aluminum allows, Nikel allows, stainless steel, copper, Teflon		

#### [55]

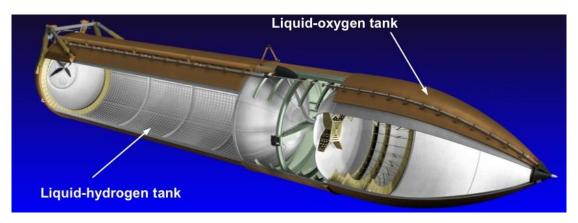
The pictures below show the shape of these tanks and how they are positioned in the spaceships liquid fuel storage system



Picture 4 Rocket parts The structure of the V-2 ballistic missile system is presented [7]



Picture 5 The main components of the Ariane 5 european space launcher [8]

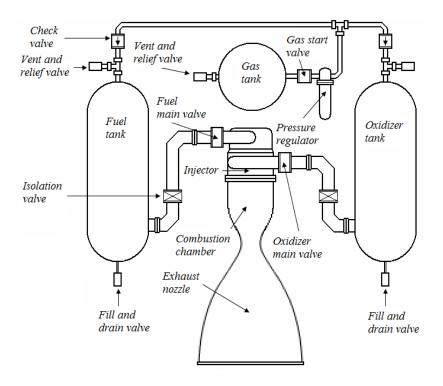


Picture 6 Internal structure of the external space shuttle tank [9]

#### 1.3 Liquid fuel pumping systems

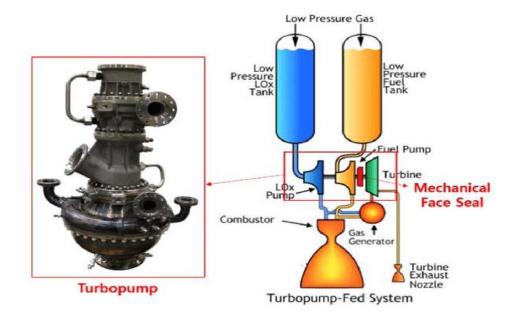
Generally, there are two kinds of pumping systems of liquid fuel in the liquid rocket engines. The pressure vessel system and the turbopump system. In the turbo pump system these is also a pressure vessel but only to prevent cavitation in the pumps[10]. These systems present the following technical characteristics :

• PRESSURE VESSEL SYSTEM : Inert gas is usually stored there, for example nitrogen or helium under very high pressure in a vessel [55]. Typical values of this pressure are from 69 bar to 690 bar [10], depending on the thrust requirements and the hydraulic losses that occur during pumping . The high pressure of the gas pushes the liquid fuel and oxidizer to mix in the combustion chamber of the rocket engine. In this way, the total hydraulic is secured, losses are overcome and the required pressure in the combustion chamber is ensured. The pressure is regulated with the use of special valves. The main advantages of this system are its reliability [55] and that it hasn't got moving parts. But they are not preferred for high-thrust launches or long missions because large pressure vessel sizes will be required. This results in an excessive increase in the weight of the spacecraft [11]. For this reason, these systems are used in the smaller missions of spacecraft which don't have particularly high thrust requirements [55].

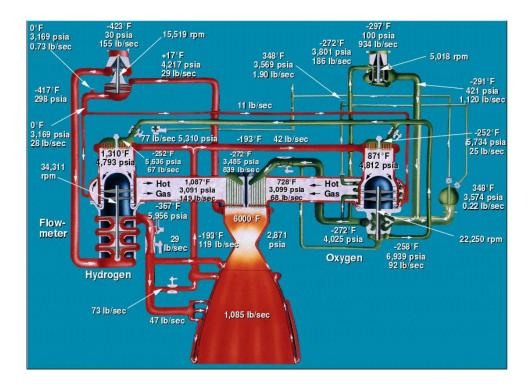


Picture 7 Scheme of a engine with pressurized gas propellant feed system [10]

• TURBOPUMP SYSTEM : This system although more complex in construction, plays a dominant role in liquid rocket engines for many reasons. First, it can secure very high pressure in combustion chamber. Second, it is very lightweight, it has small volume and develops a very high thrust relative to its volume[11]. Two centrifugal pumps pump the fuel and the oxidizer from tanks and they press them with high pressure in the combustion chamber. These pumps are rotated by a turbine that runs on a very small amount of fuel in a pre-burner or by exploiting the heat generated by the rocket engine. Furthermore, Turbopump systems are usually also combined with smaller pressure vessels which offer a little assistance.



Picture 8 Configuration of 7-ton class turbopump and layout of turbopump type rocket engine [12]



Picture 9 The Space Shuttle Main Engine (SSME) Propellant Flow Schematic [13]

### Unit 2 General description and the structural parts of a turbopump

#### 2.1 Development history of the turbopump and its current developments

The idea of developing the turbopump came from physics professor Robert Hutchings Gorddad, in 1934. He is considered the father of modern rocket propulsion as he was the one who built and successfully tested the first liquid fuel rocket launch in 1926 [14],[15]. However, the idea the idea that would grow to become the modern rocket turbopump begins life on the design offices in Kummersdorf, Peenemünde and Frankenthal in 1935 in German. Presented by Robert J Dalby. This turbopump was used for program V-2 rockets[16]. Over the years turbopumps evolved and have been used in aerospace for many decades [10]. Nowadays, the 59% of launch failures are attributed to the propulsion system. The main reasons for propulsion system failure are burner instability and turbopump inaccuracy. Due to cavitation instability in pumps, many space missions have failed. Notable examples include NASA's Apollo, Fastrac programs, ESA's Vulcain program, and JAXA's LE-7 program. Unfortunately, cavitation instability in these pumps has not been fully understood to date, and there is no established method for predicting it during the design process [11].



Picture 11 V-2 engine on display at the NationalMuseum of the United States Air Force near Dayton, Ohio [17]

Picture 10 The V-2 rocket that was successfully launched on October 3, 1942 [18]

#### 2.2 Turbopump structure

The following images show the external form of a turbopump :



Picture 13 This turbopump, intended for the SpaceX Merlin engine, is one of many designs manufactured by Barber-Nichols. The company credits work it did on NASA's Fastrac program with enabling better and cheaper methods of building these machines. Credit: Barber-Nichols Inc [19]



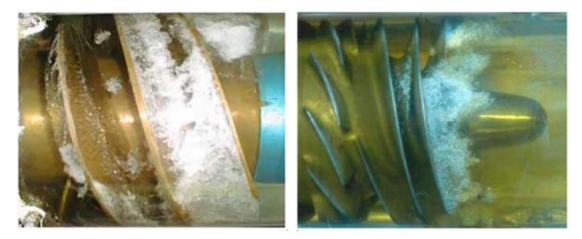
Picture 11 Inducer and impeller and turbine of a rocket engine turbopump [11]

Picture 12 LOX/CH4 single stage single shaft turbopump - Le Bourget 2015 [20]

A simple classical turbopump consists of the following main mechanical components:

1) INDUCER : The inducer essentially is an endless scroll which is positioned in front of the centrifugal pump impeller. It is an integral part of the pump and is necessary to combat cavitation [55]. Cavitation is a phenomenon where vapor bubbles which form

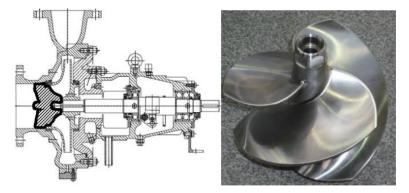
in the flowing fluid collapse suddenly – potentially causing surface damage of the impeller, performance degradation, as well as catastrophic failure. The inductor must be properly designed to protect the pump from cavitation .The number of vanes ranges from 3 to 4 typically [21].



Picture 13 Cavitation is turbopump inducers. Here is shown the vaporization of the fluid that occurs on the vanes of the switch during its operation [11]



Picture 14 Model a typical inducer [22]



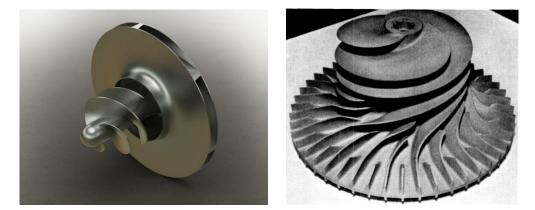
Picture 15. Inducer (bold black line) placed in front of impeller in centrifugal pump and photo of inducer [23]

2) CENTRIFUGAL IMPELLER : In these applications, the pump must be able to significantly increase the pressure of the fuel and oxidizer. The centrifugal pump has been an important element of the history of pump-fed liquid propellant rocket engines. The use of this type of pump is the resultant of its relative simplicity and reliability, wide opcrating flow range, and adequate performance[24]. The hydraulic efficiency can exceed 80% [25].



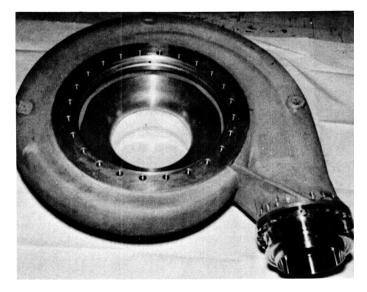
Picture 16 A typical impeller with inducer [26]

Picture 17 A Ti-5Al-2.5Sn fuel pump impeller used in the space shuttle main engine [27]



Picture 20 A typical impeller with inducer [28] Picture 18 typical unshrouded LH<sub>2</sub> pump impeller (NERVA I)[24]

3) VOLUTE-DIFUSSER : It is the classic volute - diffuser used in centrifugal water pumps as well. It has the geometry of a spiral and ensures an additional increase in the static pressure of the fluid. This is achieved via a gradually increasing crosssectional area in the volute that increases the fluid's flow-through area, thus increasing the static pressure of the fluid [29]. The diffuser gives an additional pressure increase at the outlet [30].

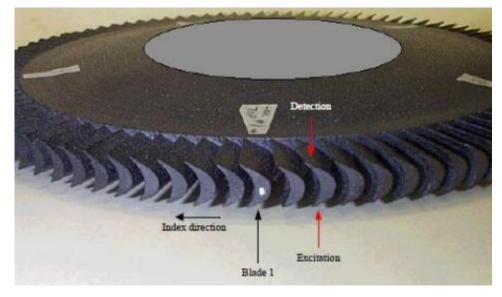


Picture 19 LH<sub>2</sub> pump housing with diffusers (NERVA I) [24]

4) TURBINE: Key role rotates the pumps very quickly. It can operate at various pressures and temperatures. They are usually axial.



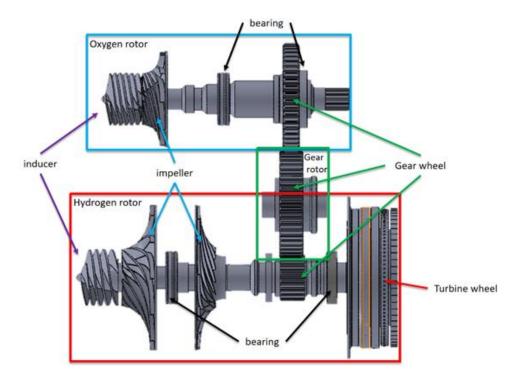
Picture 20 Turbine rotor (blisk type) [31]



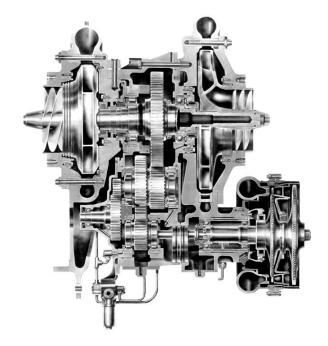
Picture 21 : Photo of SIMPLEX turbopump blisk (Courtesy of NASA MSFC) [32]

In addition to these main components, the centrifugal pump also includes the casing, which houses all the aforementioned parts, specifically bearings for support and sealing, and the inlet and outlet pipes of the system.

Additionally, there are several cases where the fuel pump must operate at different speeds than the oxidizer pump for various technical and design reasons. For this reason, in some cases the oxidizer and fuel pumps work together and rotate with the turbine with a gearbox.



Picture 22 Internally geared turbopump model - A standard schematic of an internally geared turbopump consists of the liquid hydrogen (LH2, fuel) and liquid oxygen (LO2, oxidizer) rotors [33]



Picture 23 NASA's propulsion-system turbopumps, such as the Rocketdyne Mark-3 turbopump shown here, are sophisticated, complex, and expensive. As part of an effort to develop smaller, simpler, more affordable turbopumps for smaller spacecraft, Marshall Space Flight Center engineers created the Generalized Fluid System Simulation Program (GFSSP) to simultaneously analyze all the interacting flows in these intricate machines [34]

#### Materials used in turbopumps

For these applications, construction materials must strictly adhere to two basic criteria: high resistance to thermal stress and being very lightweight. Turbopumps are typically made from titanium or nickel alloys or titanium alloys[1]. Also, the gears of gearbox are made from various alloy AISI or AMS [35]. It is very important to select the material of a turbopump because Turbopump weight continues to be a dominant parameter in the trade space for reduction of engine weight. The table below lists some common materials used in these applications.

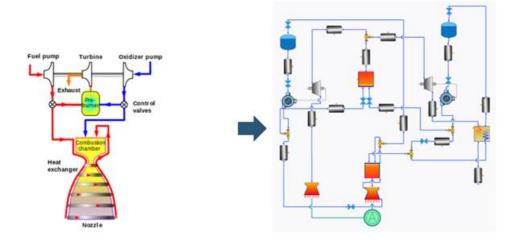
MATERIAL	σ <sub>0,2</sub>	σ <sub>TS</sub> (Mpa)
99,2Ti	450	525
Ti-5Al-2,5Sn	800	900
Ti-6Al-4V	950	1000
Al2014-T6	430	480
Inconel 718	760	860

#### [57],[36],[54]

#### 2.3 Analysis of combustion cycles

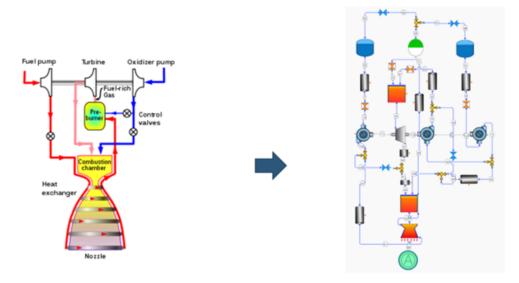
The method by which a turbopump system is implemented and operated in a liquid rocket engine varies depending on the load and thrust requirements for the launch. For these reasons, certain combustion cycles have been developed with which a turbopump can operate.

1) OPEN CYCLE GAS GENERATION : In this cycle, a small amount of oxidizer and fuel are burned separately in a preburner and the exhaust gases produced are blown into the air turbine and then released into the environment. This cycle satisfies high thrust loads and is reliable but has poor efficiency as extra heat is wasted which is released to the environment and not utilized by the engine. It is simple in design without much complexity[37].



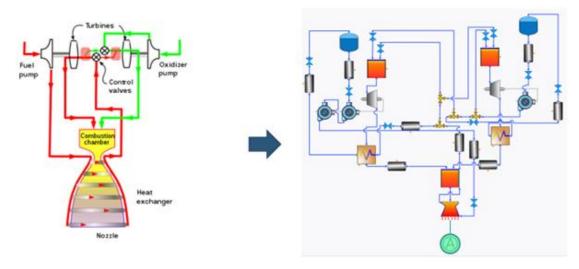
Picture 24 Gas-generator rocket cycle [37]

2) CLOSED CYCLE FUEL RICH : In this case, uses the remaining fuel-rich exhaust gases to drive the turbine that powers the propellant pumps. This allows for more efficient use of the propellants, as the remaining exhaust gases burn entirely, and the energy is not wasted. A key advantage of this cycle is the high specific impulse and efficiency, resulting in a greater thrust-to-weight ratio. However, this construction is more expensive and It is a complex and challenging cycle to design and operate, requiring high-precision and high-temperature materials to withstand the harsh combustion environment [37].



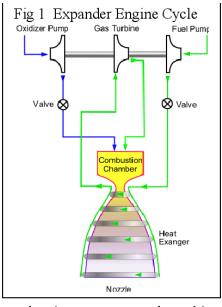
Picture 25 Fuel-rich staged combustion cycle [37]

3) FULL FLOW STAGED COMBUSTION CYCLE : This cycle satisfies high thrust loads and is one of the most efficient. The fuel and oxidizer are initially fed into separate pre-burners, where they are partially burned to generate a high-pressure and high-temperature gas mixture. This gas mixture is then fed into the main combustion chamber, which is fully burned to produce high-pressure exhaust gases that exit through the nozzle, providing thrust.. However, a key disadvantage presented by this cycle is manufacturing complexity. This also increases the cost of construction[37].



Picture 26 Full-flow staged combustion rocket cycle [37]

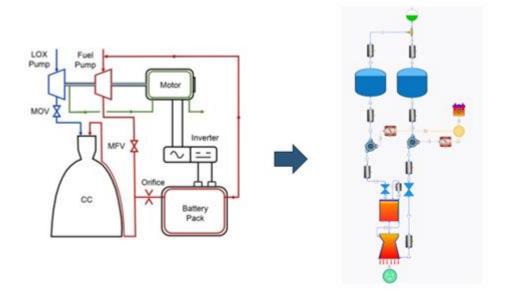
4) EXPANDER CYCLE : In this cycle, there is no pre-burner which makes it even simpler to design. The fuel that cools the nozzle is used as a working medium for the operation of the air turbine. It essentially takes some of the heat from the walls of the rocket engine. However, this cycle cannot meet high thrust requirements [38].



combustion generator to the turbine.

Picture 27 Expander rocket engine (closed cycle) [38]

5) ELECTRICAL PUMP : Practical attempts have been made to adapt an electric motor to rotate pumps in aerospace. The fuel and oxidizer are then burned in a combustion chamber, producing exhaust gases that exit through the nozzle to provide thrust. The electric pump-fed cycle has several advantages, including simplicity, reliability, and the ability to use a wide range of fuels. But this method has some disadvantages, such as lower efficiency, high battery weight, and lower thrust-to-weight ratio than the gas generator cycle [37].



Picture 28 Electric Pump-Fed Engine Cycles [37]

#### 2.4 Examples of rocket engines using a turbopump and their technical characteristics

In this subsection, some known rocket engines using turbopump are presented. The familiar liquid hydrogen fueled Space Shuttle rocket engine has two turbines one for the fuel pump and one for the oxidizer pump. They have a power of 54.36 Mw (fuel) and 19.28 Mw (oxidizer). The outlet pressure is 441 bar (fuel pump) and 499.9 bar. Finally the turbine inlet temperature is 838°C (fuel) and 671.3°C (oxidizer)[39]. Another well-known rocket engine is that of Space-X called Raptor. It works with liquid methane and liquid oxygen. The fuel supply is 194 Kg/s and 737.2 Kg/s of oxidizer respectively[40]. The picture below shows the RD-170 rocket engine :



Picture 29 The RD-170 rocket engine which was used for the rocket Energia [41]

It is one of the most well-known rocket engines that the Russians have built and is one of the largest in power on a global scale [41]. The RD-170 is designed to run on kerosene(RP-1). Fuel mass delivery 166.2 Kg/s and oxidizer mass delivery 436Kg/s [42]. Below is the H-1 rocket engine with its turbocharger technical characteristics for a specific thrust. It concerns the H-1 C and H-1 D engine version.



Picture 30 The H-1 liquid-fuel rocket engine was the first stage power plant for the Saturn 1 and Saturn 1B launch vehicles [43]

In this case, although the oxidizer pump is on a common shaft with the fuel pump, the air turbine is connected to the pump shaft through a gearbox. This engine has historical value. as it was used in the famous Saturn project in 1961. However, the fuel whichused was kerosene (RP-1) [44]. Some of its technical characteristics of turbopump are listed below.

TECHNICAL CHARACTERISTICS		
Power turbine	3.088 MW	
mass flow of oxidizer	247.4 Kg/s	
mass flow of fuel	110 Kg/s	
Fuel	RP-1	
Total effience n <sub>t</sub> of turbine	70%	
turbine intel Pressure	43 bar	
Type turbine	two stage axial	
RPM pumps (RP-1 & LO <sub>2</sub> )	6680 min <sup>-1</sup>	
hydraulic efficiency pump fuel	71.78%	
hydraulic efficiency pump oxidizer	77.88%	
Intel pressure total of fuel	3.93 bar	
Outlet pressure total of fuel	69.78 bar	
Intel pressure total of oxidizer	4.48 bar	
Outlet pressure total of oxidizer	66.88 bar	

These data is from rocket H-1C and H-1D for thrust 205,000 pounds [45]. Appropriate US to SI unit conversions have been made.

# Unit 3. Laws of fluid mechanics in turbine applications and design parameters

3.1 Basic principles of fluid mechanics

1. The continuity equation

$$\frac{D\rho}{Dt} + \nabla(\rho w) = 0$$

For pumps where we have practically incompressible flow the equation becomes:

$$A_1 u_1 = A_2 u_2$$

[55]

But in the case of both axial and radial turbines where we have expansion of the exhaust gases, that is, a reduction in the density of the fluid at their exit, the equation becomes:

$$A_1 u_1 \rho_1 = A_2 u_2 \rho_2$$

[56]

#### 2. The momentum equation (Navier – Stokes & Bernoulli equation)

The most generalized form of this equation considering both compressible flow and temperature change is :

$$\rho \frac{Dw}{Dt} + \rho(w\nabla)w = \rho(\frac{Dw}{Dt} + \nabla \frac{w^2}{2} - wX(\nabla Xw))$$

In one-dimensional flow with constant density, it is simplified and the formula takes the following form:

$$\rho u \frac{du}{dx} = -\frac{dp}{dx} + \frac{d}{dx} \left( \mu \frac{du}{dx} \right) + F_2$$

Bernoulli's equation is essentially a specialized and simplified form of the Navier-Stokes equation.

$$\rho \frac{u1^2}{2} + p1 + \rho gz1 = \rho \frac{u2^2}{2} + p2 + \rho gz2$$

[55]

#### 3. The energy equation

It is known that the energy equation results from the balance of motion, pressure, friction and field energy. Therefore, we have the following form:

$$\rho \frac{Dh}{Dt} = \frac{Dp}{Dt} + \nabla(\kappa \nabla T) + \mu \Phi$$

Where  $\Phi$  is the irreversible loss function and is given below :

$$\Phi = (rotw)^{2} + 2div\left(grad\frac{w^{2}}{2} - wXrotw\right) - 2wgraddivw - \frac{2}{3}(divw)^{2}$$

Accordingly, in a one-dimensional form where we have no field forces, the energy equation takes the following form :

$$\rho u \frac{dh}{dx} = u \frac{dp}{dx} + \frac{d}{dx} \left( k \frac{dT}{dx} \right) + \frac{4}{3} \mu \left( \frac{du}{dx} \right)^2$$

An even more simplified form where we have a one-dimensional form and incompressible flow :

$$\frac{{u_1}^2}{2} + \frac{p_1}{\rho} + gz_1 = \frac{{u_2}^2}{2} + \frac{p_2}{\rho} + gz_2 + \frac{\Delta pv}{\rho}$$

Where  $\Delta pv$  the total hydraulic losses.

Correspondingly for air turbines where compressible flow prevails the form of this energy equation is :

$$\frac{{u_1}^2}{2} + \frac{\gamma}{\gamma - 1} \frac{p_1}{\rho_1} = \frac{{u_2}^2}{2} + \frac{\gamma}{\gamma - 1} \frac{p_2}{\rho_2}$$

These equations have a similar form to Bernoulli's [52]

#### 4. 1<sup>st</sup> Thermodynamic axiom

The first law of thermodynamics states that if a system completes a complete thermodynamic cycle during which there is heat input and work output then:

$$\oint dQ - dW = 0$$

In this relation heat and work are the same. For a change of state from 1 to 2 there is the energy change of the system:

$$E_2 - E_1 = \oint_1^2 dQ - dW$$

Where : 
$$E = U + \frac{1}{2}mC^2 + mgz$$

For elementary change :

$$dE = dQ - dW$$

From the first law of thermodynamics, the steady flow energy equation is proved::

$$\dot{Q} - \dot{W}x = \dot{m}[(h_1 - h_2) + \frac{1}{2}(c_1^2 - c_2^2) + g(z_2 - z_1)]$$

Because the contribution of the kinetic energy and the dynamic energy of the fluid is usually very small, they are considered negligible and the relationship becomes :

$$\dot{Q} - \dot{W}_x = \dot{m}(h_{02} - h_{01})$$
 [56]

5. 2<sup>nd</sup> thermodynamic axiom

Included in this axiom is the concept of entropy and ideal thermodynamic processes. A well-known consequence of this law is the Clausius inequality:

$$\oint \frac{dQ}{T} \le 0$$
[25]

If the processes were reversible then :

$$dQ = dQ_R$$

For a finite change of state the entropy change is :

$$S_2 - S_1 = \int_1^2 \frac{dQ_R}{T}$$

But in reality there are no reversible processes so the relationship becomes :

$$\dot{m}(S_2 - S_1) = \int_1^2 \frac{d\dot{Q}}{T} + \Delta S_{irrev}$$

In case where we have an adiabatic change (dQ = 0) we have :

 $S_2 \ge S_1$ 

If hypothetically there was a reversible process then :

$$S_2 = S_1$$
 [56]

#### 6. Euler's law

First, state the torque law applied to a control volume of a centrifugal pump. Let it be fluid entering the control volume with a twist with radius  $r_1$  and tangential velocity  $C_{\theta_1}$  and exiting with radius  $r_2$  and tangential velocity  $C_{\theta_2}$ . In one-dimensional steady flow, the torque will be :

$$T_A = \dot{m}(r_2 C_{\theta 2} - r_1 C_{\theta 1})$$

The pump rotating at an angular velocity  $\Omega$  imparts mechanical work to the fluid in order to convert it into hydraulic work. The mechanical work given through the rotary motion of the rotor is :

$$\dot{W}_c = T_A \Omega = \dot{m} (U_2 C_{\theta 2} - U_1 C_{\theta 1})$$

Therefore, the special project will be :

$$\Delta Wc = \frac{\dot{W_c}}{\dot{m}} = T_A \Omega = (U_2 C_{\theta 2} - U_1 C_{\theta 1})$$

Correspondingly, for the radial air turbine where the fluid gives mechanical work to the rotor, it will be :

$$\Delta W t = \frac{\dot{W}_t}{\dot{m}} = T_A \Omega = (U_1 C_{\theta 1} - U_2 C_{\theta 2})$$

But in the axial air turbine, where we have a constant radius in the rotor body, there is a constant peripheral speed U, and the specific work is :

$$\Delta W t = \frac{\dot{W}_t}{\dot{m}} = T_A \Omega = U(C_{\theta 1} - C_{\theta 2})$$

In general, in turbine engines, the energy equation of the steady flow, which is also known as Euler's equation, is written in this form :

$$\Delta W_x = (h_2 - h_1) = (U_1 C_{\theta 1} - U_2 C_{\theta 2}) \text{ or } \Delta h 0 = \Delta (U C_{\theta}) [55]$$

7. All sizes & perfect gases

For perfect gases the well-known constitutive equation applies :

$$pv = mRT$$

Accordingly, we also have the heat capacities under constant pressure Cp and volume Cv and the relationship that connects them :

$$R = C_p - C_v$$

Total enthalpy :  $h_{01} = h_1 + \frac{C_1^2}{2}$ Total temperature :  $T_{01} = T_1 + \frac{C_1^2}{2Cp}$ Total pressure :  $P_{01} = P_1(1 + \frac{\gamma - 1}{2}M_1^2)^{\frac{\gamma}{\gamma - 1}}$ Total density :  $\rho_{01} = \rho_1(1 + \frac{\gamma - 1}{2}M_1^2)^{\frac{1}{\gamma - 1}}$ Total rotational enthalpy:  $I_{01} = h_1 + \frac{w_1^2}{2} - \frac{U^2}{2}$ 

#### 8. Relationships from Velocity triangles

Radial Pump :

$$C^{2} = C_{r}^{2} + C_{\theta}^{2} + C_{x}^{2} \& C_{\theta} - U = W_{\theta}$$

Axial turbine :

$$C_{\theta 2} - U = W_{\theta 2} \& C_{\theta 3} + U = W_{\theta 3}$$

$$h_2 + \frac{W_2^2}{2} = h_3 + \frac{W_3^2}{2}$$

Radial turbine :

$$C_{m2} = C_{r2} = W_2 \& C_{m3} = C_{x3} = W_3$$

$$h_2 + \frac{W_2^2}{2} - \frac{U_2^2}{2} = h_3 + \frac{W_3^2}{2} - \frac{U_3^2}{2}$$
 [56]

#### 3.2 Inducer design

One of the key factors considered in both conventional pumps and inducers is the suction specific speed coefficient Ss.

$$S_s = \frac{N\sqrt{Q}}{NPSHR^{\frac{3}{4}}}$$

There is also the corrected suction specific speed coefficient, which is a number that appears in a hypothetical impeller with zero shroud diameter at the inlet, operating at the same speed and axial velocity. In this way, correction is made by increasing the flow rate at the inlet to counteract the area blocked by the shroud.

$$S_s^* = \frac{S_s}{(1 - v^2)^{1/2}}$$

Where is v the ratio of the diameter of the hub to the diameter of the inducer (the tip of the fin) [36]

$$v = \frac{D_h}{D_t}[36]$$

Typical values of this ratio range from 0.2 to 0.4 [46]

Another equally important parameter is the flow coffencient  $\Phi$  and it usually takes values from 0.07 to 0.14[21].

$$\Phi_m = \frac{C_x}{U}$$

Similarly, we have for the loading factor  $\psi$ 

$$\Psi = \frac{\Delta C_{\theta}}{U} = \frac{2gH}{U^2}$$

Correspondingly, there is also the flow coefficient at the edge of the blade which is also often used in design.

$$\Phi = \frac{C_x}{U_t}$$

Similarly, we have for the loading factor  $\psi$ 

$$\Psi = \frac{2gH}{U_t^2}$$

#### 3.2.1 Inducer blade design

The most effective method preferred by designers for shaping the impeller geometry is the method of free vortex. Essentially, the blade is shaped in such a way that the product  $rC_{\theta}$  remains constant from the base to the tip of the blade. Although it is a reliable method, it presents manufacturing difficulties.

Usually in inductors the radial lead is constant for flat blades but in practice it is not done by construction with method of free vortex :

$$rtan(\beta) = R_t tan(\beta_t)$$

Where Rt and  $\beta$ t are the inducer radius and the vane tip angle  $\beta$  respectively.

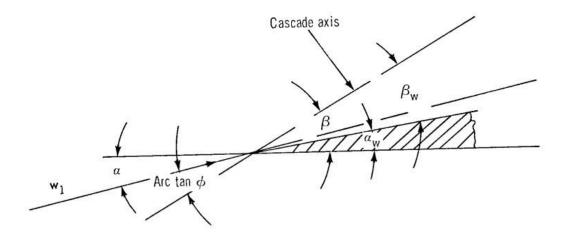
Radial lead :  $\lambda = 2\pi rtan(\beta)$ 

Another characteristic element for impeller design is the aspect ratio of the blade, defined as the chord length of the blade divided by the axial distance of the blade. It is known as the solidity ratio and significantly contributes to the effectiveness of cavitation reduction. The larger the value, the greater the protection against cavitation. Typical values for better suction performance from 2 to 2.5 at all blade sections from tip to hub[46]. Solidity is given by:

$$\sigma = \frac{C}{S}$$

Where C chord length and S spacing blade[46].

The blade profile must not interfere with the free boundary of the cavitation flow, i.e., the blade must remain within the cavity and operating conditions. The figure below shows the wedge, wing and  $\beta w$  angles.



Picture 31 Here are the fluid flow angle, the wedge angle and the vane angle [46]

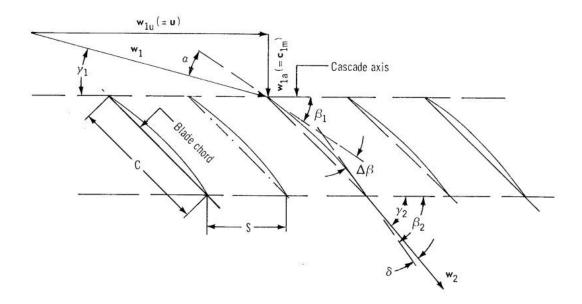
The wedge angle is calculated from the following relationship :

$$a_w = \beta - \beta_w$$

The angle  $\beta w$  respectively :

$$\beta_w = \arctan(1, 1\Phi_d)$$
[29]

Where  $\Phi_d$  the design flow coefficient [46].



Picture 32 Velocity triangles, fluid flow angles, vane angles and inducer inlet and outlet recall angles[46]

#### 3.2.2 Cavitation

Normally the allowable suction head should be less than the available suction head NPSH prevailing at the inlet of the inducer.

$$NPSH_{required} \leq NPSH_{available}$$

Then follows the required suction head which is determined by a pump running an ideal flow (approximate from cold water) and suction coefficient  $S_s^*$ :

$$NPSH_{required} = \left(\frac{n\sqrt{Q'}}{S_s^*}\right)^{4/3}$$

Where Q' is a corrected flowrate

The available tank height is at the inducer inlet is calculated from the following relationship :

$$NPSH_{available} = \frac{P_{total} - P_{v}}{g\rho_{F}}$$

Forthermore, where is thermodynamic suppression head TSH

The thermodynamic suppression height can be estimated from the equation below :

Where NPSH<sub>ideal fluid</sub> is the net height of the tank without the hydraulic losses

$$NPSH_{ideal\ fluid} = \left(\frac{P_{total} - P_{v}}{g\rho_{F}}\right)tank - H_{loss}$$

In addition, the estimate of TSH can be estimated by a coefficient called the NPSH coefficient and denoted by Z. To calculate it, the following relationship is given :

$$Z = \frac{2g(NSPH)_{tank}}{C_m^2}$$

Average of this coefficient of TSH height is calculated from the following relationship :

$$(TSH) = \frac{(Z_{opt} - Z)C_m^2}{2g}$$

Where Zopt is the optimal NPSH factor. In this way, the NPSH<sub>required</sub> can be calculated accordingly :

$$NPSH_{required} = NPSH_{tank} + (TSH) = \frac{Z_{opt}C_m^2}{2g}$$

Where :

$$Z_{opt} = 3(1 - 2\Phi_{opt}^2)$$

For very small  $\Phi_{opt}$  we consider  $Z_{opt} \approx 3[46]$ .

In order to analyze in greater depth the phenomenon of cavitation in the inducer blades, models and empirical relationships have been developed to describe the geometry and dimensions of the bubbles and the frequency with which they appear and destroy.

The relationship that describes the growth rate of the bubble radius is a well-known equation Rayleigh – Plesset :

$$\rho \left[ R\ddot{R} + \frac{3}{2}\dot{R}^2 \right] = \left[ p_v - p_\infty(t) \right] + p_{go} \left( \frac{R_0}{R} \right)^{3k} - \frac{2S}{R} - 4\mu \frac{\dot{R}}{R}$$

where k is the multimodal coefficient. It takes the value 1 in case of isothermal change. Below is the law relating the pressure of the gas inside the bubble and its radius.

$$p_g R^{3k} = p_{g0} R_0^{3k}$$

This relationship essentially presents the thermodynamic behavior of the gas prevailing inside the bubble. Usually, this law contributes greatly to the increase of the bubble radius in cases where we have heat transfer which does not practically occur in the pump switch. However, there are two factors in the Rayleigh–Plesset equation that reduce bubble growth. The first is the surface tension that develops on the surface of the bubble. These stresses are described by a stress coefficient denoted S and measured in units of N/m. The second is forces due to the potential viscosity of the liquid itself.

The aforementioned forces contribute effectively to reducing the bubble for small radii. For sufficiently large bubbles these forces are considered negligible. Therefore the Rayleigh – Plesset equation is simplified and takes the following form :

$$\rho\left[R\ddot{R} + \frac{3}{2}\dot{R}^2\right] = \left[p_v - p_{\infty}(t)\right]$$

If the static pressure is also considered constant with respect to time, the following relationship results :

$$\dot{R}^{2} = \frac{2}{3} \frac{p_{v} - p_{\infty}}{\rho} \left[1 - \left(\frac{R}{R_{0}}\right)^{3}\right]$$

In addition, there are other possibilities and conditions in which cavitation in the inducer can prevail which simplify the Rayleigh – Plesset equation. A characteristic case is bubble equilibrium where the rate of increase of the bubble radius is zero (R=0), the pressure is constant with respect to time and we have isothermal gas transport. With these conditions the Rayleigh – Plesset equation takes the following form :

$$p_{\infty} = p_{g0} [\frac{R_0}{R}]^3 + p_{\nu} - \frac{2S}{R}$$

But the pressure resulting from this relationship differs inside and outside the bubble as different surface tensions prevail. It is very important to mention that the bubble equilibrium is not always stable. There is a minimum pressure that the liquid can reach and it is below the pressure pv. This pressure is called critical pressure and is denoted pc. Corresponding to this pressure we also have the corresponding radius of the bubble Rc.

$$p_c = p_v - \frac{4S}{3R_c}$$
$$R_c = \sqrt{\frac{3p_{g0}R_0^3}{2S}}$$

#### Bubble development

The rapid growth of bubbles requires the condition  $p\infty < pv$ , i.e. the static applied pressure is less than the vapor pressure. In this case R>R0 will hold and the bubbles will be so large enough in size that shear stresses and viscous forces are considered negligible. With these conditions the growth rate of the bubble will be:

$$\dot{R} \cong \sqrt{\frac{2(p_v - p_\infty)}{3\rho}}$$

This relationship shows 2% less error than the original one.

#### Collapse of bubbles

In a turbopump where bubble collapse is required, the static applied flow pressure must be greater than the vapor pressure (  $p\infty > pv$ ). In this case we have the destruction of the bubble and R<R0. If viscous forces and shear stresses are neglected the rate of bubble reduction will be :

$$\dot{R} \cong \sqrt{\frac{2(p_v - p_\infty)}{3\rho}} \left[ \left(\frac{R_0}{R}\right)^3 - 1 \right]$$

Until the bubble is completely destroyed, it takes a certain amount of time, which is called the Rayleigh time. It is the time where R=0. This is estimated from the following relation:

$$\tau_p \cong 0.915 R_0 \sqrt{\frac{\rho}{p_\infty - p_\nu}} \, [57]$$

A cavitation number Kc, based on the cavity pressure instead of the liquid bulk vapor pressure, has been defined :

$$K_c = \frac{p_s - p_v}{\frac{1}{2}\rho w_1^2}$$

where  $p_s$  fluid static pressure,  $p_v$  fluid vapor pressure in cavity at leading edge,  $\rho$  fluid density,  $w_1$  fluid velocity relative to blade at tip and g gravitational constant.

he cavity velocity  $w_c$ , follows from Bernoulli's law and the definition of the cavitation number, giving :

$$w_c = w_1 \sqrt{1 + K_c}$$

Next, the relative speed at the output of the inducer is calculated :

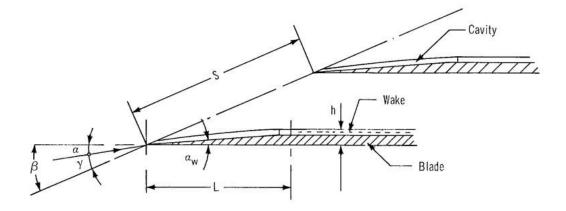
$$w_2 = \frac{w_c}{F + (F^2 - 1)^{1/2}}$$

Where F is a coefficient calculated from the following relationship :

$$F = \frac{(1 + K_c)^{1/2} \sin(\beta) + (1 + K_c)^{-1/2} \sin(\beta - 2\alpha)}{2\sin(\beta - \alpha)}$$

To find the cavity height that develops above the fin the cavitation ratio the following relationship is used :

$$\frac{h_c}{S} = \sin\beta - \frac{w_1}{w_2}\sin\left(\beta - \alpha\right)$$



Picture 33 Blade in cavity. Cavitation develops on the flap [46]

The minimum value of the cavitation coefficient K can be estimated from the following equation :

$$K_{min} = \frac{2sinasin(\beta - \alpha)}{1 + cos\beta}$$
 [46]

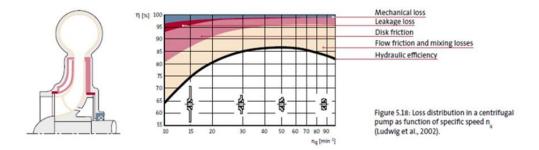
#### 3.3 Impeller design

There are 2 types of fenders, open and padded. The hydraulic efficiency of the jacketed impellers is generally higher as the deformations in the shell are more tolerable and smooth without causing particular problems in the gap between the impeller and the wall[47]. In general, although the geometric characteristics of an impeller contribute significantly to the

hydraulic degree of efficiency, the most basic quantity by which the efficiency is determined is the specific number of revolutions Ns [25].

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$
 (*rpm*, *gpm*, *ft*) or  $n_q = \frac{N\sqrt{Q}}{H^{3/4}}$  ( $\frac{1}{min}, \frac{m^{3}}{s}, m$ ) [48]

Below is a graph of hydraulic efficiency for turbopumps as a function of specific speed nq



Picture 34 Loss breakdown in centrifugal pumps. From this diagram it is easy to start the design of a centrifugal pump or an impeller-converter system based on the degree of efficiency [49]

Another equally important parameter is the specific diameter Ds of the impeller

$$D_s = \frac{DH^{1/4}}{\sqrt{Q}}$$
 or  $\delta = \frac{\psi^{1/4}}{\phi^{1/2}}$  [48], [25]

Also, as with the commutator, we also have the loading factor  $\psi$  in the impeller

$$\psi = \frac{2gH}{U^2}$$

Accordingly, we also have the flow coefficient  $\Phi$ 

$$\Phi = \frac{C_x}{U}$$

In this case the flow coefficient can take values usually from 0.05 to 0.3. The criteria for selecting the flow coefficient at the inlet and outlet are different. At the inlet the flow coefficient  $\Phi 1$  depends on the required suction while the flow coefficient  $\Phi 2$  is determined by the vane angle at the outlet and the head coefficient[36]. Typicals value head coefficient is 0,5 to 0,8 [50].

Other design parameters that are known and common in radial pumps are:

- 1)  $\frac{D_{h1}}{D_{S1}}$ : The ratio of hub diameter to vane diameter at the impeller inlet.
- 2)  $\frac{b_2}{D_2}$ : The ratio of vane height in the discharge area to the impeller outlet diameter.
- 3)  $\frac{D_{s1}}{D_2}$ : The ratio of vane diameter to the impeller outlet diameter [48].

#### 3.3.1 Number of blades

The number of blades in an impeller is highly dependent on the head coefficient. Usually for small head coefficients the number of blades is from 3 to 5 and for large coefficients the number of blades exceeds 20. Of course, the number of blades varies depending on the geometric characteristics of the impeller and the slip coefficient M which will be mentioned below . Another factor for the number of vanes is the flow coefficient as the diffusion limits of the impeller (surface dial velocity) must be respected[36]. Alternatively the number of vanes can be calculated from the following formula generally used in pumps:

$$Z = K_{z} \frac{d_{2} + d_{1}}{d_{2} - d_{1}} sin(\frac{\beta_{1} + \beta_{2}}{2})$$

Where  $K_z$  is from 5 to 6,5[48].

It was mentioned above that an equally important factor for the number of blades is the slip coefficient M. This essentially represents in percentage terms how close the actual speed of the  $C_{\theta}$  fluid is to the theoretical one.

$$M = \frac{C_{\theta}}{C_{\theta\infty}} \ [48]$$

Theoretically, if we had an infinite number of blades, we would have the theoretical speed, that is, the coefficient would be M = 1.

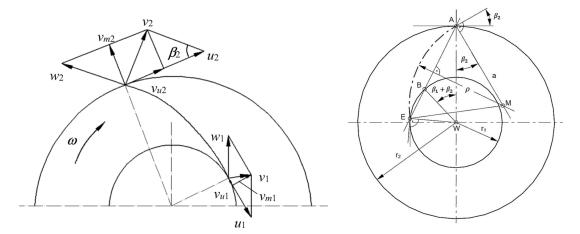
There is no universal equation that is widely applicable to calculate the drag coefficient in every turbine. However, several empirical equations that are acceptable have been developed and used to calculate the M factor[36].

## 3.3.2 Wing design

The blade design for a centrifugal impeller is known to be based on the single circular arc :

$$R = \frac{1}{2} \frac{r_2^2 - r_1^2}{2r_2 COS(\beta_2) - 2r_1 COS(\beta_1)}$$

There is also the method with the double circular arc, but it is not preferred as with this method we have a discontinuity in the geometry and a lower efficiency of the wing.



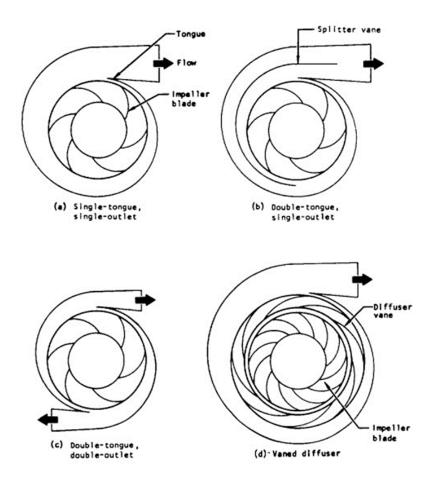
Picture 35 The gear triangles at the inlet and outlet of the centrifugal impeller[31] Picture 36 The arc with which the impeller blade is formed [51]

Usually for centrifugal liquid pumps the method by which the vane is designed is by the linear distribution of the relative velocity W = W(r) with respect to the radial position r.

$$W = W_1 + (W_2 - W_1) \frac{r - r_1}{r_1 - r_2} [55]$$

#### 3.4 Volute – Diffuser design

The volute - diffuser in the pumps essentially helps to smooth out and increase the pressure at the exit of the flow. An asymmetric volute cross section is preferred because it produces a single vortex thut is stable and that improves the efficiency of the conical diffuser at the exit. The conical diffuser at the volute exit will operate efficiently when the included angle forcircular cross sections is between 7° and 9°; for square cross sections, 6°; and for two parallel walls, 11 °[36]. The diffuser depending on the design of the turbopump takes the following forms:



Picture 37 Types of diffusers [36]

Generally, a good method for spiral design is with principle of conservation of angular momentum ( $C_{\theta r}$  = constant). In a classic diffuser without static vanes with a single outlet the flow is calculated from the following relation:

$$Q = aNA_{th}C_{th}$$

Where  $\alpha$  is a correction factor associated with the boundary layer displacement and a typical value is close to 0.9. The number N is the number of outputs of the diffuser or otherwise

bulbs. The velocity  $C_{th}$  at the diffuser throat can be related to other velocities by the constant momentum law.

$$C_{th}r_{th} = C_{\theta,0}r_{\theta,0} [47]$$

In all spiral diffusers, regardless of their type, there is a limit to the diffusion coefficient D. For single-stage diffusion, D $\leq$ 0.6 applies and is calculated from the relationship:

$$D = \frac{P_s - P_{min}}{P_t - P_{min}} [36]$$

#### 3.4.1 Logarithmic spiral design

This is how the static fins of the diffuser analyzed below are formed :

$$\varphi - \varphi_1 = tan\beta \ln\left(\frac{r}{r_1}\right)$$

Where  $tan\beta = \frac{C_{\theta}}{C_r}$  [47].

### 3.4.2 Volute -Diffuser with fins

In order to increase the overall efficiency of the pump inside the diffuser, special static vanes are added which further increase the pressure in the fluid discharge area. The pressure distribution at the outlet is also normalized and eddies and vortices are reduced. Another additional purpose of using static vanes is to avoid the amplification of wave pressure caused between the diffuser and the impeller blade. The superposition of these waves may cause large swings in the discharge pressure. Therefore, for a vane diffuser to be effective, it must have a certain number of vanes Zb that corresponds correctly to that of the vane Z2 taking into account as parameters the mean distance Dv and the relatively low flows. In order to identify these amplifications, a special index m is used. When it becomes an integer, then we have a wave amplification. This indicator is calculated in 2 cases.

I. Case : Av  $Z_2 > Z_b$   $m = j \frac{Z_2}{Z_b} \left\{ \frac{Z_b - Z_2}{Z_2} + \frac{\pi D_v N}{(a + W)} \right\}$ II. Case : Av  $Z_b > Z_2$  $m = j \frac{Z_2}{Z_b} \left\{ \frac{Z_b - Z_2}{Z_2} - \frac{\pi D_v N}{(a + W)} \right\}$ 

Where :

j : order of the harmonic of the fundamental wave frequency

Dv : average distance from center of pump to center of volute passage

a : velocity of sound in liquid

W :average relative velocity of fluid in volute passag

However, the main disadvantage of the diffuser with static vanes is that at large changes in the flow, the efficiency decreases sharply and the operating range is very small. In contrast to the classic diffuser that does not have fins, it shows high performance in a wider range of supply operation [36].

#### 3.4.3 Force due to uneven pressure distribution

Precisely because there are heterogeneities in the pressure and momentum peripherally, we have a logarithmic spiral shape in the diffuser. But these heterogeneities cause unequal radial forces on the impeller. Diffusers with one bulb (output duct) usually have this effect. For this reason, in the design, some diffusers may have 2 or more bulbs in order to achieve the balancing of the radial forces. This radial force is calculated from the following relationship:

## $F = k_F g_0 \rho H D_0 b_0$

Where  $k_F$  is a ratio factor and is determined theoretically and experimentally. Studies have shown that for a single bulb diffuser near shutdown conditions the highest values are from 0.3 to 0.65. However, during the operation of the pump we have values of the order of 0.1. This factor may be constant throughout its operating range [47].

#### 3.5 Turbine design

The air turbine is the one that ensures the necessary rotational work so that the pumps operate at the desired speeds. Although in recent years electric motors have also been developed for this work, they are still in the preliminary stage and are not so prioritized as the weight of the batteries and the electric motor are too high for space applications. Usually air turbines are axial single-stage or two-stage. But there are rarer cases where we have radial turbines which are also very efficient. The inlet operating temperature varies according to the load and the material of which the air turbine is made.

As in pumps, in air turbines we have the flow coefficient and the loading coefficient respectively :

$$\Phi = \frac{C_x}{U}$$
$$\Psi = \frac{\Delta C_\theta}{U}$$

Next, we have the degree of reaction R which represents the rate of reduction of the enthalpy of the fluid where it is done by the rotor. Usually for this application we have prices of 50% and above.

$$R = \frac{h_2 - h_3}{h_1 - h_3} = 1 - \frac{\Phi}{2} (tana_2 - tana_1) [2]$$

Other equally important design parameters are the total pressure ratio :

$$\Pi_{tt} = \frac{P_{02}}{P_{03}} \ [48]$$

The total temperature intel  $T_{02}$  and the total pressure intel  $P_{02}$ . Also another quantity used in the design of a turbine is the isentropic speed ratio :

$$v_{tt} = \frac{U}{C_o} [48]$$

### 3.5.1 Turbine losses and efficiency ratings

The turbine losses are mainly due to the geometrical characteristics of the blade, the boundary layers developed on the walls and the enthalpy losses. These losses are expressed by a factor  $\zeta$ . We have losses in the stator and the rotor of the turbine therefore we have the factors  $\zeta_N$  and  $\zeta_R$  respectively. The following relations state the enthalpy losses :

• For the stator

$$h_2 - h_{2s} = \zeta_N \frac{1}{2} C_2^2$$

• For the rotor

$$h_3 - h_{3s} = \zeta_R \frac{1}{2} W_3^2$$

In air turbines, as in any turbine engine, three different degrees of efficiency can be distinguished :

Isentropic degree of efficiency total to total

The isentropic overall efficiency of the turbine depends on the aforementioned losses.

$$n_{tt} = \frac{h_{01} - h_{03}}{h_{01} - h_{03ss}} \cong \left[ 1 + \frac{T_{03}}{T_3} \frac{\zeta_N C_2^2 T_3}{T_2} + \zeta_R W_3^2}{2(h_{01} - h_{03})} \right]^{-1}$$

This essentially represents the actual work produced relative to the theoretical enthalpy change of the fluid.

Isentropic overall to static efficiency

The overall to static efficiency of the turbine is calculated respectively from the following relationship :

$$n_{ts} = \frac{h_{01} - h_{03}}{h_{01} - h_{3ss}} \cong \left[ 1 + \frac{\frac{\zeta_N C_2^2 T_3}{T_2} + \zeta_R W_3^2 + C_3^2}{2(h_{01} - h_{03})} \right]^{-1}$$

However, this degree of efficiency is not particularly interesting for air turbines in general, but more so for compressors.

Multimodal efficiency

The multimode efficiency represents the efficiency of a hypothetical very small air turbine stage regardless of how many stages the actual air turbine has. Essentially, it is the degree of efficiency of an elementary change during the expansion of the fluid. It is different from the isentropic degree of efficiency and is due to the pressure ratio of the air turbine.

$$n_p = \frac{\delta W_{min}}{\delta W}$$
$$n_p = \frac{dh}{dh_{th}} = \frac{v dp}{C_P dT}$$

In the air turbine in an adiabatic change taking into account the multimodal degree of efficiency has the following form:

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{n_p(\gamma-1)/\gamma}$$

Accordingly, the equation that connects the isentropic degree of efficiency of the turbine with the multimodal degree of efficiency is derived.

$$n_t = \frac{1 - \left(\frac{p_2}{p_1}\right)^{n_p(\gamma - 1)/\gamma}}{1 - \left(\frac{p_2}{p_1}\right)^{(\gamma - 1)/\gamma}}$$

### 3.5.2 Design blades

Depending on the geometrical dimensions of the turbine and the construction requirements, certain blade design methods have been developed for axial turbines which are applicable to compressor and fan blades respectively. These methods are used with the main goal of achieving the radial balance of the fluid, i.e. all constitutive centrifugal forces exerted on an elementary rotating fluid are balanced by the pressure forces.

$$(p+dp)(r+dr)d\theta - prd\theta - \left(p + \frac{1}{2}dp\right)drd\theta = \frac{dmC_{\theta}^{2}}{r}$$

For  $dm = \rho r d\theta dr$  and ignoring the second-order terms the relationship is simplified :

$$\frac{1}{\rho}\frac{dp}{dr} = \frac{C_{\theta}^2}{r}$$

Assuming that the  $C_{\theta}$  velocity and density are known and with the following assumptions,

✓ 
$$h_0 = h + \frac{1}{2}(C_{\theta}^2 + C_{\chi}^2)$$
: Radially stable (dho/dr=0)  
✓ S: Radially stable (ds/dr=0)

and combining the radial equilibrium equations with enthalpy, with entropy and with the original equation the following radial equilibrium relation for adiabatic and ideal machines is obtained :

$$C_x \frac{dC_x}{dr} + \frac{C_\theta}{r} \frac{d}{dr} (rC_\theta) = 0$$

 FREE VORTEX METHOD : With this method, practically along the wing from its base to the top, the product C0r is constant at the inlet and outlet. This method has high manufacturing accuracy and is reliable. However, the main disadvantage of this method is that for very low foot-head ratios (very long fins) sharp changes in the fluid exit angle are caused. Additionally at flows outside the design operating range twist flows can be induced. Thus, construction becomes more difficult and expensive, and significant pressure losses occur. Suppose there is a circulation of fluid C.

$$\Gamma = \oint C ds$$

Turbulence is defined as the elementary circulation  $\delta C$  toward an elementary surface  $\delta A$  to which it is applied.

$$\omega = \frac{d\Gamma}{dA}$$

Where  $d\Gamma$  is :

$$d\Gamma = (C_{\theta} - dC_{\theta})(r - dr)d\theta - C_{\theta}rd\theta = \left(\frac{dC_{\theta}}{dr} + \frac{C_{\theta}}{r}\right)rd\theta dr$$

If the lower order terms are ignored, the following relationship will result:

$$\frac{d\Gamma}{dA} = \frac{1}{r} \frac{dC_{\theta}r}{dr}$$

Therefore we have :

$$\frac{dC_{\theta}r}{dr} = 0 \Longrightarrow C_{\theta}r = K$$

Where K constant term.

2) FORCED VORTEX METHOD: In this case the speed C $\theta$  varies proportionally with the radius r.

$$C_{\theta} = K_1 r$$

By substitution in the last radial equilibrium relation we have :

$$\frac{d}{dr}\left(\frac{C_{x1}^2}{2}\right) = -K_1 \frac{d}{dr}(K_1 r^2)$$

It results from integration

$$C_{x1}^2 = \alpha - 2K_1^2 r^2$$

Where a constant term. We have a fixed distribution of work radially.

3) VORTEX VARIABLE METHOD: With this method the tangential velocity of the fluid has the following distribution at the entrance and exit of the rotor :

$$C_{\theta 1} = ar^n - \frac{b}{r}$$
$$C_{\theta 2} = ar^n + \frac{b}{r}$$

For a value of n = 1, a constant degree of radial reaction R is practically achieved.

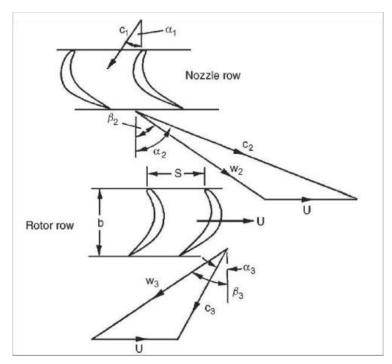
4) MIXED VORTEX METHOD: With this way of design, the disadvantages and difficulties presented by the free vorticity mentioned above are dealt with. Essentially, it is a method that combines free vorticity with forced vorticity.

$$C_{\theta 2} = \frac{a}{r} + br$$

In this case the vorticity C $\theta$ r varies parabolically with the radius r [56].

## 3.5.3 Rotor and stator flow in an axial flow turbine

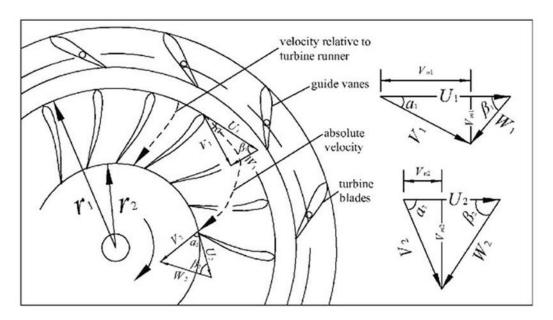
The following images describe the flow of the fluid and how the speed relationships are connected to the calls of the vanes at the inlet and outlet.



Picture 38 Velocity triangles at the inlet and outlet of an axial turbine [52]

## 3.5.4 Rotor and stator flow in a radial air turbine

Below it is shown that the exhaust gases get the proper call from the static diffuser vanes to enter the radial air turbine and are removed from them.



Picture 39 Velocity triangles at the inlet and outlet of an radial turbine[53]

In these applications the pressures and forces are very high strength relationships for tension and bending are given below.

$$\sigma = E\varepsilon$$

Where  $\varepsilon = \frac{\Delta l}{l}$  and E the modulus of elasticity

 $T = G\gamma$ 

Where  $\gamma$  shear angle and G the modulus of shear

[57]

For an axial turbine blade the elemental force is:

$$dF_c = -\Omega^2 r dm$$

Where :  $dm = \rho_m A dr$ 

Accordingly the elementary voltage will be :

$$\frac{d\sigma_c}{\rho_m} = \frac{dF_c}{\rho_m A} = -\Omega^2 r dm$$

For fins of fixed and conical cross-section the developing stresses are:

$$\succ \quad \frac{\sigma_c}{\rho_m} = \Omega^2 \int_{r_h}^{r_t} r dr = \frac{U_t^2}{2} \left[ 1 - \left(\frac{r_h}{r_t}\right)^2 \right]$$

$$\succ \quad \frac{\sigma_c}{\rho_m} = \frac{KU_t^2}{2} \left[ 1 - \left(\frac{r_h}{r_t}\right)^2 \right]$$

Where K is stress factor ratio of cross-sectional reduction [56].

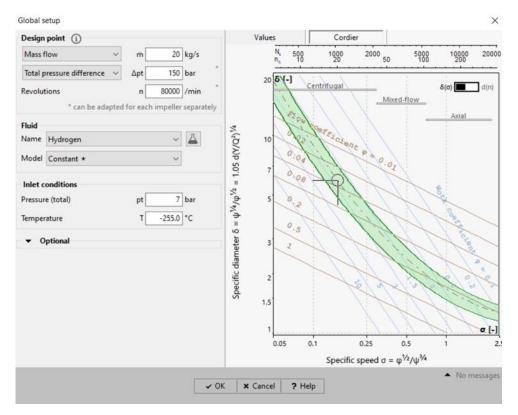
# Unit 4 Turbopump design

4.1 Fuel pump design

The fuel is liquid hydrogen in cryogenic stability. With the criterion that the specific speed should be in the range of 1000 to 2000 to have a high efficiency and the difference of total pressure revs and supply were adjusted. The following images show the design point.

ilobal setup			1
Design point (j)	Values Cordier	•	
Mass flow v m 20 kg/s	General machine type: Centrifugal	(high pressure)	
Total pressure difference Revolutions * can be adapted for each impeller separately			
luid	specific	speed	
Name Hydrogen v A	Specific speed (EU)	ng 24	11
Constant *	Specific work		E5 m <sup>2</sup> /s <sup>2</sup>
nlet conditions	Power output	PQ 40	69 kW
ressure (total) pt 7 bar	Volume flow	Q 9	77 m³/h
T -255.0 °C	Head	H 207	50 m
✓ Optional			
~ 0	OK X Cancel ? Help	•	No messag

Picture 40 The design point of the LH2 pump CFturbo program



Picture 41 The design point of the  $LH_2$  pump CFturbo program

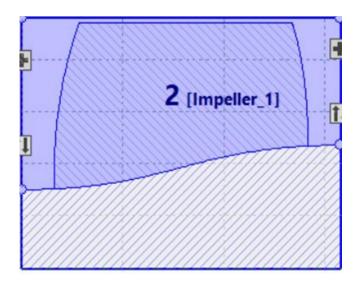
The next step is determining the percentage pressure difference of the inducer and the impeller.

umber of stages (impeller)		F	2 🗘		Values Meridian			
moer or sugges (impenel)		-			✓Project			
[Impeller_1]	¥				Required driving power	PD	4727	
- Impeller shape		Axial			Efficiency	η	86.1	%
- Impeller type		Inducer			✓[Impeller_1]: Global values			
- Energy fraction		modeen	10	%	Specific speed (EU)	nq	135.5	
Pressure difference				bar	Design flow rate	Q*	1131	m³/h
	∆pt		15	bar	Work coefficient	ψ	0.201	
[Impeller_2]					Specific diameter	δ	2.473	
<ul> <li>Impeller shape</li> </ul>		Centrifugal			Total flow coefficient	φt	0.073	
- Impeller type		Standard			Meridional flow coefficient	φm	0.092	
- Energy fraction			90	%	Meridional velocity ratio	cm2/cm1	1.251	
- Pressure difference	∆pt		135	bar	Relative velocity ratio	w2/w1	0.921	
					Inlet area	A1	8230	mm
					Outlet area	A2	6580	mm
					Area ratio	A2/A1	0.8	
					Axial force (thrust)	Fax	11110	Ν
					~[Impeller_1]: Reynolds numbers			
					Reynolds number (d1)	Re(d1)	2.104E8	
					Reynolds number (b1)	Re(b1)	7.37E7	
					Reynolds number (d2)	Re(d2)	2.104E8	
					Reynolds number (b2)	Re(b2)	5.03E7	
					~[Impeller_1]: Power			
					Torque	T	64	Nm
					Required driving power	PD	537	kW
					Required power incl. motor losses	PR	671	kW
					Power loss	PL	129.6	kW
					√[Impeller_1]: Stage efficiency			

Picture 42 Pressure differential rates of the inducer and the impeller CFturbo program

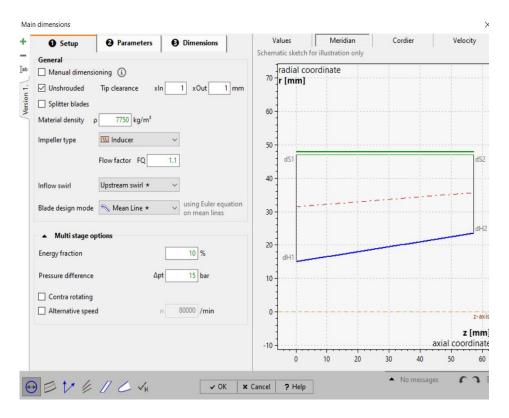
The 10% of total pressure is from inducer and the 90% from impeller. After the design point of the turbopump has been properly defined, the design of the inducer begins.

## 4.1.1 Inducer

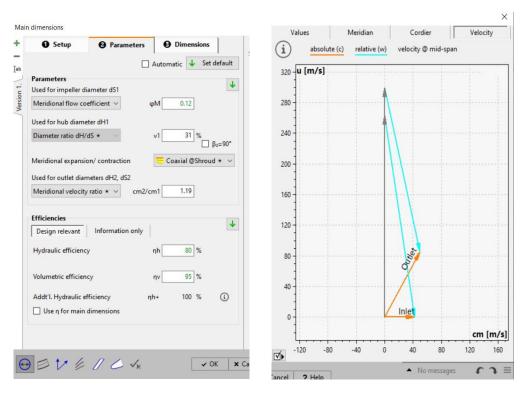


Picture 43 Inducer CFturbo program

The Design of inducer was based the coffecient flow, the ration  $v_1$  and the ration of axial velocities. The tip clearance will be considered 0.1mm but the price is 1mm because to make the CFD analysis easier.



Picture 44 Start of set up of inducer CFturbo program



Picture 45 The design parameters CFturbo program

Picture 46 Inducer input and output speed triangles CFturbo program

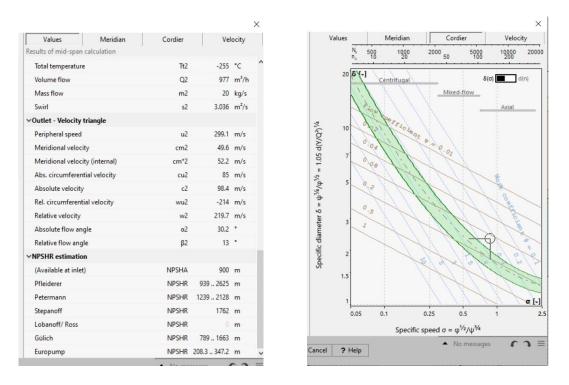
Setup     Parameters     Dimensions	Values Meridian	Cordier	Velocity
Main dimensions	Results of mid-span calculation		
Calculate Automatic	✓Global values		
Inlet	Work coefficient	ψ	0.253
Hub diameter inlet dH1 30 mm	Specific diameter	δ	2.208
	Total flow coefficient	φt	0.103
Shroud diameter inlet dS1 95.8 mm dTip = 93.8 mm	Meridional flow coefficient	φm	0.124
Outlet	Meridional velocity ratio	cm2/cm1	1.188
Hub diameter outlet dH2 47 mm	Relative velocity ratio	w2/w1	0.824
	Inlet area	A1	6500 mn
Shroud diameter outlet dS2 95.8 mm dTip = 93.8 mm	Outlet area	A2	5470 mn
	Area ratio	A2/A1	0.842
	Axial force (thrust)	Fax	8980 N
	✓Reynolds numbers		
	Reynolds number (d1)	Re(d1)	1.678E8
	Reynolds number (b1)	Re(b1)	5.76E7
	Reynolds number (d2)	Re(d2)	1.678E8
	Reynolds number (b2)	Re(b2)	4.273E7
	∨Inlet - Flow properties		
	Density	p1	73.7 kg/
	Static pressure	p1	6.36 bar
	Temperature	T1	-255 °C
	Total density	pt1	73.7 kg/
	Total pressure	pt1	7 bar

Picture 47 Input and output dimensions of the inducer and the all values CFturbo program

Peripheral speed	u2	299.1	m/s
VOutlet - Velocity triangle			
Swirl	s2	3.036	m²/s
Mass flow	m2	20	kg/s
Volume flow	Q2	977	m³/h
Total temperature	Tt2	-255	°C
Total pressure	pt2	22	bar
Total density	pt2	73.7	kg/m <sup>3</sup>
Temperature	T2	-255	°C
Static pressure	p2	18.43	-
Density	p2	73.7	kg/m <sup>3</sup>
Outlet - Flow properties	P.		
Relative flow angle	ß1		
Absolute flow angle	a1	90	
Relative velocity	wu1	266.8	
Absolute velocity Rel. circumferential velocity	c1 wu1	-263.5	
Abs. circumferential velocity	cu1		m/s
Meridional velocity (internal)	cm*1		m/s
Meridional velocity	cm1		m/s
Peripheral speed	<b>u</b> 1	263.5	
Inlet - Velocity triangle			
Results of mid-span calculation			
Values Meridian	Cordier	Vel	locity

Picture 48 The values of all velocities, pressures and flow angles at the inlet and outlet of the inducer CFturbo program

Picture 49 The values of all velocities, pressures and flow angles at the inlet and outlet of the inducer CFturbo program



Picture 50 The values of all velocities, pressures and flow angles at the inlet and outlet of the inducer CFturbo program

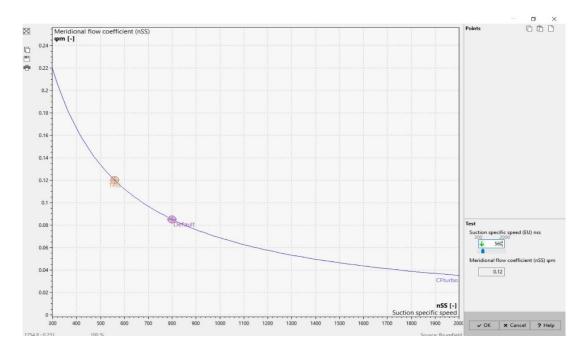
#### Picture 51 Inducer design point CFturbo program

In the set up for pump according the Pfleiderer the NPSHR is bigger than NPSHA but for inducer the NPSHR is smaller and below an estimate is made for the NPHR calculation from the set up data and from the flow coefficient curve.

NPSHR = 
$$\lambda_c \frac{C_{m1}^2}{2g} + \lambda_w \frac{W_1^2}{2g}$$

- For  $C_{m1} = 41.7 m/s$ ,  $\lambda_c = 1.1$ , ,  $\lambda_w = 0.03$  and  $W_1 = 266.8 m/s$ : NPSHR = 206 m
- For  $C_{m1} = 41.7 \text{m/s}$ ,  $\lambda_c = 1.35$ ,  $\lambda_w = 0.06$  and  $W_1 = 266.8 \text{m/s}$ : NPSHR = 337 m

A range of NPSHR height was estimated for all cases. The available suction height is above these values. Therefore, the possibility of cavitation is very small. Below is the curve of nss versus flow coefficient  $\varphi m$ .



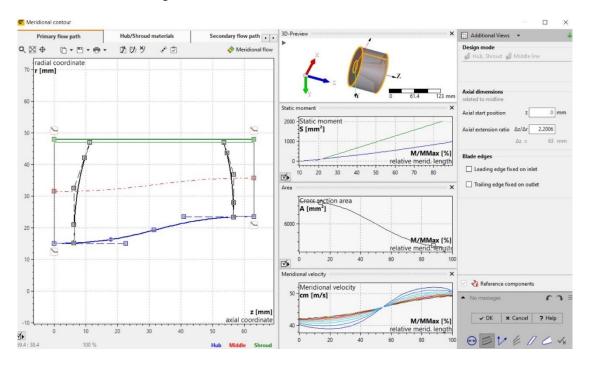
Picture 52 Flow coefficient function curve with specific speed number CFturbo program

For  $\phi_m=0,12$  the  $n_{ss}=560.$  The suction specific speed  $n_{ss}$  is calculated from the following relationship :

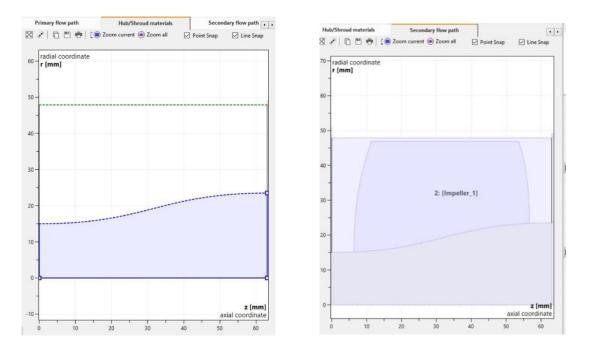
$$n_{ss} = \frac{n\sqrt{Q}}{NPSHR^{3/4}}$$

From data and the function the NPSHR = 313 m < NPSHA = 900 m

Below is the meridional contour. Minimal changes have been made with B-spline curves. The axial coordinate adjusted to be 63 mm. The last diagramma shows the the axial velocity distribution over the wing.



Picture 53 Meridian contour and B-spline configuration CFturbo program



Picture 54 The shaft material (shaft shape) CFturbo program

Picture 55 Picture of inducer CFturbo program

The blade shape was made with base the methods Radial elements 3D. The thickness is 1mm, 0,1mm higher than default. The number of blades in an inducer, usually, 3. The iRelhub is default and the appeal  $\delta$  is corrected from the number solidy l/t, which will be analyzed below.

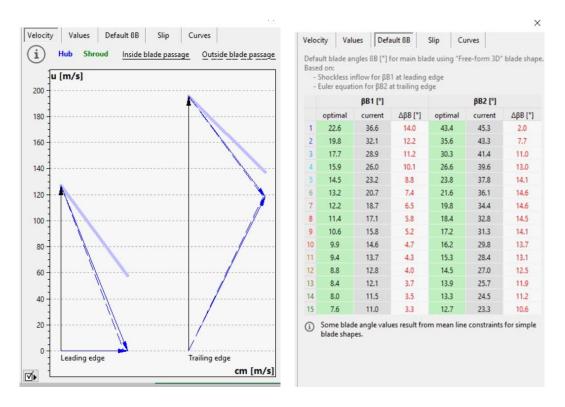
O Blade setup 🛛 Spans	Cu,cm	Blade angles	(i)		1 (Hub)	Span = 15	5 (Shroud)		
•   • •		1 1		Leading edge	Trailing edge	Leading edge	Trailing edg		
Blade shape	Blade thicknes		z	6.38	56.7	11.35	53.6		
Free-form 3D" will provide best res	ults To consider blac	le blockage 🔹 🗸	d	30.27	46.71	93.8	93.8		
Radial elements 3D \star	✓ Leadir	ng edge Trailing edge	αF	90	26.4	(auto)	(auto)		
	Hub 1	1	βF	22	37.3	(auto)	(auto)		
	Shroud 1	1	u	126.8	195.7	(auto)	(auto)		
	Thickness mode	Tangential *	cm	51.2	58.9	(auto)	(auto)		
See help page for design rule radial element blades	s for Inickness mode		cu	0	118.4	(auto)	(auto)		
- Idulal element biddes			cr	2.5	3	(auto)	(auto)		
			cax	51.1	58.8	(auto)	(auto)		
			c	51.2	132.2	(auto)	(auto)		
			wu	-126.8	-77.3	(auto)	(auto)		
<b>β1: Incidence</b> i = βE	a second a s	<b>β2: Slip</b> δ = βB - βF		136.8	97.1	(auto)	(auto)		
eviation from shockless inflow	Deviation from	blade-congruent flow	τ	1.056	1.03	(auto)	(auto)		
efinition Angle relative *	Slip model	User defined V	i δ	14.7	8	(auto)	(auto)		
Rel = Ratio incidence i/ blade angl	евв	Angular deviation Velocity ratio			.71				
	Angular deviat				583	-			
iRelHub 40 %	5	δHub 8.0 °			53. <del>6</del>	-			
	δ direct	δ direct		15.3		12			
	8	δShr 8.0 °			8.7				
			γ	0.	903				
			∆(cu•r)	2.	765				
QMax/Q 125 % i(QMax)	= 9 *				.34				
			∆pt	17	7.08				

Blade properties

X

Picture 56 Space for shape and wing design settings and corner deviations CFturbo program

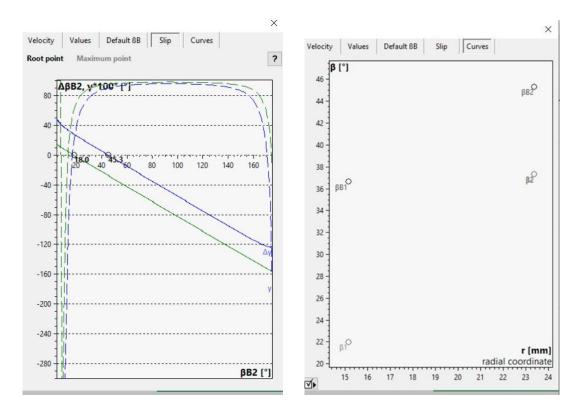
#### These pictures present the real velocities, the appeal and the corrected fin angles



Picture 57 Velocity triangles at the inlet and outlet of the inductor with the deviation angles

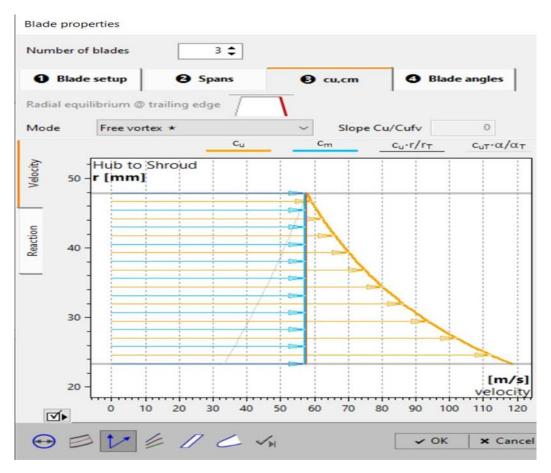
Picture 58 The corrected inlet and outlet vane angle values

This is where slippage occurs. The curves that show how the angle changes are not displayed because of the Radial elements 3D command.



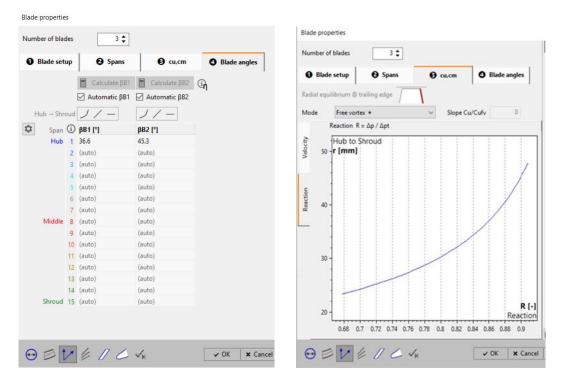
Picture 59 The slip of flow on blade CFturbo program

Picture 60 Angle distribution CFturbo program



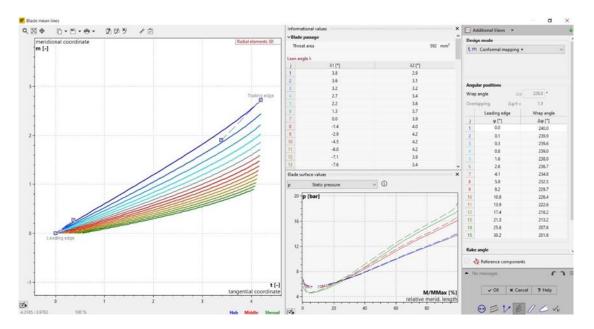
Picture 61 The radial distribution of Cu velocity with the free vorticity method CFturbo program

The methods which were used for blade design hub to shroud is free vortex. This method is considered the most suitable for the design of the inducer. Below, the blades angle aren't present in the picture from Radial elements 3D. The ratio R from the hub to the top of the wing shroud.



Picture 62 Automatic angle adjustment CFturbo program

The curvature of the wing lines and the wrap angle  $\Delta \Phi = 240^{\circ}$  were adjusted with static pressure as the main criterion. It is noted that in the indicative diagrams for Cm and Cu velocities, there are large exclusions.



Picture 68 Blade configuration and design space with some criteria CFturbo program

Based on the solidity of the wing and the angles  $\beta B1$  and  $\beta B2$  at the middle distance of the wing, the recall  $\delta$  was calculated. The relationship with which recall is calculated, is :

$$\delta = \left(2 + \frac{\beta B 2 - \beta B 2}{3}\right) \left(\frac{t}{l}\right)^{1/3}$$

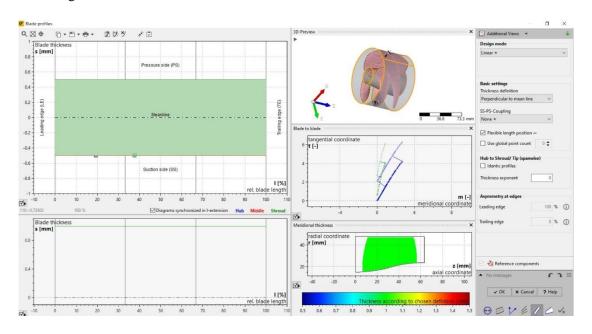
First, the  $\delta$  was 5° but it was corrected because for 1/t = 2.05,  $\beta B2 = 30.4^{\circ}$  and  $\beta B1 = 17.4^{\circ}$  the  $\delta = 4.99^{\circ}$ . Because an incorrect estimate of  $\delta$  was made, the corrected value was kept at 8 degrees, although normally should have been 5 degrees it at so that the whole design does change. not The estimate was made at the average distance of the wing (at point 8) for the hub and for the Shroud.

j	βB1 [°]	βB2 [*]	γ[*]	^
1	36.6	45.3	32.0	
2	32.1	43.3	29.5	
3	28.9	41.4	27.3	
4	26.0	39.6	25.4	
5	23.2	37.8	23.7	
6	20.7	36.1	22.3	
7	18.7	34.4	21.0	
8	17.1	32.8	19.8	
9	15.8	31.3	18.8	
10	14.6	29.8	17.8	
11	13.7	28.4	17.0	
12	12.8	27.0	16.2	
13	12.1	25.7	15.6	
14	11.5	24.5	14.9	
15	11.0	23.3	14.4	
15				
1	004 (2)		003 (8)	¥

j	l/t	IML/t	Δz/t	Δr/t
1	2.33	2.31	1.25	0.20
2	2.27	2.25	1.13	0.17
3	2.23	2.21	1.03	0.14
4	2.19	2.17	0.95	0.12
5	2.15	2.13	0.87	0.10
6	2.12	2.10	0.81	0.08
7	2.08	2.07	0.75	0.07
8	2.05	2.04	0.70	0.06
9	2.01	2.00	0.65	0.05
10	1.97	1.97	0.61	0.04
11	1.93	1.93	0.57	0.03
12	1.89	1.89	0.53	0.02
13	1.84	1.84	0.49	0.01
14	1.79	1.79	0.46	0.01
15	1.73	1.74	0.43	0.00

### Picture 69 The entry and exit angles CFturbo program

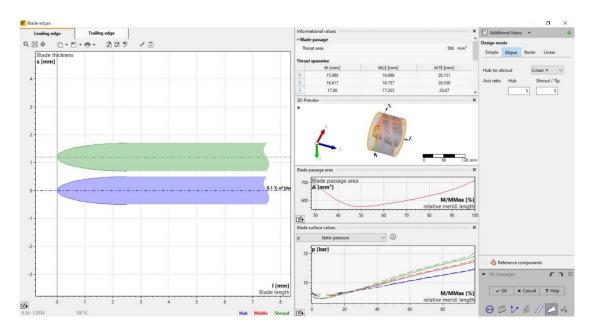
Picture 70 Solidity of blade CFturbo program



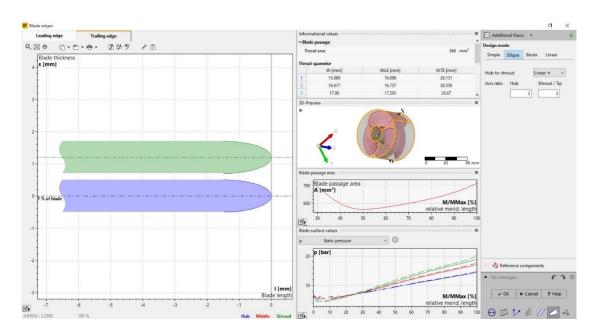
# No changes were made here

Picture 71 Blade profiles CFturbo program

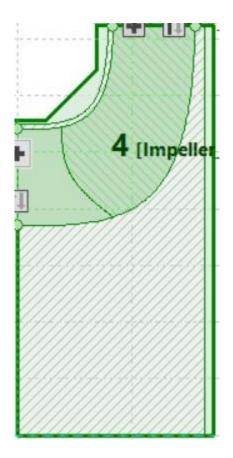
Based on the pressure the elliptical nose adjusts to the ratio of 5 at the leading edge of the blade



Picture 72 Blade leading edge CFturbo program



Picture 73 Blade trailing edge CFturbo program



Picture74 Impeller CFturbo program

The impeller design was based on the head coefficient  $\psi$ , the ds/dh ratio, the b2/d2 ratio and the base diameter dH. The values were adjusted appropriately after several design iterations to have a retardation of the velocity W at each point of the blade

Seneral	<ol> <li>Setup</li> </ol>	Parameters	Dimensions	Values	Meridian	Cordier	Veloci	ty
Manual dimensioning (i)       Volume flow       Q       977       m³         Manual dimensioning (i)       Mass flow       m       20       kgg         Splitter blades       Mass flow       m       20       kgg         Material density ρ       7750 kg/m³       Mass flow       m       80000       /m         mpeller type       C Standard *        Pressure difference       Δpt       150       bar         flow swirl       Upstream swirl *        PQ       4069       kW         Addt'l. Hydraulic efficiency       nh+       100       %         Image options       90 %       Specific speed (EU)       ng       26.1         Specific speed (EU)       ng       26.1       Kgg       Kgg         Contra rotating       90 %       Kgg       Kgg       Kgg       Kgg	ieneral		I. I.	✓Design Point				
J Unshrouded Tip clearance xin 0.72 xOut 0.72 mm   Pressure difference Δpt 150 bar   Splitter blades n 80000 /m   aterial density p 7750 kg/m³ kg/m³   mpeller type Standard * ~   flow swirl Upstream swirl * ~   Multi stage options 90 %   rergy fraction 90 %   Contra rotating Apt		ioning (j)		Volume flow		Q	977	m³/
Splitter blades     Pressure difference     Δpt     150 bar       sterial density ρ     7750 kg/m³     Revolutions     n     80000     /m       peller type     Standard ★      Specific speed (EU)     nq     24.1       Power output     PQ     4069 kW       Addt'l. Hydraulic efficiency     nh+     100 %       Multi stage options     90 %       ergy fraction     90 %       Contra rotating     Apt     135 bar	Unshrouded	Tip clearance xir	0.72 xOut 0.72 m	Mass flow		m	20	kg/
Atterial density p 7750 kg/m³   peller type Standard ★   low swirl Upstream swirl ★   low swirl Upstream swirl ★   Multi stage options 90 %   ergy fraction 90 %   contra rotating Apt 135 bar	Solitter blades				e	Δpt	150	bar
Specific speed (EU)     nq     24.1       Specific speed (EU)     PQ     4069 kW       Addt'l. Hydraulic efficiency     nh+     100 %       Vimpeller     Specific speed (EU)     nq     26.1       Specific speed (EU)     nq     26.1		7750 kg/m <sup>3</sup>		Revolutions		n	80000	/m
Iow swirl     Upstream swirl *     Addt'l. Hydraulic efficiency     ηh+     100 %       Multi stage options     Multi stage options     Specific speed (EU)     nq     26.1       sssure difference     Δpt     135 bar     Image: Contra rotating     Image: Co	itenal density p	7750 kg/m		Specific speed (EU	J)	nq	24.1	
Impeller       Multi stage options       ergy fraction       90 %       essure difference       Δpt       135       bar	peller type	Standard *	~	Power output		PQ	4069	kW
Multi stage options     Specific speed (EU)     nq     26.1       ergy fraction     90 %	low swirl	Upstream swirl *	~	Addt'l. Hydraulic	efficiency	ηh+	100	%
Multi stage options       ergy fraction       90 %       essure difference       Δpt       135 bar				✓Impeller				
ergy fraction 90 % essure difference Δpt 135 bar Contra rotating	Multi stage o	ntions		Specific speed (EU	J)	nq	26.1	
Alternative speed n 80000 /min	essure difference							
	Alternative spee	ed i	80000 /min					
						<ul> <li>No messages</li> </ul>	-	-

Picture 75 Start of set up of impeller CFturbo program

<ol> <li>Setup</li> </ol>	Parameters	Dimensions	Values	Meridian	Cordier	Veloc	ity
	_		~Power				
		Automatic 🤟 Set def	Torque			T 500	N
Parameters			Required drivi	ing power	PC	4191	k٧
Used for suction di			Required pow	ver incl. motor losses	PF	R 5240	k\
Diameter ratio	~ d!	6/d2 0.75	Power loss		PI	L 528	k\
Used for impeller d	iameter d2		✓Stage efficient	ю			
Work coefficient *		ψ 1	Internal efficie	ency	η	87.6	%
The countration		¥	Stage efficien	cy	ηS	t 87.4	%
Used for outlet wid	th b2		Stage efficien	cy incl. motor	ηSt	* 69.9	%
Outlet width ratio	* ~ bi	2/d2 0.1					
Efficiencies Design relevant	Information only		4				
Hydraulic efficiency	у	ηh 92.4 %					
Volumetric efficien	cy	ην 97.7 %					
Tip clearance efficie	ency	ηt 100 %					
Addt'l. Hydraulic ef	fficiency	ηh+ 100 %	(i)				
Use η for main	dimensions						
							_

Picture 76 The design parameters CFturbo program

The coefficient  $\psi = 1$  and other parameters were applied in order to where is retardation of the velocity W at each point of the blade. Also, the hub diameter dH = 74mm was set, based on the deceleration.

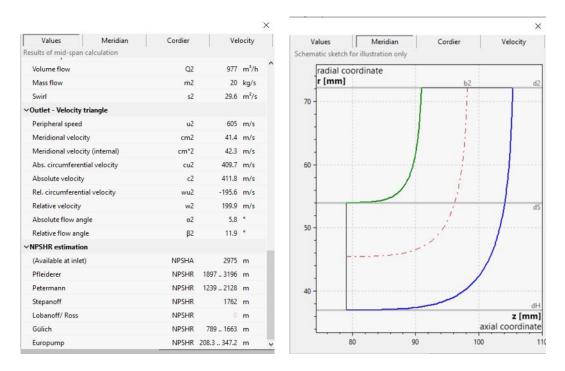
<li>Setup</li>	<b>O</b> Parameters	Dimensions		leridian Cordier	Ve	locity
Shaft	1		Results of mid-span calcula	tion		
Allowable stress	τ 15 1	ИРа	✓Global values			
			Work coefficient	ψ	1	
Factor of safety	SF 1.15		Specific diameter	δ	6.05	
Min. shaft diameter	d 58 r	nm	Total flow coefficient	φt	0.027	
			Meridional flow coefficie	nt ym	0.068	
Main dimensions			Meridional velocity ratio	cm2/cm1	0.741	
Hub diameter	dH 74 r	nm	Relative velocity ratio	w2/w1	0.779	
Calculate	Automatic		Inlet area	A1	4860	mm
Suction diameter	dS 108 r	nm	Outlet area	A2	6560	mm
			Area ratio	A2/A1	1.35	
Impeller diameter		nm βB2 = 23.4 *	Outlet width ratio	b2/d2	0.1	
Outlet width	b2 14.45 r	nm	Axial force	Fax	59000	Ν
-			✓Reynolds numbers			
Get 🗖 II	nlet 🔲 Outlet fr	om neighboring componen	Reynolds number (d1)	Re(d1)	2.132E8	
			Reynolds number (b1)	Re(b1)	3.356E7	
			Reynolds number (d2)	Re(d2)	3.817E8	
			Reynolds number (b2)	Re(b2)	3.817E7	
			✓Inlet - Flow properties			
			Density	p1	73.7	kg/n
			Static pressure	p1	14.56	bar
			Temperature	T1	-255	•

Picture 77 Input and output dimensions of the impeller and the all values CFturbo program

Values Meridian	Cordier	Vel	ocity							×
esults of mid-span calculation					Values	Meridian	Cordier	Vel	ocity	
Density	p1	73.7	kg/m³	^	Results of mid-span	calculation				
Static pressure	p1	14.56	bar		Density		p2		kg/m <sup>3</sup>	1
Temperature	T1	-255	°C		Static pressure		p2	94.5		
Total density	pt1	73.7	kg/m³		Temperature		T2	-255		
Total pressure	pt1	22	bar		Total density		pt2		kg/m <sup>3</sup>	
Total temperature	Tt1	-255	°C		Total pressure		pt2		bar	
Volume flow	Q1	977	m³/h		Total temperatur	e	Tt2	-255		
Mass flow	m1	20	kg/s		Volume flow		Q2		m³/h	
Swirl	s1		m²/s		Mass flow		m2		kg/s	
Inlet - Velocity triangle					Swirl	01000000	s2	29.6	m²/s	
Peripheral speed	u1	381.2	m/s		VOutlet - Velocity					
Meridional velocity	cm1	55.8	m/s		Peripheral speed		u2		m/s	
Meridional velocity (internal)	cm*1		m/s		Meridional veloc		cm2		m/s	
Abs. circumferential velocity	cu1	130.7			Meridional veloc		cm*2	1.10	m/s	
Absolute velocity	c1	142.1			Abs. circumferen	10000000000000000000000000000000000000	cu2	409.7		
Rel. circumferential velocity	wu1	-250.5			Absolute velocity		c2	411.8		
Relative velocity	w1	256.7	114 -		Rel. circumferent		wu2	-195.6		
Absolute flow angle	α1	23.1			Relative velocity		w2	199.9		
Relative flow angle	β1	12.6			Absolute flow an	-	α2	5.8		
3	pi	12.0			Relative flow ang		β2	11.9	-	f
Outlet - Flow properties					✓NPSHR estimation					
Density	ρ2	73.7	kg/m <sup>3</sup>		(Available at inle	t)	NPSHA	2975	m	

# Below are the impeller inlet and outlet fluid characteristics, velocities and blade image.

Picture 78 The values of all velocities, pressures and flow angles at the inlet and outlet of the impeller CFturbo program Picture 79 The values of all velocities, pressures and flow angles at the inlet and outlet of the impeller CFturbo program

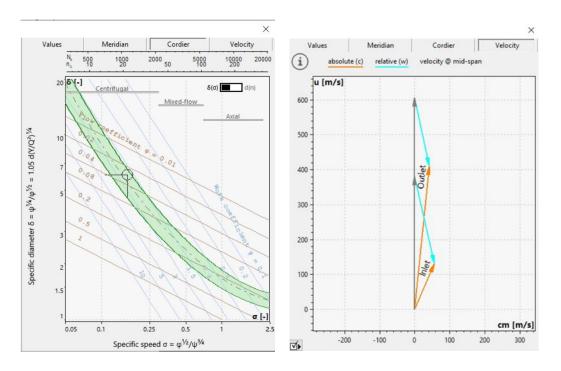


Picture 80 The values of all velocities, pressures and flow angles at the inlet and outlet of the impeller CFturbo program

Picture 81 Side view of blade CFturbo program

It is noted that :

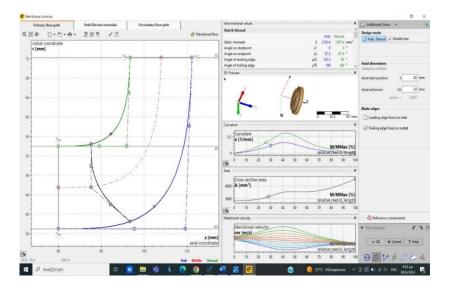
- The NPSHA height is slightly less than the worst-case NPSHR value. Therefore there is a very small possibility of cavitation
- Angles  $\beta 1$  and  $\beta 2$  are very close in value although deceleration is achieved
- Typical values of head coefficient in turbopumps are from 0.5 to 0.8.



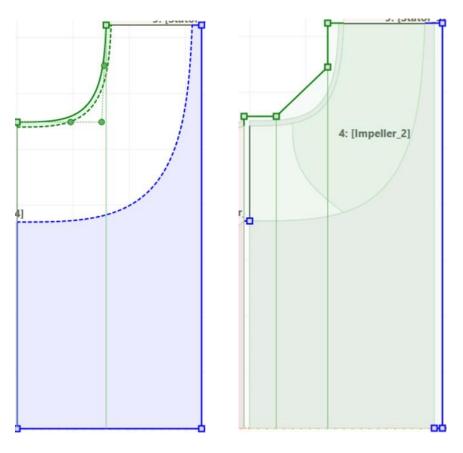
Picture 82 Inducer design point CFturbo program

Picture 83 Impeller input and output speed triangles CFturbo program

Below is the meridional contour. Minimal changes have been made with B-spline curves. shaped the aileron height to the default with  $\Delta Z = 24$ mm and the leading edge of the wing with B-spline curves. The last diagramma shows the axial velocity distribution over blade.



Picture 84 Meridian contour and B-spline configuration CFturbo program



Picture 85 The material of the impeller shaft, disc and shroud CFturbo program

Picture 86 The impeller with an indicative static cover CFturbo program

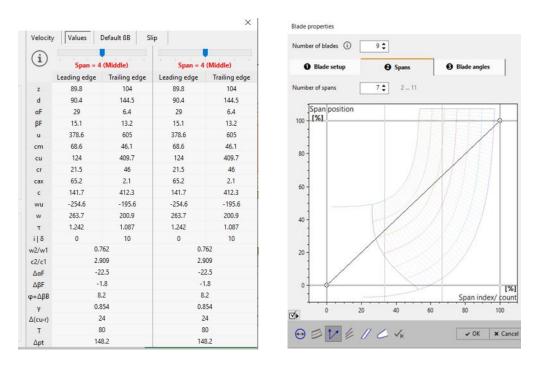
With the methods Free-form 3D the shape of blade was made. The appeal is default and to control the shape of the vane, it was divided into 7 points. The number of vanes was determined by the relationship below :

$$Z = K_z \frac{d_2 + d_1}{d_2 - d_1} \sin(\frac{\beta_1 + \beta_2}{2})$$

For  $K_z = 6,5$  and the data from set up results in Z = 9,5. Therefore, 9 vanes are chosen. In water pumps, the ratio T is from 1.1 to 1.25. Based on this criterion the thickness of the blade is set. The thickness was set to 1.6 mm so that the T ratio is close to the value of 1.25[55] at the front of the impeller. The deviation angle was manually selected to the default of 10 degrees. Of course in the end because the fillet became very large, this ratio at the base of the wing exceeds the standard limit.

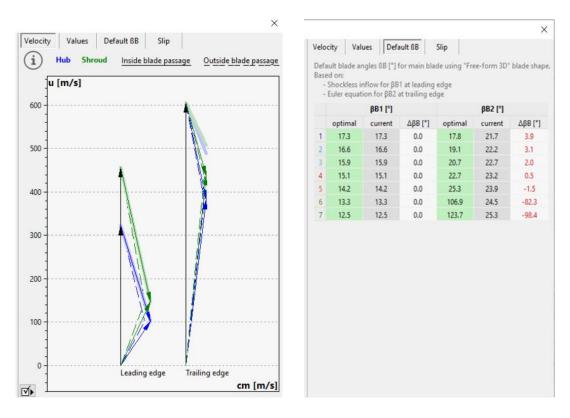
umber of blades (i)	9 🛊				Velocity	Values	Default BB SI	ip			
Blade setup	😧 Spi	ans 🛛 🕑	Blade	angles	(i)	Span =	1 (Hub)	Span = 7	(Shroud)		
I		1				Leading edge	Trailing edge	Leading edge	Trailing edg		
Blade shape		Blade thickness		s [mm]	z	96.7	111.2	87.7	96.8		
Free-form 3D" will provide	best results	To consider blade	blocka	ge 🌣 🗸	d	77.1	144.5	108.9	144.5		
🖉 Free-form 3D \star	~	Leading	g edge	Trailing edge	αF	34.2	6.9	25.1	6		
		Hub 1.6		1.6	βF	17.3	11.7	12.5	15.3		
		Shroud 1.6		1.6	u	322.8	605	456.2	605		
		Thickness mode	Tenner	tial *	cm	69	46.3	68.6	45.7		
		Inickness mode	langer		cu	101.6	381.6	146.5	437.8		
					cr	22.2	46.3	14.5	45.7		
					cax	65.4	2.1	67.1	2.1		
					c	122.8	384.4	161.8	440.2		
					wu	-221.2	-223.7	-309.8	-167.5		
1: Incidence	i = βB - βF	β2: Slip		δ = βB - βF	w	231.8	228.4	317.3	173.6		
eviation from shockless in	flow	Deviation from b	lade-co	ngruent flow	τ	1.249	1.094	1.242	1.08		
efinition Shockless flow	rate \star 🛛 🗸	Slip model	User d	efined $\checkmark$	i δ	0	10	0	10		
Q = Flow ratio shockless /	design	Angular deviation     Velocity ratio       δ direct     δHub			w2/w1	0.9	986	0.547			
					c2/c1	3.	3.13		2.722		
RQHub 100 %					ΔαΕ	-27.3 -5.6		-19.1 2.8			
					ΔβF						
RQShr 100 %		85	Shr	10.0 •	φ=ΔβΒ	4	.4	12.8			
					γ	0.8	323	0.8	83		
					∆(cu·r)	23	.66	23	.66		
					Т	39	.43	39	.43		
					∆pt	14	6.1	14	6.1		

Picture 87 Space for shape and wing design settings and corner deviations CFturbo program



Picture 88 The values at the inlet and outlet of the impeller at the central point of the blade CFturbo program

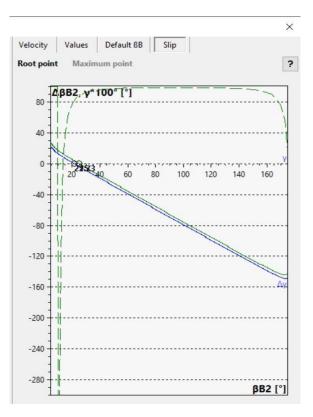
Picture 89 Number of spans CFturbo program



These photos show the speeds, the corrected blade angles and the slip.

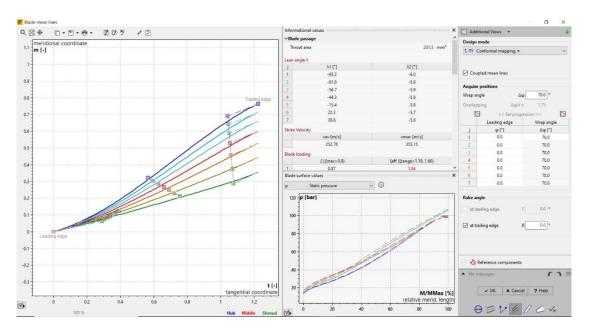
Picture 90 Velocity triangles at the inlet and outlet of the impeller with the deviation angles CFturbo program

Picture 91 The corrected inlet and outlet vane angle values CFturbo program

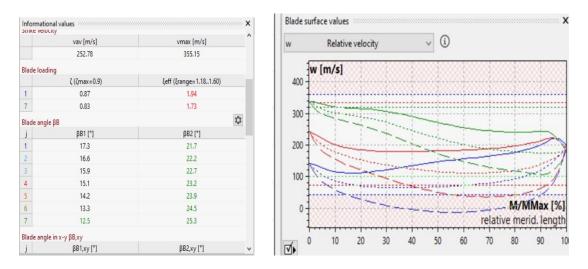


Picture 92 The slip CFturbo program

The fin was modeled based on the methods comfortable mapping. Appropriate meridional offsets were adjusted based on static pressure and by monitoring velocities. The scroll angle was left at the default  $\Delta \Phi = 70^{\circ}$ .

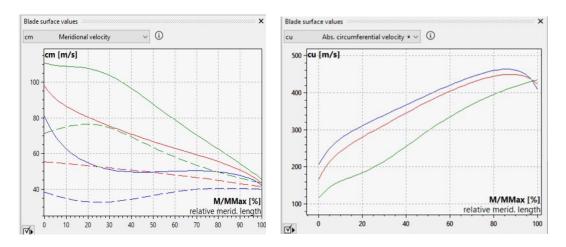


Picture 93 Blade configuration and design space with some criteria CFturbo program



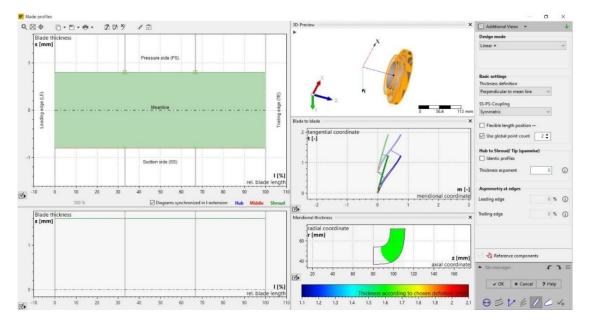
Picture 94 The entry and exit angles CFturbo program

Picture 95 The relative velocity distribution over the meridional length of the blade in percent CFturbo program



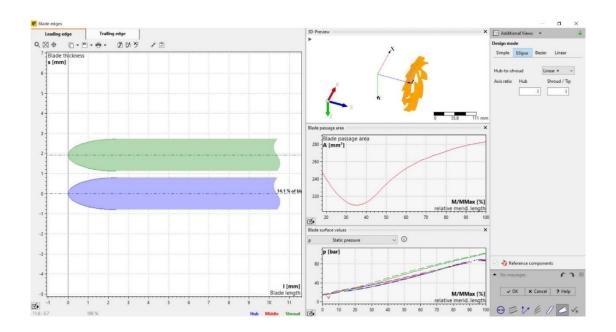
Picture 96 The meridional velocity distribution over the meridional length of the blade in percent CFturbo program Picture 97 The absolute circumferential velocity distribution over the meridional length of the blade in percent CFturbo program

There are big appeals in the meridional velocity in the intel of impeller.

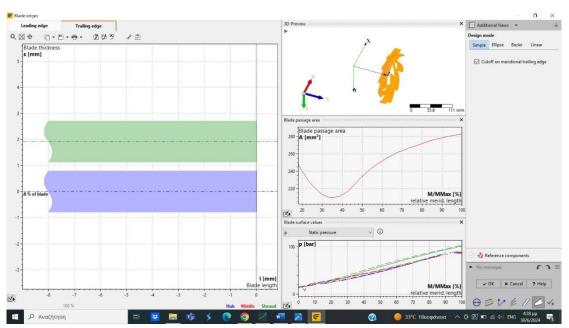


Picture 98 Blade profiles CFturbo program

Typically, the front tips of the impeller in turbopumps are elliptical with a ratio of 2 to 3[36]. Therefore, the default value of 3 was chosen based on the pressure.



Picture 99 Blade leading edge CFturbo program

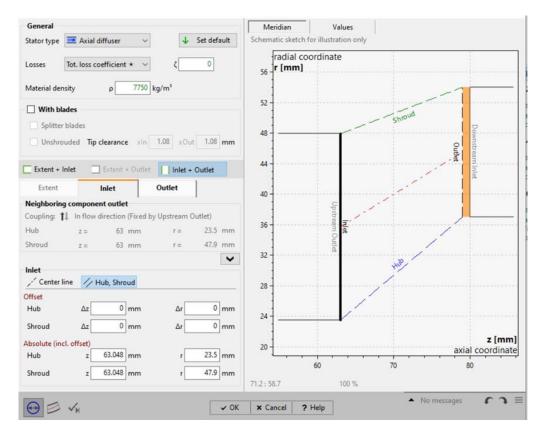


No changes were made to the rear of the impeller.

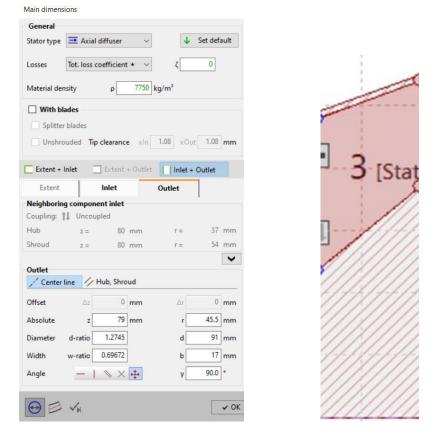
Picture 100 Blade trailing edge CFturbo program

## 4.1.3 Stator

Because the diameter of the commutator base and that of the impeller differ, a conical shaft was adapted to rotate inside a conical tube. This will bring additional hydraulic losses. Alternatively, fitting a diffuser would have been a better solution but was not used to simplify the CFD analysis. No design losses are taken.



Picture 101 Conical stator input set up CFturbo program

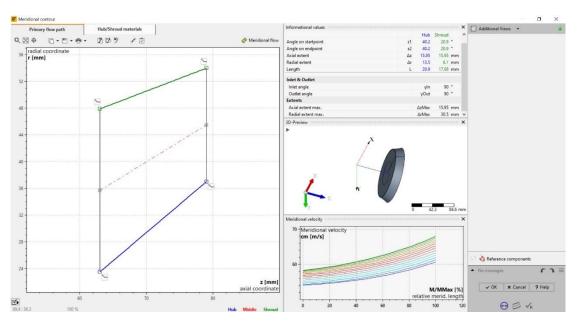


Picture 102 Conical stator output set up CFturbo program

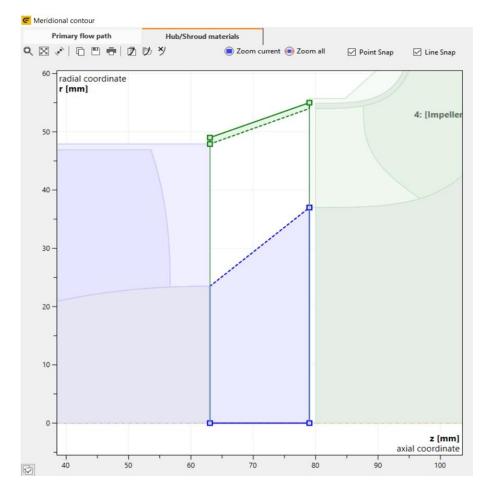
#### Picture 103 Stator CFturbo program

Meridian Values				Meridian Values			
/ Inlet				voutlet			
Average diameter	dm1	71.4	mm	Average diameter	dm2	01	mm
Width	b1	24.4	mm	Width	b2		mm
Area	A1	5470	mm²	Area	A2		mm
• Ratio to upstream outlet				~Ratio to downstream inlet	-	1000	
Diameter ratio	d-Ratio	1		Diameter ratio	d-Ratio	1	
Width ratio	b-Ratio	1		Width ratio	b-Ratio	1	
Area ratio	A-Ratio	1		Area ratio	A-Ratio	1	
Inlet - Flow properties				✓Outlet - Flow properties			
Meridional velocity	cm1	49.6	m/s	Meridional velocity	cm2	55.8	m/s
Abs. circumferential velocity	cu1	166.5	m/s	Abs. circumferential velocity	cu2	130.7	m/s
Absolute velocity	c1	173.8	m/s	Absolute velocity	c2	142.1	m/s
Absolute flow angle	α1	16.6	•	Absolute flow angle	α2	23.1	•
Density	p1	73.7	kg/m <sup>3</sup>	Density	p2	73.7	kg/m
Static pressure	p1	10.87	bar	Static pressure	p2	14.56	bar
Temperature	T1	-255	°C	Temperature	T2	-255	*с
Total density	pt1	73.7	kg/m <sup>3</sup>	Total density	pt2	73.7	kg/m
Total pressure	pt1	22	bar	Total pressure	pt2	22	bar
Total temperature	Tt1	-255	°C	Total temperature	Tt2	-255	*C
Volume flow	Q1	977	m³/h	Volume flow	Q2	977	m <sup>1</sup> /h
Mass flow	m1	20	kg/s	Mass flow	m2	20	kg/s
	s1	5.95	m²/s	Swirt	52	5.95	m²/s

Picture 104 The values of all velocities, pressures and flow angles at the inlet of the stator CFturbo program Picture 105 The values of all velocities, pressures and flow angles at the outlet of the stator CFturbo program It is observed that the input and output data are different from those of the output of the inducer. In practice, the pressure will change very little and the velocity will be what comes out of the inducer.

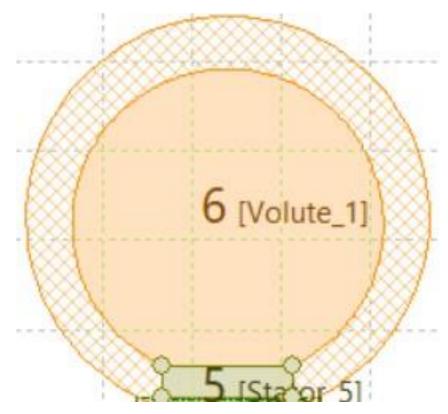


Picture 106 Meridian contour and B-spline configuration CFturbo program



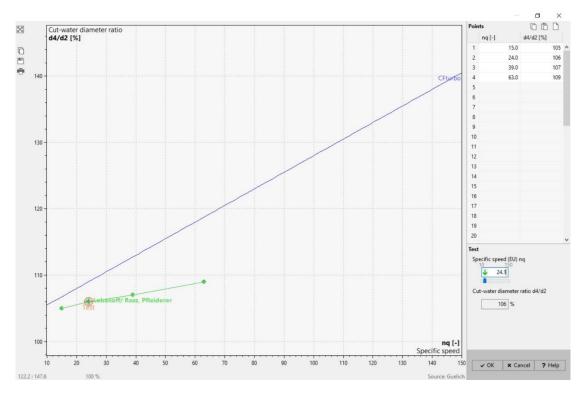
Picture 107 The material of the conic shaft and conic stator CFturbo program

### 4.1.4 Volute - Diffuser



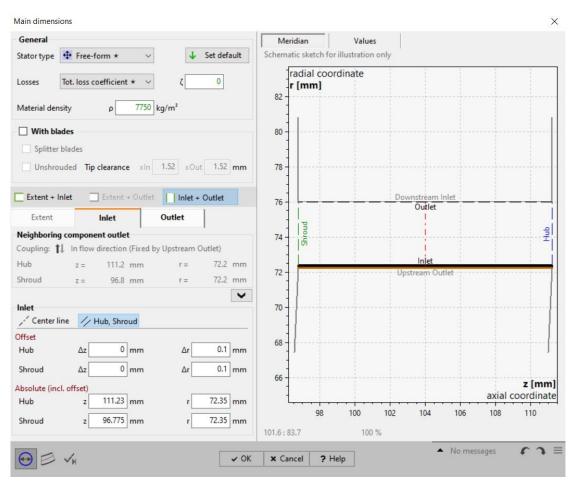
Picture 108 Volute - Diffuser

The distance from the volute - diffuser is usually very short and especially at high flows as in this case. In this case, the ration  $d_4/d_2$  was chosen 1,05 and combined with the following function with the special number of revolutions comes out 1,06. The difference is small. The distance includes the distance between the impeller and the shell which is 0,1mm.



Picture 109 Function of the ratio d4/d2 with the specific speed number

In this way the distance  $\Delta r = 3.65$ mm was set together with the distance of the wall to the impeller which is 0.1mm

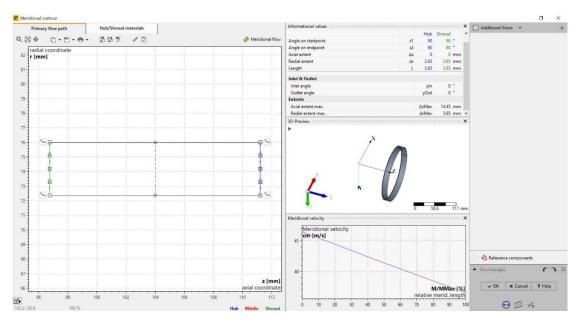


Picture 110 Stator input set up CFturbo program

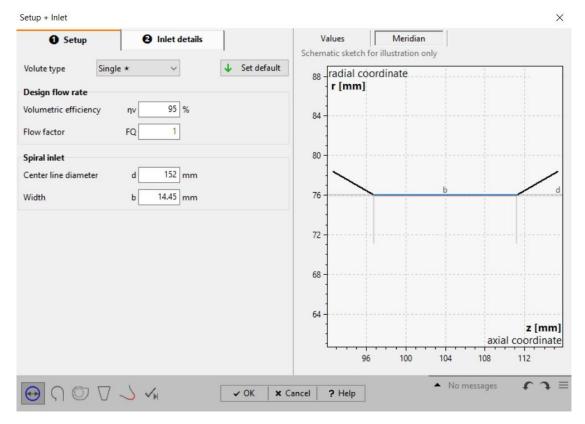
Meridian Values				Meridian Values			
				∽Outlet			
Average diameter	dm1	144.7	-	Average diameter	dm2	152	mm
Width	b1	14.45		Width	b2	14.45	mm
Area	A1		mm <sup>2</sup>	Area	A2	6900	mm²
Ratio to upstream outlet	0	0370		✓Ratio to downstream inlet			
Diameter ratio	d-Ratio	1.001		Diameter ratio	d-Ratio	1	
Width ratio	b-Ratio	1.001		Width ratio	b-Ratio	1	
Area ratio	A-Ratio	1.001		Area ratio	A-Ratio	1	
Inlet - Flow properties	,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,,			∨Outlet - Flow properties			
Meridional velocity	cm1	41.3	m/s	Meridional velocity	cm2	39.3	m/s
Abs. circumferential velocity	cu1	409.1		Abs. circumferential velocity	cu2	389.5	m/s
Absolute velocity	c1	411.2		Absolute velocity	c2	391.5	m/s
Absolute flow angle	α1	5.8	0.000	Absolute flow angle	α2	5.8	•
Density	p1	73.7	kg/m <sup>3</sup>	Density	ρ2	73.7	kg/n
Static pressure	p1	94.7		Static pressure	p2	100.5	bar
Temperature	TI	-255	°C	Temperature	T2	-255	°C
Total density	pt1	73.7	kg/m <sup>3</sup>	Total density	pt2	73.7	kg/n
Total pressure	pt1	157	bar	Total pressure	pt2	157	bar
Total temperature	Tt1	-255	*C	Total temperature	Tt2	-255	°C
Volume flow	Q1	977	m²/h	Volume flow	Q2	977	m³/ł
Mass flow	m1	20	kg/s	Mass flow	m2	20	kg/s
IVIDSS ITUW	s1	20.6	m²/s	Swirl	s2	29.6	2.

Picture 111 The values of all velocities, pressures and flow angles at the inlet of the stator CFturbo program

Picture 112 The values of all velocities, pressures and flow angles at the outlet of the stator CFturbo program

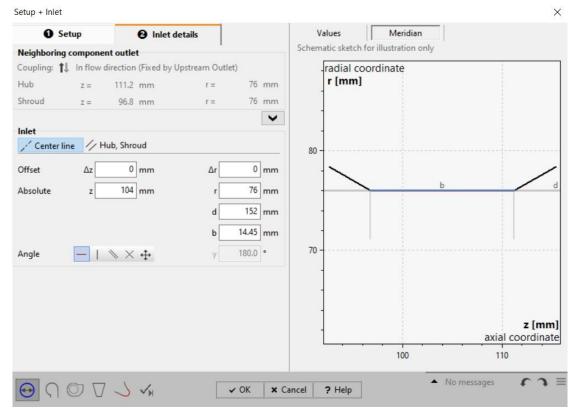


Picture 113 Meridian contour and B-spline configuration CFturbo program



Picture 114 Start of set up of volute - diffuser CFturbo program

In general, no particular changes were made to the diffuser. The thickness was formed a little higher with 3 mm and the method designed is with the constant vorticity (x=1) where with this method we have a high hydraulic efficiency. By estimation, the hydraulic grade manually entered 95%. the diameter and height were adjusted so that we have an outlet pressure of about 130 bar.

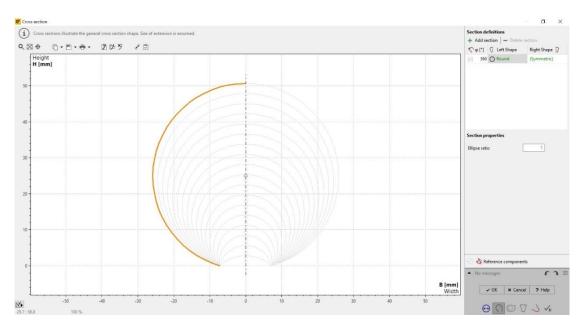


Picture 115 Entrance dimensions CFturbo program

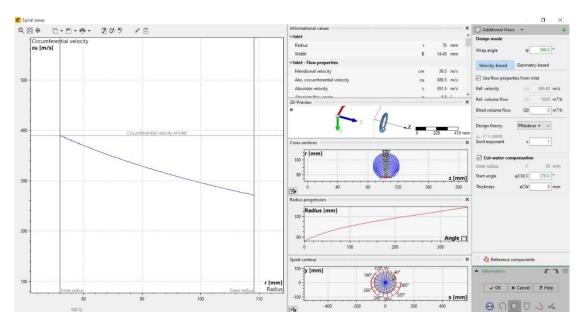
Values	Meridian			
Internal volume	flow	Qi	1028	m³/h
Ratios to previo	us component			
Spiral Inlet diam	eter ratio	d-Ratio	100	%
Spiral Inlet width	n ratio	b-Ratio	100	%
Inlet - Flow prop	oerties			
Meridional velo	ity	cm	39.3	m/s
Abs. circumfere	ntial velocity	cu	389.5	m/s
Absolute velocit	у	c	391.5	m/s
Absolute flow a	ngle	α	5.8	•
Density		ρ	73.7	kg/m <sup>3</sup>
Static pressure		р	100.5	bar
Temperature		т	-255	°C
Total density		ρt	73.7	kg/m <sup>3</sup>
Total pressure		pt	157	bar
Total temperatu	re	Tt	-255	°C
Volume flow		Q	977	m³/h
Mass flow		m	20	kg/s
Swirl		S	29.6	m²/s
		No messages	6	

Picture116 The values of all velocities, pressures and flow angles at the inlet of the diffuser CFturbo program

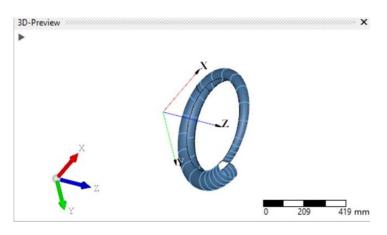
It is noted that the stator added before the volute - diffuser is considered part of the diffuser. For this reason, d4/d2 = 105% and not 100% is mentioned in the set up.



Picture 117 Configuration of spiral tube shape and size CFturbo program



Picture 118 Spiral design space based on some criteria CFturbo program

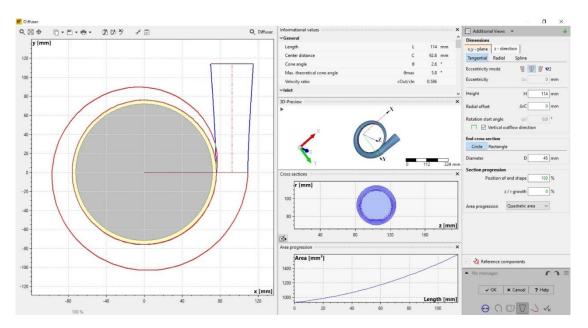


Picture 119 The spiral CFturbo program

Informational values				×	Informational values				>
∽Inlet				^	Height	н	33.17	mm	1
Radius	r	76	mm		Width	В	34.74	mm	
Width	В	14.45	mm		Side ratio	H/B	95.5	%	
Inlet - Flow properties					Area	А	933	mm²	
Meridional velocity	cm	39.3	m/s		Area/Radius	A/rc	10.13	mm	
Abs. circumferential velocity	cu	389.5	m/s		∨Last spiral section - Flow properties				
Absolute velocity	c	391.5	m/s		Meridional velocity	cm	0	m/s	
Absolute flow angle	α	5.8	•		Abs. circumferential velocity	cu	290.9	m/s	
Density	ρ	73.7	kg/m³		Absolute velocity	c	290.9	m/s	
Static pressure	p	100.5	bar		Absolute flow angle	α	90	•	
Temperature	т	-255	°C		Density	ρ	73.7	kg/m <sup>3</sup>	-
Total density	pt	73.7	kg/m <sup>3</sup>		Static pressure	р	113.4	bar	
Total pressure	pt	157	bar		Temperature	т	-255	°C	
Total temperature	Tt	-255	°C		Total density	pt	73.7	kg/m <sup>1</sup>	E.
Volume flow	Q	977	m³/h		Total pressure	pt	144.6	bar	
Mass flow	m	20	kg/s		Total temperature	Tt	-255	°C	
Swirl	5	29.6	m²/s		Volume flow	Q	977	m³/h	
/Last spiral section					Mass flow	m	20	kg/s	
Inner radius	r'	76	mm		Swirl	5	26.78	m²/s	
Outer radius	r	109.2	mm		∼Losses				
Equivalent diameter	D	34.46	mm		Sizing parameter	SP	1.105		
Min. axial coordinate	z	86.6	mm		Meridional loss coefficient	km	0.01		
Height	н	33.17	mm		Tangential loss coefficient	ku	0.074		
Width	В	34.74	mm		Wall loss coefficient	kw	0.135		
Side ratio	H/B	95.5	%		Overall loss coefficient	k	0.22		
Area	A	933	mm²	~	Total pressure loss	Δpt	12.4	bar	

Picture 120 The values of all velocities, pressures and flow angles at the inlet and the last spiral section CFturbo program

Picture 121 The values of all velocities, pressures and flow angles at the inlet and the last spiral section CFturbo program

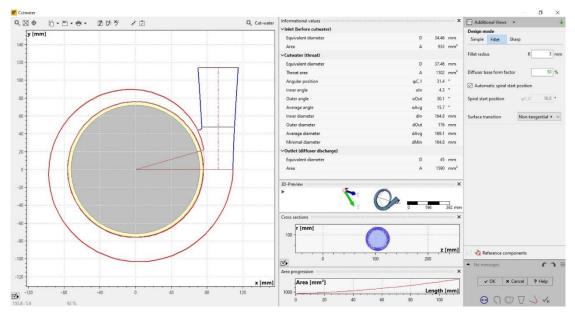


Picture 122 Configuration of conical diffuser at the outlet CFturbo program

Informational values				×	Informational values				>
∨General				^	Temperature	т	-255	*C	
Length	L	114	mm		Total density	pt	73.7	kg/m <sup>3</sup>	
Center distance	С	92.6	mm		Total pressure	pt	144.6	bar	
Cone angle	θ	2.6	•		Total temperature	Tt	-255	°C	
Max. theoretical cone angle	ϑmax	5.8	•		Volume flow	Q	977	m³/h	
Velocity ratio	cOut/cln	0.586			Mass flow	m	20	kg/s	
~ Inlet					Swirt	\$	26.78	m²/s	
Equivalent diameter	D	34.46	mm		∨Outlet - Flow properties				
Area	А	933	mm²		Absolute velocity	c	170.6	m/s	
- Outlet					Density	ρ	73.7	kg/m <sup>1</sup>	
Equivalent diameter	D	45	mm		Static pressure	P	130.1	bar	
Area	A	1590	mm²		Temperature	т	-255	*C	
Diffuser center position	Cx	92.6	mm		Total density	pt	73.7	kg/m <sup>3</sup>	
Diffuser center position	Су	114	mm		Total pressure	pt	140.8	bar	
Diffuser center position	Cz	104	mm		Total temperature	Tt	-255	*C	
Last spiral section - Flow properties					Volume flow	Q	977	m³/h	
Meridional velocity	cm	0	m/s		Mass flow	m	20	kg/s	
Abs. circumferential velocity	cu	290.9	m/s		~Losses				
Absolute velocity	c	290.9	m/s		Cone loss coefficient	kc	0.094		
Absolute flow angle	α	90	•		Wall loss coefficient	kw	0.028		
Density	ρ	73.7	kg/m <sup>3</sup>		Overall loss coefficient	k	0.122		
Static pressure	р	113.4	bar	~	Total pressure loss	Δpt	3.817	bar	

Picture 123 The values of all velocities, pressures and flow angles at the outlet of diffuser CFturbo program

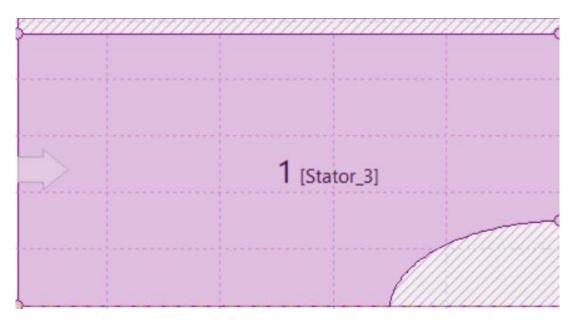
Picture 124 The values of all velocities, pressures and flow angles at the outlet of diffuser CFturbo program



## A curvature with a radius of 3mm was formed

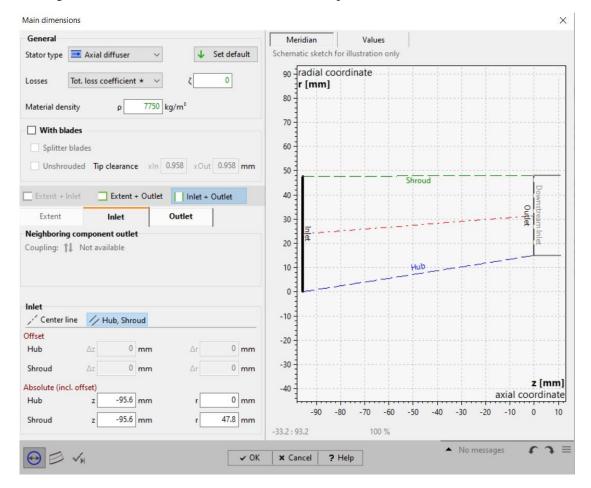
Picture 125 Curvature shaping space between conical diffuser and spiral CFturbo program

## 4.1.5 Rotary part of the inducer



Picture 126 Rotary inducer nose & inducer input space CFturbo program

A distance almost equal to the inlet diameter was set. A rotary nose was designed in front of the inducer. This tip was designed with B-spline curves and shaped to smooth entry velocity. Its height is the same as the diameter of the inducer input hub.

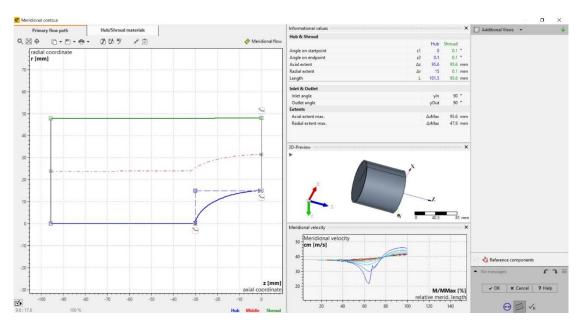


Picture 127 Set up the input area of the inducer CFturbo program

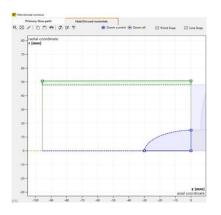
			$\times$				×
Meridian Values				Meridian Values			
Inlet				~Outlet			
Average diameter	dm1	47.8	mm	Average diameter	dm2	62.9	mm
Width	b1	47.8	mm	Width	b2		mm
Area	A1	7180	mm²	Area	A2	6500	
Inlet - Flow properties				~Ratio to downstream inlet			
Meridional velocity	cm1	37.8	m/s	Diameter ratio	d-Ratio	1	
Abs. circumferential velocity	cu1	0	m/s	Width ratio	b-Ratio	1	
Absolute velocity	c1	37.8	m/s	Area ratio	A-Ratio	1	
Absolute flow angle	α1	90		VOutlet - Flow properties	11000		
Density	p1	73.7	kg/m <sup>1</sup>	Meridional velocity	cm2	41.7	m/s
Static pressure	p1	6.47	bar	Abs. circumferential velocity	cu2		m/s
Temperature	TI	-255	*C	Absolute velocity	c2	41.7	
Total density	pt1	73.7	kg/m <sup>2</sup>	Absolute flow angle	a2	90	
Total pressure	pt1	7	bar	Density	p2	73.7	ka/r
Total temperature	Tt1	-255	°C	Static pressure	p2	6.36	1007
Volume flow	Q1	977	m³/h	Temperature	12	-255	
Mass flow	m1	20	kg/s	Total density	pt2	73.7	
Swirl	s1	0	m²/s	Total pressure	pt2		bar
				Total temperature	Tt2	-255	°C
				Volume flow	Q2	977	m²/
				Mass flow	m2	20	kq/s
				Swirl	s2	0	m²/
× Cancel ? Help	<ul> <li>No messages</li> </ul>	ſ	∋ ≡	X Cancel 2 Help	<ul> <li>No messages</li> </ul>	٢	٦

Picture 128 The values of all velocities, pressures and flow angles at the inlet CFturbo program

Picture 129 The values of all velocities, pressures and flow angles at the end of the rotary nose CFturbo program



Picture 130 Meridian contour and B-spline configuration CFturbo program



Picture 131 The material of the rotary nose CFturbo program

### 4.2 Oxidizer pump design

The usual oxidizer, liquid oxygen, was chosen. The theoretical combustion of oxygen with hydrogen is:

$$2H_2 + O_2 \rightarrow 2H_2O$$

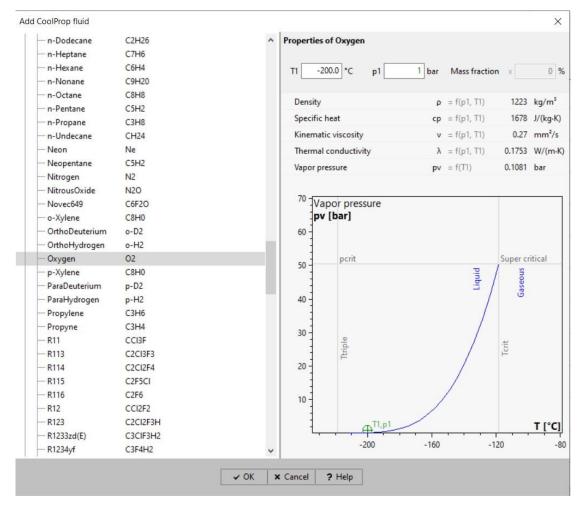
But in the combustion chamber of the rocket engine, the hydrogen and oxygen are in liquid form and the speeds are very high. This means that the fuel and oxidizer do not remain long enough for the chemical reaction to be fully completed. For this reason, the real reaction is this:

$$5H_2 + O_2 \rightarrow 2H_2O + 3H_2$$

It follows that the mass ratio of liquid oxygen and hydrogen is 3,2 [55]. Therefore, the supply of the oxygen pump will be:

$$\dot{m}_{02} = 3,2\dot{m}_{H2} = 64 Kg/s$$

However, this ratio would actually change a bit as a small amount of fuel and oxidizer would also be consumed if combined in an open cycle gas generation. This also depends on the power of the turbine. It was just used to make a simple dissection to provide the oxidizer to start the design. The following procedure is the same as for the fuel pump design. With the criterion that the specific speed should be in the range of 1000 to 2000 to have a high efficiency, the total pressure difference and the number of revolutions were adjusted. A variation was made on liquid oxygen as it is required to be in fully liquid form. The following images show the design point



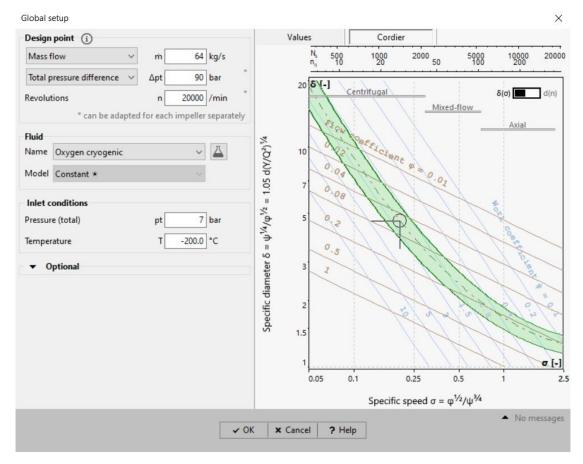
Picture 132 Formation space fluid material CFturbo program

Fluids				>
🗁 Open 🛛 🖭 Save	as D:	\tvagg\Fluid 11.cftfl		
Fluids	+ 🎮 M			
Name ^	Symbol	Description	Properties of Oxygen	cryogenic
<ul> <li>Incompressible</li> <li>G50-T20</li> <li>Glycol-H2O 50</li> <li>HC30</li> <li>Hydrogen</li> <li>Oxygen</li> <li>Oxygen cryoge</li> <li>Water</li> <li>Water (20°C)</li> </ul>	H2 O2 H2O	Dynalene HC30, props deri. props derived @ T = -255 *. props derived @ T = -150°C props derived @ T = 298.15	. Thermal conductivity	p 1223 kg/m <sup>3</sup> v 0.27 mm <sup>2</sup> /s λ 0.1753 W/(m·K) cp 1678 J/(kg·K) pv 0.1081 bar
		✓ OK X Cancel	? Help	

Picture 133 Properties of material fluid CFturbo program

Global setup				$\times$
Design point (j)	Values	Cordier		
Mass flow V m 64 k	g/s General machine typ	e: Centrifugal (medium pressure)		
Total pressure difference     Δpt     90       Revolutions     n     20000 /       * can be adapted for each impeller set	min 1		T	
Fluid Name Oxygen cryogenic ~		specific speed		
Model Constant *	Specific speed (EU)	nc	31.9	
	Specific work	١	7360	m²/s²
Inlet conditions	Power output	PC	t 471	kW
Pressure (total) pt 7 b	var Volume flow	c	188.4	m³/h
Temperature T -200.0 °	C Head	F	H 750	m
✓ Optional				
	✓ OK X Cancel ? H	Help	▲ No	o messages

Picture 134 The design point of the LO2 pump CFturbo program

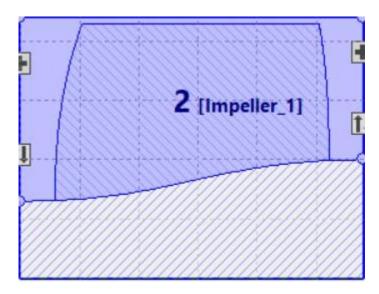


Picture 135 The design point of the  $LO_2$  pump CFturbo program

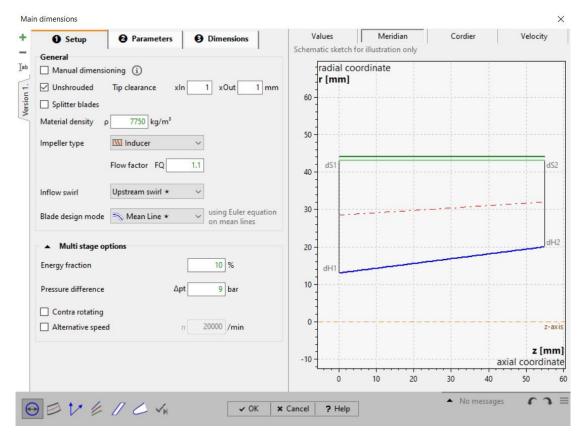
umber of stages (impeller) 2 🗘	Values Meridian		
	Required driving power	PD	553 kW
☐ [Impelier_1] →	Efficiency	n	85.1 %
- Impeller shape Axial	✓[Impeller_1]: Global values		
- Impeller type Inducer	Specific speed (EU)	ng	179.4
- Energy fraction 10 %	Design flow rate	Q*	218.1 m <sup>3</sup>
Pressure difference $\Delta pt$ 9 bar		ψ	0.139
[Impeller_2]	Specific diameter	δ	2.251
- Impeller shape Centrifugal	Total flow coefficient	φt	0.073
- Impeller type Standard	Meridional flow coefficient	φm	0.092
Energy fraction 90 %	Meridional velocity ratio	cm2/cm1	1.25
Pressure difference Δpt 81 ba	Relative velocity ratio	w2/w1	0.999
	Inlet area	A1	6920 m
	Outlet area	A2	5540 m
	Area ratio	A2/A1	0.8
	Axial force (thrust)	Fax	5610 N
	✓[Impeller_1]: Reynolds numbers		
	Reynolds number (d1)	Re(d1)	3.755E7
	Reynolds number (b1)	Re(b1)	1.314E7
	Reynolds number (d2)	Re(d2)	3.755E7
	Reynolds number (b2)	Re(b2)	8.99E6
	∨[Impeller_1]: Power		
	Torque	т	29.8 Nn
	Required driving power	PD	62.4 kV
	Required power incl. motor losses	PR	78 kV
	Power loss	PL	15.31 kV
	~[Impeller_1]: Stage efficiency		

Picture 136 Pressure differential rates of the inducer and the impeller CFturbo program

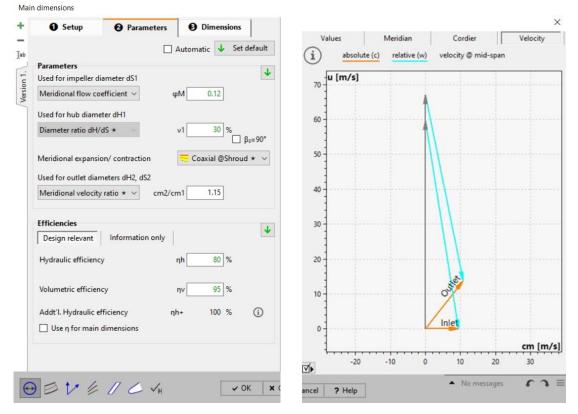
## 4.2.1 Inducer



Picture 137 Inducer CFturbo program



Picture 138 Start of set up of inducer CFturbo program



Picture 139 The design parameters CFturbo program

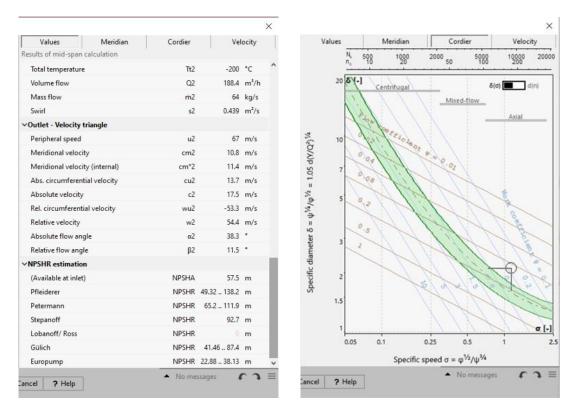
Picture 140 Inducer input and output speed triangles CFturbo program
--

<ol> <li>Setup</li> </ol>	Parameters	Dimensions	Values Meridian	Cordier	Ve	locity
Main dimensions			Results of mid-span calculation			
Calculate	Automatic		✓Global values			
Inlet			Work coefficient	ψ	0.173	
Hub diameter inl	let dH1 2	6 mm	Specific diameter	δ	2.014	
Hub diameter ini		5 mm	Total flow coefficient	φt	0.103	
Shroud diameter	inlet dS1 8	8 mm dTip = 86 mn		φm	0.118	
Outlet			Meridional velocity ratio	cm2/cm1	1.15	
Hub diameter ou	itlet dH2 4	0 mm	Relative velocity ratio	w2/w1	0.9	
			Inlet area	A1	5550	mm
Shroud diameter	outlet dS2 8	8 mm dTip = 86 mn	Outlet area	A2	4825	mm
			Area ratio	A2/A1	0.869	
			Axial force (thrust)	Fax	4670	Ν
			✓Reynolds numbers			
			Reynolds number (d1)	Re(d1)	3.004E7	
			Reynolds number (b1)	Re(b1)	1.058E7	
			Reynolds number (d2)	Re(d2)	3.004E7	
			Reynolds number (b2)	Re(b2)	8.19E6	
			∨Inlet - Flow properties			
			Density	ρ1	1223	kg/n
			Static pressure	p1	6.46	bar
			Temperature	T1	-200	°C
			Total density	pt1	1223	kg/n
			Total pressure	pt1	7	bar

Picture 141 Input and output dimensions of the inducer and the all values CFturbo program

Values	Meridian	Cordier	Vel	locity		Values Meridian	Cordier	Vel	locity
esults of mid-span					~	Results of mid-span calculation			
Inlet - Velocity tri	angle		50.7			✓Outlet - Flow properties			
Peripheral speed		u1		m/s		Density	ρ2		kg/m <sup>1</sup>
Meridional velocit		cm1		m/s		Static pressure	p2	14.13	
Meridional velocit		cm*1		m/s		Temperature	T2	-200	-
Abs. circumferent	ial velocity	cu1		m/s		Total density	pt2		kg/m <sup>1</sup>
Absolute velocity		c1		m/s		Total pressure	pt2		bar
Rel. circumferenti	al velocity	wu1	-59.7			Total temperature	Tt2	-200	°C
Relative velocity		w1		m/s		Volume flow	Q2	188.4	m³/h
Absolute flow ang	le	α1	90			Mass flow	m2	64	kg/s
Relative flow angl	e	β1	9	•		Swirl	s2	0.439	m²/s
Outlet - Flow prop	perties					✓Outlet - Velocity triangle			
Density		ρ2	1223	kg/m <sup>3</sup>		Peripheral speed	u2	67	m/s
Static pressure		p2	14.13	bar		Meridional velocity	cm2	10.8	m/s
Temperature		T2	-200	°C		Meridional velocity (internal)	cm*2	11.4	m/s
Total density		pt2	1223	kg/m <sup>3</sup>		Abs. circumferential velocity	cu2	13.7	m/s
Total pressure		pt2	16	bar		Absolute velocity	c2	17.5	m/s
Total temperature		Tt2	-200	°C		Rel. circumferential velocity	wu2	-53.3	m/s
Volume flow		Q2	188.4	m³/h		Relative velocity	w2	54.4	m/s
Mass flow		m2	64	kg/s		Absolute flow angle	α2	38.3	•
Swirl		s2	0.439	m²/s		Relative flow angle	β2	11.5	٠
Outlet - Velocity t	riangle					VNPSHR estimation			
Peripheral speed		u2	67	m/s	~	(Available at inlet)	NPSHA	57.5	m

Picture 143 The values of all velocities, pressures and flow angles at the inlet and outlet of the inducer CFturbo program Picture 144 The values of all velocities, pressures and flow angles at the inlet and outlet of the inducer CFturbo program



Picture 145 Input and output dimensions of the inducer and the all values CFturbo program

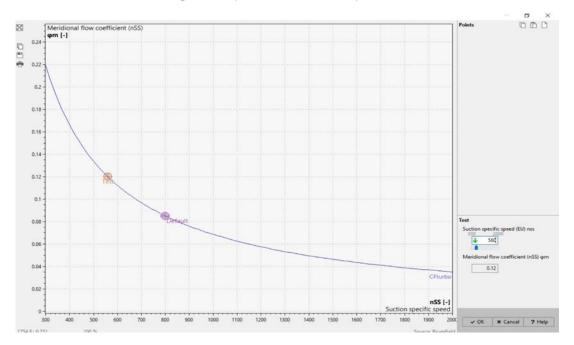
Picture 146 Inducer design point CFturbo program

Similar to the hydrogen pump, you also estimate the NSPHR height in this case

NPSHR = 
$$\lambda_c \frac{C_{m1}^2}{2g} + \lambda_w \frac{W_1^2}{2g}$$

- For  $C_{m1}$  = 9,4m/s ,  $\lambda_c$  = 1,1 ,  $~,~\lambda_w$  = 0,03 and  $W_1$  = 60,4m/s : NPSHR = 10,5m
- For  $C_{m1} = 9.4 \text{m/s}$ ,  $\lambda_c = 1.35$ ,  $\lambda_w = 0.06$  and  $W_1 = 60.4 \text{m/s}$ : NPSHR = 17 m

A range of NPSHR height was estimated for all cases. The available suction height is above these values. Therefore, the possibility of cavitation is very small.

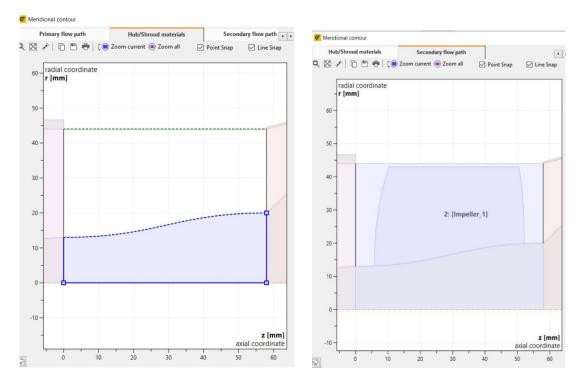


Picture 148 Flow coefficient function curve with specific speed number CFturbo program

C Meridional contour × ndary flow path 3D-Prev × 🔝 Additional Views 👻 Primary flow path Hub/Shroud materials Seco 4 Design mode 9.00 広めり \* 🖻 I Meridional flow 🚽 Middle lin radial coordinate 60 Axial din 113 mr Static × 0 mm Axial start p z 50 Static mo on ratio Az/Ar 2.1087 Axial ext a 1000  $\Delta z =$ 58 mm 40 M/MMax [%] Blade edges relative merid. length Leading edge fixed on inlet 30 40 20 50 60 70 80 ☑ Trailing edge fixed on outlet 30 × Gross action area 20 5000 M/MMax [%] relative merid. Hengitr 10 -80 20 ⊉ Meri al velocity Reference components 11 Meridional velocity cm [m/s] 0 . r n = 10 z [mm] ✓ OK X Cancel ? Help M/MMax [%] ordinate -10 relative merid. length **V**• 0 0 V 4 VH 50 10 20 30 40 60 100 110 120 130 Project 

From data and the function the NPSHR = 16,4m < NPSHA = 57,5 m

Picture 149 Meridian contour and B-spline configuration CFturbo program



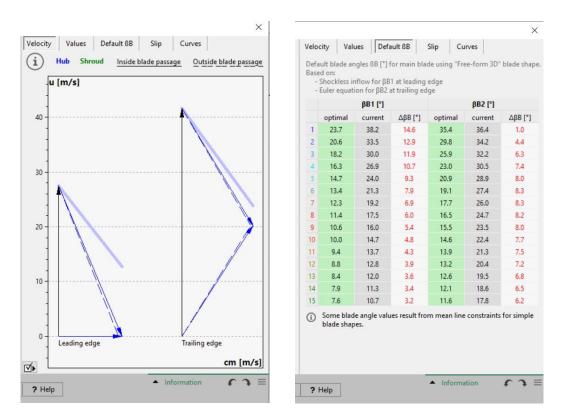
Picture 150 The shaft material (shaft shape) CFturbo program

Picture 151 Picture of inducer CFturbo program

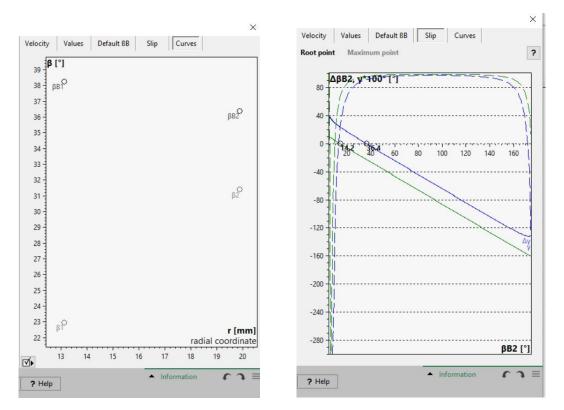
umber of blades 3 🗘			Velocity	Values	Default BB	Slip Curves			
Blade setup B Spans	Cu,cm	Blade angles	í	Span	= 1 (Hub)	Span = 15	i (Shroud)		
-				Leading edge	Trailing edge	Leading edge	Trailing ed		
Blade shape	Blade thickness	s [mm]	z	5.85	52.1	10.44	50.5		
Free-form 3D" will provide best results	To consider blade	blockage 🎝 🕹	d	26.24	39.74	86	86		
Radial elements 3D * 🛛 🗸	Leading	edge Trailing edge	αF	90	32.8	(auto)	(auto)		
	Hub 1	1	βF	22.9	31.4	(auto)	(auto)		
	Shroud 1	1	u	27.5	41.6	(auto)	(auto)		
	Thisbury and a	Tangential * V	cm	11.6	13	(auto)	(auto)		
See help page for design rules for radial element blades	Thickness mode	langential * V	cu	0	20.2	(auto)	(auto)		
- Tadiai element blades			cr	0.5	0.6	(auto)	(auto)		
			cax	11.6	13	(auto)	(auto)		
			c	11.6	24.1	(auto)	(auto)		
			wu	-27.5	-21.4	(auto)	(auto)		
<b>31: Incidence</b> $i = \beta B - \beta F$	β2: Slip	δ = βB - βF	w	29.8	25	(auto)	(auto)		
eviation from shockless inflow	Deviation from bl	ade-congruent flow	τ	1.062	1.042	(auto)	(auto)		
efinition Angle relative * ~	Slip model	User defined V	i δ	15.3	5	(auto)	(auto)		
tel = Ratio incidence i/ blade angle βΒ			w2/w1	(	0 <mark>.84</mark>	0			
	Angular deviatio	n Velocity ratio	c2/c1	2.07 -57.2 8.4			-		
iRelHub 40 %	δΗι	ub 5.0 °	ΔαΕ						
	δ direct		ΔβF						
	δS	φ=ΔβΒ	-1.9		>				
			Y	0	.912	-			
			∆(cu·r)	0	.402				
QMax/Q 125 % i(QMax) = 10.4 °			Т	(	0.59				
			∆pt		10.3	1.0	-		

The thickness in this case is 0.4 mm more than the default.

Picture 152 Space for shape and wing design settings and corner deviations CFturbo program

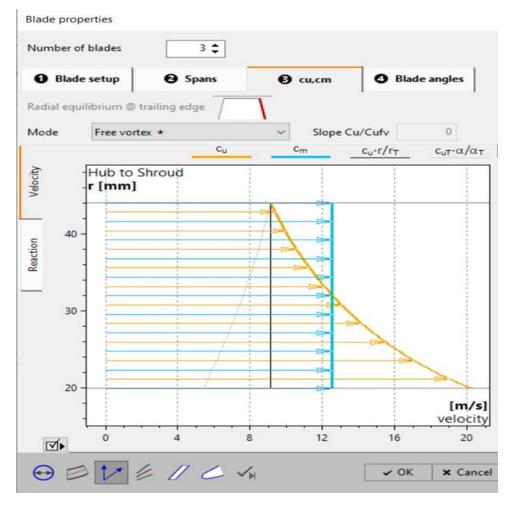


Picture 153 Velocity triangles at the inlet and outlet of the inductor with the deviation angles CFturbo program

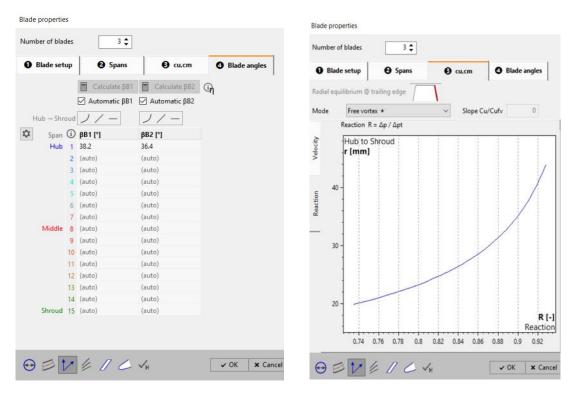


Picture 155 The slip of flow on blade CFturbo program

Picture 156 Angle distribution CFturbo program



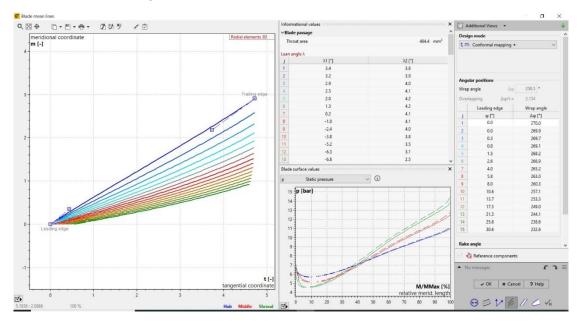
Picture 157 The radial distribution of Cu velocity with the free vorticity method CFturbo program



Picture 158 Automatic angle adjustment CFturbo program

Picture 159 The R ratio CFturbo program

The curvature of the wing lines and the wrap angle  $\Delta \Phi = 270^{\circ}$  were adjusted with static pressure as the main criterion. It is noted that in the indicative diagrams for Cm and Cu velocities, there are large exclusions.



Picture 160 Blade configuration and design space with some criteria CFturbo program

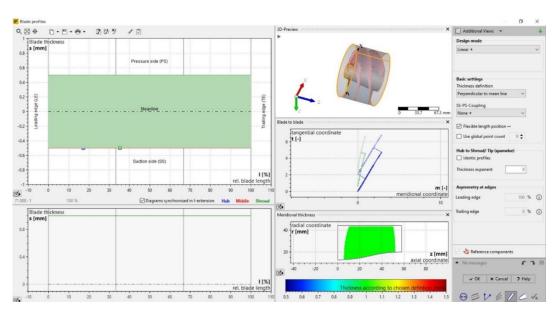
In this case, the deviation for 5 degrees results in angles  $\beta B1 = 17.5^{\circ}$  and  $\beta B2 = 24.7^{\circ}$ , for degree of solidity of 2.29,  $\delta = 3.33^{\circ}$ . The difference is small. The estimate was made at the average distance of the wing (at point 8) for the hub and for the Shroud.

j	βB1 [°]	βB2 [°]	γ [°]	^	Blade s	olidity				
1	38.2	36.4	30.8		j	l/t	IML/t	Δz/t	Δr/t	
2	33.5	34.2	28.1		1	2.59	2.58	1.34	0.20	
3	30.0	32.2	25.8		2	2.52	2.51	1.20	0.16	
4	26.9	30.5	23.8		3	2.47	2.46	1.09	0.13	
5	24.0	28.9	22.1		4	2.43	2.42	0.99	0.11	
2					5	2.40	2.38	0.91	0.09	
6	21.3	27.4	20.6		6	2.36	2.35	0.84	0.08	
7	19.2	26.0	19.3		7	2.33	2.32	0.77	0.06	
8	17.5	24.7	18.2		8	2.29	2.29	0.72	0.05	
9	16.0	23.5	17.2		9	2.26	2.25	0.67	0.04	
10	14.7	22.4	16.2		10	2.22	2.22	0.62	0.03	
11	13.7	21.3	15.4		11	2.18	2.18	0.58	0.02	
12	12.8	20.4	14.7		12	2.14	2.14	0.54	0.02	
13	12.0	19.5	14.0		13	2.09	2.09	0.51	0.01	
14	11.3	18.6	13.4		14	2.04	2.04	0.48	0.00	
15	10.7	17.8	12.9		15	1.99	1.99	0.44	0.00	

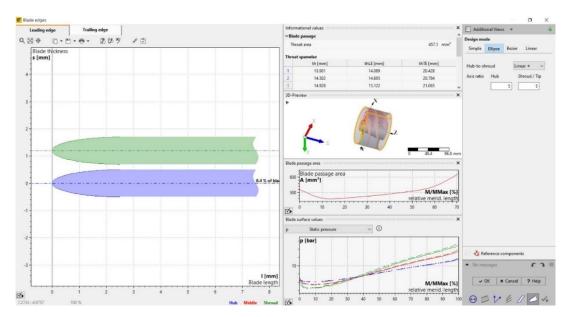
Picture 161 The entry and exit angles CFturbo program

Picture 162 Solidity of blade CFturbo program

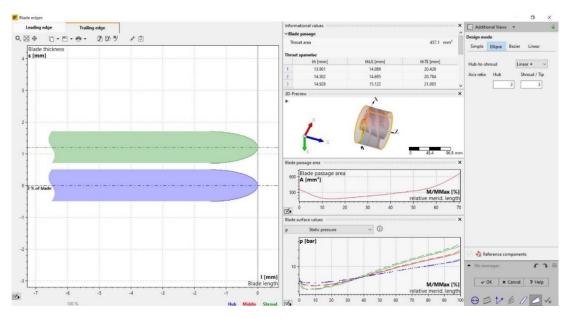
## Below are similar steps as in the previous inducer:



Picture 163 Blade profiles CFturbo program

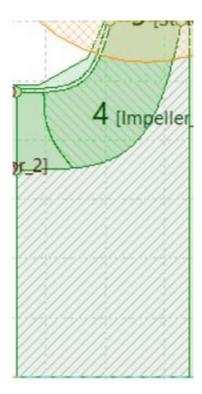


Picture 164 Blade leading edge CFturbo program



Picture 165 Blade trailing edge CFturbo program

4.2.2 Impeller



Picture 165 Impeller CFturbo program

Manual dimensioning ①       Q       188.4         Manual dimensioning ①       Mass flow       m       64         Displicter blades       Mass flow       m       64         Material density ρ       7750 kg/m³       Mass flow       m       20000         Impeller type       Standard *       Pressure difference       Δpt       90         Multi stage options       Power output       PQ       471         Addt'l. Hydraulic efficiency       nh+       100         Impeller type       Apt       81 bar       Specific speed (EU)       ng       34.5	General       Q       188.4       Manual dimensioning ①         Manual dimensioning ①       Mass flow       Mass flow       Mass flow         Dystream swirl p       7750 kg/m³       Mass flow       Mass flow       Mass flow         Material density p       7750 kg/m³       Mass flow       Mass flow       Mass flow       Mass flow         Impeller type       Standard *       Volume flow       Q       188.4       Mass flow         Inflow swirl       Upstream swirl *       Volume flow       Mass flow       m       64         Power output       Power output       PQ       471       Madd'1. Hydraulic efficiency       mh+       100       90         Multi stage options       90 %       Specific speed (EU)       mq       34.5       Mass flow       M       34.5         Pressure difference       Apt 81       bar       bar       Mass flow       m       20000       ////	<ol> <li>Setup</li> </ol>	Parameters	Dimensions	Values Meridian	Cordier	Veloci	ity
Manual dimensioning (1)       Mass flow       m       64         Unshrouded       Tip clearance       xln       0.6       xOut       0.6       mm         Splitter blades       Mass flow       m       64         Pressure difference       Δpt       90         Revolutions       n       20000         Specific speed (EU)       nq       31.9         Power output       PQ       471         Addt'l. Hydraulic efficiency       nh+       100         Impeller type       90 %       Specific speed (EU)       nq       34.5         Multi stage options       90 %       %       Specific speed (EU)       nq       34.5	Markaa dimensioning Mass flow m 64 k   Unshrouded Tip clearance x1n 0.6 xOut 0.6   Splitter blades Aaterial density ρ 7750 kg/m³ m 64 k   Mass flow m 64 k   Pressure difference Δpt 90 k   Specific speed (EU) nq 31.9   Power output PQ 471 k   Addt'l. Hydraulic efficiency nh+ 100 s   VImpeller Vimpeller Vimpeller   Specific speed (EU) nq 34.5	ieneral		1				
J Unshrouded       Tip clearance       xln       0.6       xOut       0.6       mm         Splitter blades       Attrial density       P       7750       kg/m³       ng       20000         Interial density       Tryp Crysto       Kg/m³       ng       31.9         Inpeller type       Standard *       V       Power output       PQ       471         Addt'l. Hydraulic efficiency       nh+       100         VImpeller       V       V       Specific speed (EU)       nq       34.5	J Unshrouded       Tip clearance       xln       0.6       xOut       0.6       mm         Splitter blades       Interial density p       7750       kg/m <sup>3</sup> Revolutions       n       20000 /         Interial density p       7750       kg/m <sup>3</sup> Pressure difference       Apt       90       k         Interial density p       7750       kg/m <sup>3</sup> Pressure difference       Apt       90       k         Iflow swirl       Upstream swirl *        Power output       PQ       471       k         Multi stage options       90       %       Mit istage options       nq       34.5         ressure difference       Apt       81       bar       Specific speed (EU)       nq       34.5         Contra rotating	] Manual dimens	ioning (i)			Q		
Splitter blades       n       2000         Material density ρ       7750 kg/m³       Revolutions       n       2000         specific speed (EU)       nq       31.9         Power output       PQ       471         Addt'l. Hydraulic efficiency       nh+       100         Impeller type       90 %       Specific speed (EU)       nq       34.5         Multi stage options       90 %       Specific speed (EU)       nq       34.5	Splitter blades       n       20000 /         faterial density ρ       7750 kg/m³       Specific speed (EU)       nq       31.9         mpeller type       Standard ★       >       Power output       PQ       471 k         flow swirl       Upstream swirl ★       >       Addt'l. Hydraulic efficiency       nh+       100 s         Multi stage options       90 %       specific speed (EU)       nq       34.5         ressure difference       Δpt       81 bar       Specific speed (EU)       nq       34.5	Unshrouded	Tip clearance xlr	0.6 xOut 0.6 m	n	ń	64	kg.
Material density ρ       7750 kg/m³       Revolutions       n       20000         Material density ρ       7750 kg/m³       Specific speed (EU)       nq       31.9         Power output       PQ       471         Addt'1. Hydraulic efficiency       nh+       100         Image options       90 %       Specific speed (EU)       nq       34.5         Specific speed (EU)       nq       34.5	Atterial density ρ       7750 kg/m³       Revolutions       n       2000 /         Material density ρ       7750 kg/m³       Specific speed (EU)       nq       31.9         Power output       PQ       471 k         Addt'1. Hydraulic efficiency       nh+       100 S         Multi stage options       90 %         Inergy fraction       90 %         Contra rotating       81 bar	Splitter blades				Δpt		
mpeller type       Image: Specific speed (EU)       ng       31.9         Multi stage options       Power output       PQ       471         Addt'I. Hydraulic efficiency       nh+       100         Image: Specific speed (EU)       ng       34.5         Specific speed (EU)       ng       34.5	mpeller type       Standard *          mflow swirl       Upstream swirl *       Power output       PQ       471         Multi stage options       Addt'l. Hydraulic efficiency       nh+       100       90         inergy fraction       90 %       Specific speed (EU)       nq       34.5         Contra rotating       Opt       81 bar       Versure difference       Apt       81 bar		7750 kg/m <sup>3</sup>			n		
Addt'l. Hydraulic efficiency     nh+     100       Multi stage options     -     -     Impeller       Specific speed (EU)     nq     34.5	Addt'l. Hydraulic efficiency     nh+     100 s       Multi stage options     ·     ·       inergy fraction     90 %       Pressure difference     Δpt     81 bar	notenti density p						
Multi stage options     Ympeller       Specific speed (EU)     nq     34.5       Pressure difference     Δpt     81	Multi stage options     γImpeller       • Multi stage options     Specific speed (EU)     nq     34.5       • Contra rotating     • Opt 000     000     000     000	mpeller type	✓ Standard ★	$\sim$				
Multi stage options     Specific speed (EU)     ng     34.5       nergy fraction     90 %     90 %     90 %     90 %       ressure difference     Δpt     81 bar     81 bar     81 bar	Multi stage options     Specific speed (EU)     ng     34.5       nergy fraction     90 %     90 %     90 %     90 %       ressure difference     Δpt     81 bar     90 %     90 %	nflow swirl	Upstream swirl *	~		ηh+	100	%
Multi stage options       inergy fraction       90 %       Pressure difference     Δpt 81 bar	▲ Multi stage options       inergy fraction       90 %       Pressure difference       Δpt       81 bar				∨Impeller			
Energy fraction     90 %       Pressure difference     Δpt       81     bar	Energy fraction     90 %       Pressure difference     Δpt 81 bar       Contra rotating	Multistage g	ntions		Specific speed (EU)	nq	34.5	
Contra rotating			. Др					
	Alternative speed n 20000 /min	Contra rotating						
Alternative speed n 20000 /min		Alternative spee	ed i	n 20000 /min				

Picture 166 Start of set up of impeller CFturbo program

The method is exactly the same as in the previous impeller.

● Setup ● Parame Parameters Used for suction diameter dS Diameter ratio  Used for impeller diameter d2 Work coefficient *  Used for outlet width b2 Outlet width ratio *  Efficiencies Design relevant Information of		I default	<ul> <li>✓ Power</li> <li>Torque</li> <li>Required driving power</li> <li>Required power incl. ma</li> <li>Power loss</li> <li>✓ Stage efficiency</li> <li>Internal efficiency</li> <li>Stage efficiency</li> <li>Stage efficiency incl. ma</li> </ul>	otor losses	T PD PR PL ηI ηSt	234.3 490.8 613 66.9 87 86.4 69.1
Used for suction diameter dS Diameter ratio Used for impeller diameter d2 Work coefficient * Used for outlet width b2 Outlet width ratio * Efficiencies	dS/d2 0.8 ψ 0.84	¥	Required driving power Required power incl. ma Power loss <b>Stage efficiency</b> Internal efficiency Stage efficiency	otor losses	PD PR PL ŋl ŋSt	490.8 613 66.9 87 86.4
Used for suction diameter dS Diameter ratio Used for impeller diameter d2 Work coefficient * Used for outlet width b2 Outlet width ratio * Efficiencies	ψ0.84	¥	Required power incl. me Power loss <b>Stage efficiency</b> Internal efficiency Stage efficiency	otor losses	PR PL ŋl ŋSt	613 66.9 87 86.4
Diameter ratio Used for impeller diameter d2 Work coefficient * Used for outlet width b2 Outlet width ratio * Efficiencies	ψ0.84		Power loss Stage efficiency Internal efficiency Stage efficiency		PL ŋl ŋSt	66.9 87 86.4
Used for impeller diameter d2 Work coefficient * v Used for outlet width b2 Outlet width ratio * v Efficiencies	ψ0.84		✓Stage efficiency Internal efficiency Stage efficiency	otor	ղl ղSt	87 86.4
Work coefficient * Used for outlet width b2 Outlet width ratio * Efficiencies	,		Internal efficiency Stage efficiency	otor	ηSt	86.4
Work coefficient * Used for outlet width b2 Outlet width ratio * Efficiencies	,		Stage efficiency	otor	ηSt	86.4
Used for outlet width b2 Outlet width ratio * ~ Efficiencies	,			otor	1000	
Outlet width ratio * ~	b2/d20.1		Stage efficiency incl. m	otor	ηSt*	69.1
Efficiencies	b2/d2 0.1					
Hydraulic efficiency Volumetric efficiency Tip clearance efficiency Addt'l. Hydraulic efficiency Use n for main dimensions	nh 91.6 % ην 96.4 % ηt 100 % η+ 100 %	•				

Picture167 The design parameters CFturbo program

<ol> <li>Setup</li> </ol>	Parameters	Dimensions	Values Meridiar	Cordier	Vel	ocity
Shaft		1	Results of mid-span calculation			
Allowable stress	τ 15 N	1Pa	✓Global values		0.000	
Factor of safety	SF 1.15		Work coefficient	ψ	0.839	
			Specific diameter	δ	4.988	
Min. shaft diameter	d 45.06 m	ากา	Total flow coefficient Meridional flow coefficient	φt	0.037	
Main dimensions				φm cm2/cm1	0.092	
Hub diameter dH 70 mm			Meridional velocity ratio	cm2/cm1 w2/w1	0.749	
			Relative velocity ratio	w2/w1	3390	
			Outlet area	A1 A2	4524	
uction diameter dS 96 mm		Area ratio	A2/A1	1.335	mr	
Impeller diameter d2 120 mm β82 = 24.4 * Outlet width b2 12 mm Get Inlet Outlet from neighboring component	Outlet width ratio	b2/d2	0.1			
	Axial force	Fax	24710	N		
		✓Reynolds numbers	FdX	24710	IN	
	om neighboring component	Reynolds number (d1)	Re(d1)	3.574E7		
			Reynolds number (b1)	Re(b1)	4.84E6	
			Reynolds number (d2)	Re(d2)	5.59E7	
	Reynolds number (b2)	Re(b2)	5.59E6			
	✓Inlet - Flow properties	(ic(ic))	5,5520			
			Density	<b>ρ</b> 1	1223	ka
			Static pressure	p1	11.46	-
			Temperature	T1	-200	
			lemperature		-200	C

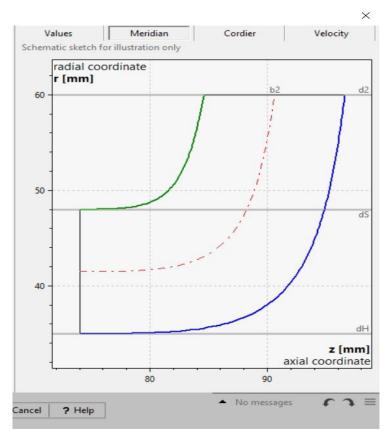
Picture 168 Input and output dimensions of the impeller and the all values CFturbo program

Values Meridian	Cordier	Vel	ocity		Values	Meridian	Cordier	Vel	locity
Results of mid-span calculation Total temperature	Tt1	-200	°C		Results of mid-spa	n calculation		in an	
Volume flow	Q1	188.4	m³/h		Volume flow		Q2	188.4	m²/ł
Mass flow	m1	64	kg/s		Mass flow		m2	64	kg/s
Swirl	s1	0.932	m²/s		Swirl		s2	4.384	m²/s
✓Inlet - Velocity triangle					VOutlet - Velocity	y triangle			
Peripheral speed	u1	86.9	m/s		Peripheral speed	9	u2	125.7	m/s
Meridional velocity	cm1	15.4	m/s		Meridional velo	city	cm2	11.6	m/s
Meridional velocity (internal)	cm*1	16	m/s		Meridional velo	city (internal)	cm*2	12	m/s
Abs. circumferential velocity	cu1	22.5	m/s		Abs. circumfere	ntial velocity	cu2	73.1	m/s
Absolute velocity	c1	27.3	m/s		Absolute velocit	ty	c2	74	m/s
Rel. circumferential velocity	wu1	-64.5	m/s		Rel. circumferer	ntial velocity	wu2	-52.6	m/s
Relative velocity	w1	66.3	m/s		Relative velocity	1	w2		m/s
Absolute flow angle	α1	34.5	•		Absolute flow a	ngle	α2	9	*
Relative flow angle	<b>β</b> 1	13.5	e		Relative flow an	gle	β2	12.4	•
Outlet - Flow properties					✓NPSHR estimation	on			
Density	ρ2	1223	kg/m <sup>3</sup>		(Available at inle	et)	NPSHA	132.5	m
Static pressure	p2	63.5	bar		Pfleiderer		NPSHR	83.5 154	m
Temperature	T2	-200	°C		Petermann		NPSHR	65.2 111.9	m
Total density	pt2	1223	kg/m <sup>3</sup>		Stepanoff		NPSHR	92.7	m
Total pressure	pt2	97	bar		Lobanoff/ Ross		NPSHR		m
Total temperature	Tt2	-200	°C		Gülich		NPSHR		
Volume flow	02	188.4	m <sup>3</sup> /h	~	Europump		NPSHR	22.88 38.13	m

Picture 169 The values of all velocities, pressures and flow angles at the inlet and outlet of the impeller CFturbo program Picture 170 The values of all velocities, pressures and flow angles at the inlet and outlet of the impeller CFturbo program

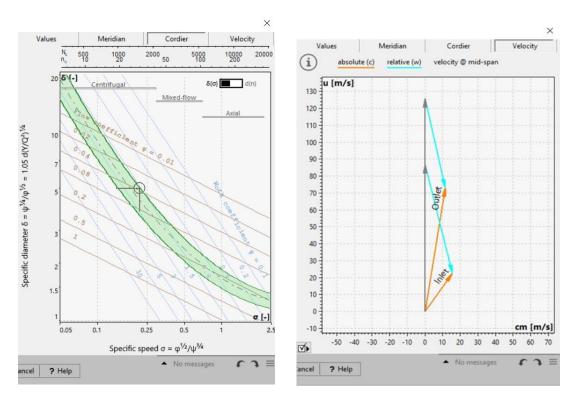
It is noted that :

• The NPSHA height is slightly less than the worst-case NPSHR value. Therefore, there is a very small possibility of cavitation.



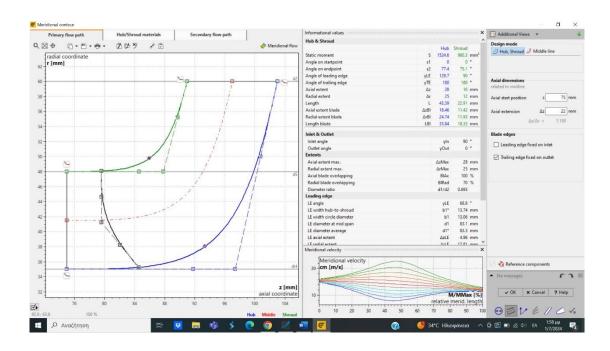
• Angles  $\beta 1$  and  $\beta 2$  are very close in value although deceleration is achieved.

Picture 171 Side view of blade CFturbo program

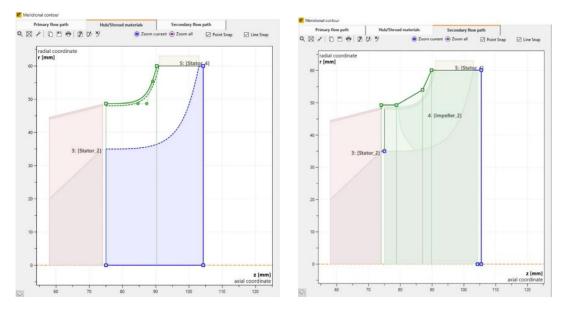


Picture 172 Inducer design point CFturbo program

#### Picture173 Impeller input and output speed triangles CFturbo program



Picture 174 Meridian contour and B-spline configuration CFturbo program



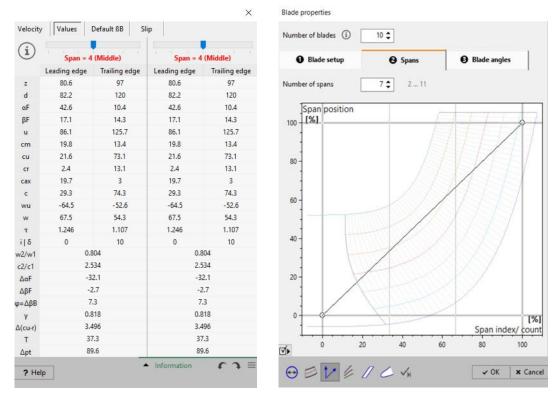
Picture 175 The material of the impeller shaft, disc and shroud CFturbo program

Picture 176 The impeller with an indicative static cover CFturbo program

Here the same methodology was applied with the previous one. The only difference is the coefficient k for estimating the number of fins. Because for  $K_z$ = 6.5, 13 fins come out, which caused a capacity issue. So, the coefficient became 5 and 10 fins were selected.

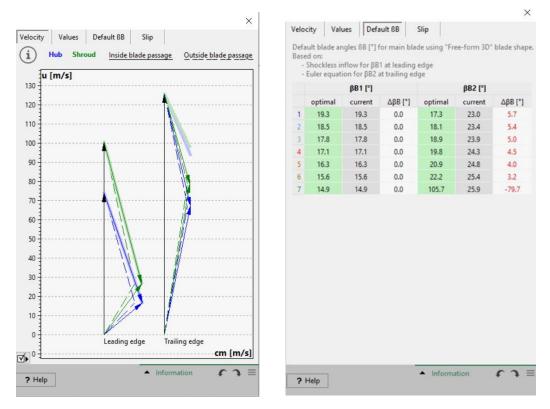
umber of blades (i) 10 🛊				Velocity	Values	Default BB SI	ip		
Blade setup	Spans	🕑 Blade	angles	(i)	Span =	1 (Hub)	Span = 7 (Shroud)		
					Leading edge	Trailing edge	Leading edge	Trailing edg	
Blade shape		Blade thickness     s [mm]       To consider blade blockage     Image: Consider blade blockage		z	84.5	103	79.6	91	
Free-form 3D" will provide best res	Its To cor			d	70.5	120	96.1	120	
Free-form 3D *	~	Leading edge	Trailing edge	αF	50.5	11.4	36.5	9.6	
	Hub	1.5	1.5	βF	19.3	13	14.9	15.9	
	Shrou	d 1.5	1.5	u	73.8	125.7	100.7	125.7	
	<b>T</b>	Thickness mode Tangential * ~			20	13.5	19.7	13.4	
	Thickr				16.5	67.2	26.7	78.9	
					2	13.2	1.1	12.9	
					19.9	3	19.7	3.4	
					26	68.6	33.2	80	
					-57.3	-58.4	-74	-46.8	
<b>β1: Incidence</b> i = βB	βF β2: SI	β2: Slip $δ = βB - βF$		w	60.7	60	76.6	48.6	
Deviation from shockless inflow	Deviat	ion from blade-co	ingruent flow	τ	1.258	1.113	1.239	1.1	
Definition Shockless flowrate *	✓ Slip m	Slip model User defined ~ Angular deviation Velocity ratio			0	10	0	10	
Q = Flow ratio shockless / design	_				0.9	988	0.635 2.413		
to = 110W totio shockiess / design	Angu				2.0	542			
RQHub 100 %					-39.1		-26.9		
	δ dir	δ direct		ΔβF	-6.2		1		
RQShr 100 %		δShr 10.0 °			3.8		11		
Nugani 100 %		0.0		Y	0.788		0.846		
				∆(cu-r)	3.4	152	3.4	52	
				т	18	.41	18.	41	
				∆pt	8	3.4	88	.4	

Picture 177 Space for shape and wing design settings and corner deviations CFturbo program



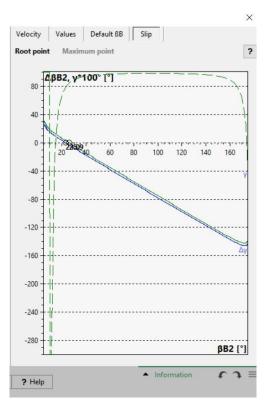
Picture 178 The values at the inlet and outlet of the impeller at the central point of the blade CFturbo program

#### Picture 179 Number of spans CFturbo program

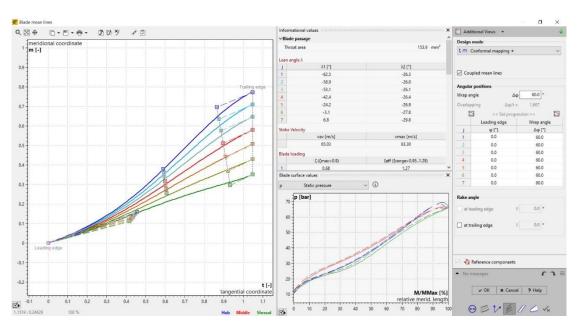


Picture 180 Velocity triangles at the inlet and outlet of the impeller with the deviation angles CFturbo program

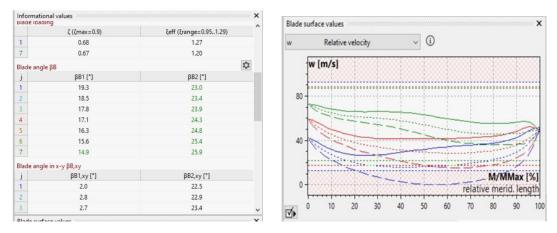
Picture 181 The corrected inlet and outlet vane angle values CFturbo program



Picture 182 The slip CFturbo program

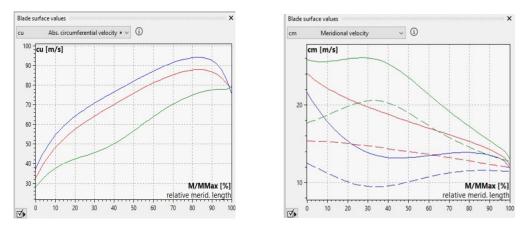


Picture 183 Blade configuration and design space with some criteria CFturbo program

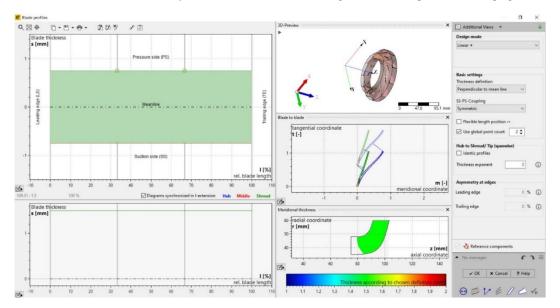


Picture184 The entry and exit angles CFturbo program

Picture 185 The relative velocity distribution over the meridional length of the blade in percent CFturbo program

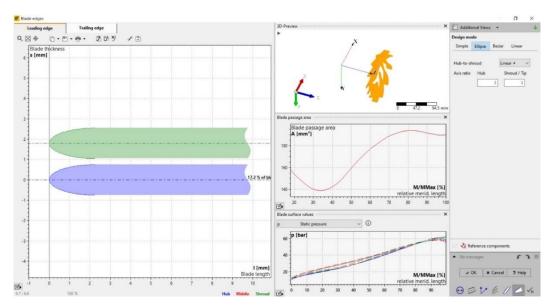


Picture 186 The absolute circumferential velocity distribution over the meridional length of the blade in percent CFturbo program

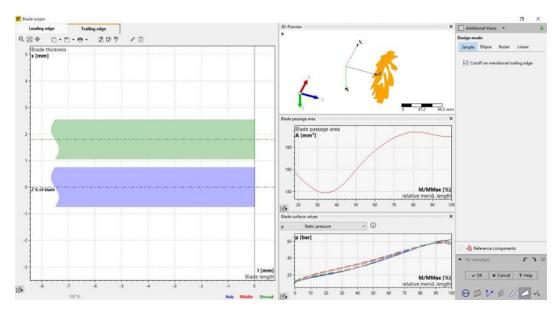


Picture 187 The meridional velocity distribution over the meridional length of the blade in percent CFturbo program

Picture 188 Blade profiles CFturbo program



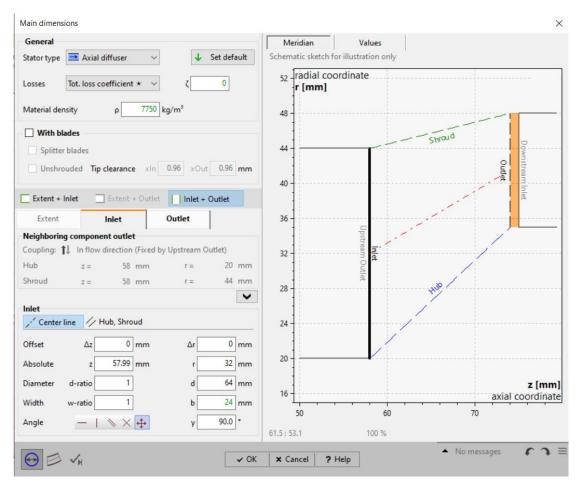
Picture 189 Blade leading edge CFturbo program



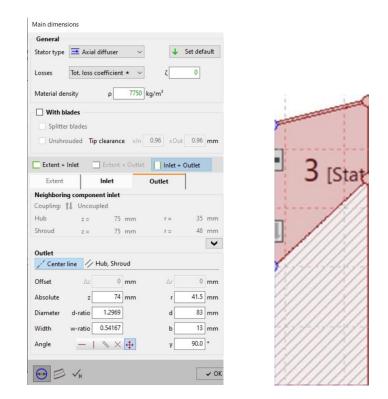
Picture 190 Blade trailing edge CFturbo program

# 4.2.3 Stator

Likewise, here is a suitably pre-drilled tapered shaft that rotates in a tapered rotor:



Picture 191Conical stator input set up CFturbo program



Picture 192 Conical stator output set up CFturbo program

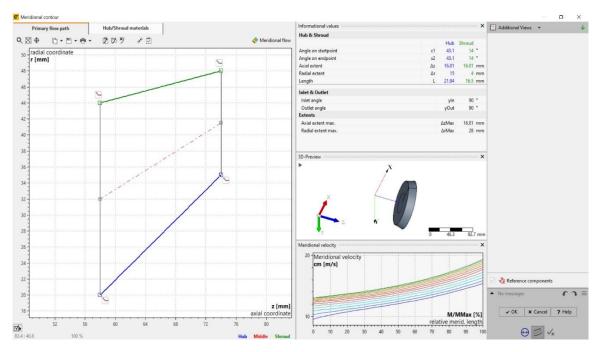
Picture 193 Stator CFturbo program
[106]

Meridian Values				∽Outlet			
/Inlet				Average diameter	dm2	83	mm
Average diameter	dm1	64	mm	Width	b2	13	mm
Width	b1	24	mm	Area	A2	3390	mm
Area	A1	4825	mm²	✓Ratio to downstream inlet			
Ratio to upstream outlet				Diameter ratio	d-Ratio	1	
Diameter ratio	d-Ratio	1		Width ratio	b-Ratio	1	
Width ratio	b-Ratio	1		Area ratio	A-Ratio	1	
Area ratio	A-Ratio	1		∨Outlet - Flow properties			
Inlet - Flow properties				Meridional velocity	cm2	15.4	m/s
Meridional velocity	cm1	10.8	m/s	Abs. circumferential velocity	cu2	22.5	m/s
Abs. circumferential velocity	cu1	29.1	m/s	Absolute velocity	c2	27.3	m/:
Absolute velocity	c1	31.1	m/s	Absolute flow angle	α2	34.5	
Absolute flow angle	α1	20.4	•	Density	o2	1223	ka/
Density	p1	1223	kg/m <sup>3</sup>	Static pressure	p2	11.46	
Static pressure	p1	10.09	bar	Temperature	T2	-200	
Temperature	T1	-200	°C	Total density	pt2	1223	
Total density	pt1	1223	kg/m <sup>3</sup>	Total pressure	pt2		bar
Total pressure	pt1	16	bar		Tt2	-200	
Total temperature	Tt1	-200	°C	Total temperature			-
Volume flow	Q1	188.4	m³/h	Volume flow	Q2	188.4	
Mass flow	m1	64	kg/s	Mass flow	m2		kg/
Swirl	s1	0.932	m²/s	Swirl	s2	0.932	m*/

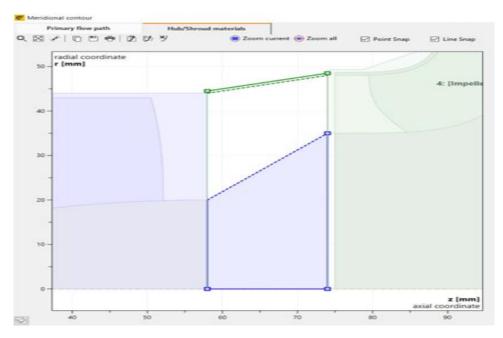
×

Picture 194 The values of all velocities, pressures and flow angles at the inlet of the stator CFturbo program

Picture 195 The values of all velocities, pressures and flow angles at the outlet of the stator CFturbo program

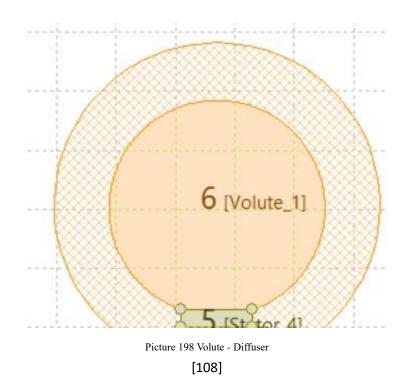


Picture 196 Meridian contour and B-spline configuration CFturbo program

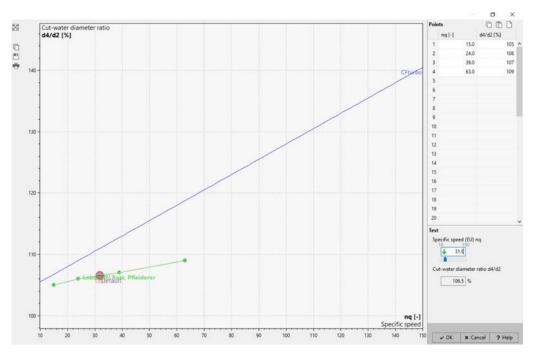


Picture 197 The material of the conic shaft and conic stator CFturbo program

## 4.2.4 Volute-Diffuser

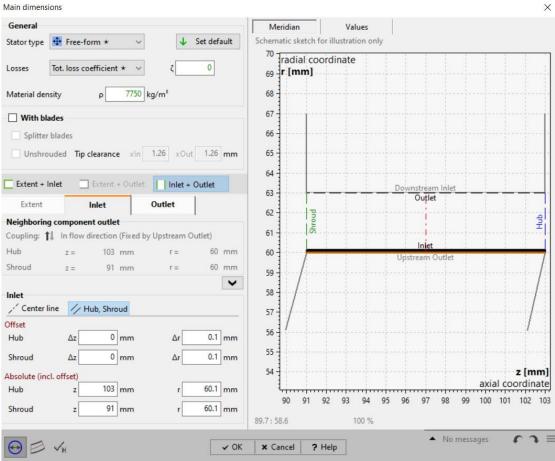


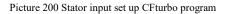
In this case, the ration  $d_4/d_2$  was chosen 1.05 and combined with the following function with the special number of revolutions comes out 1.065. The difference is small. The distance includes the distance between the impeller and the shell which is 0.1mm.



Picture 199 Function of the ratio d4/d2 with the specific speed number CFturbo program

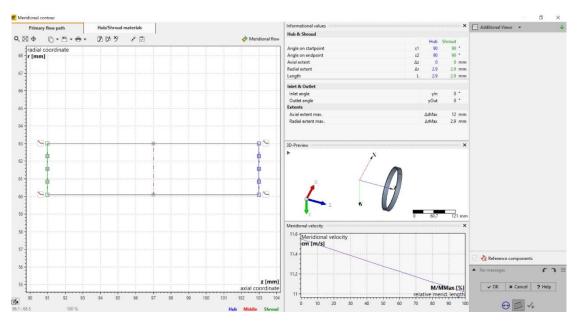
The distance  $\Delta r$  was similarly defined.



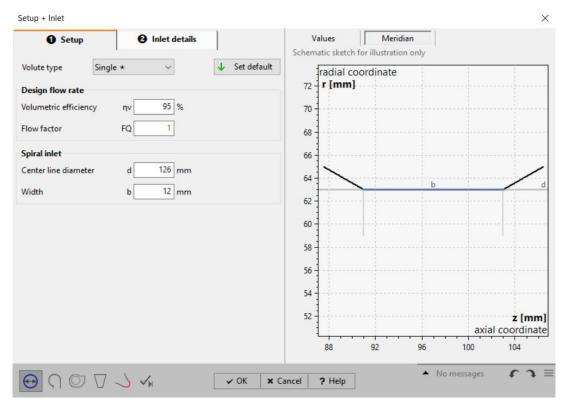


			×				×
Meridian Values				Meridian Values			
∽Inlet				YOutlet			
Average diameter	dm1	120.2	mm	Average diameter	dm2	126	mm
Width	ь1	12	mm	Width	b2	12	mm
Area	A1	4531	mm²	Area	A2	4750	mm²
✓Ratio to upstream outlet				✓Ratio to downstream inlet			
Diameter ratio	d-Ratio	1.002		Diameter ratio	d-Ratio	1	
Width ratio	b-Ratio	1		Width ratio	b-Ratio	1	
Area ratio	A-Ratio	1.002		Area ratio	A-Ratio	1	
✓Inlet - Flow properties				VOutlet - Flow properties			
Meridional velocity	cm1	11.5	m/s	Meridional velocity	cm2	11	m/s
Abs. circumferential velocity	cu1		m/s	Abs. circumferential velocity	cu2	69.6	m/s
Absolute velocity	c1	73.9	m/s	Absolute velocity	c2	70.5	m/s
Absolute flow angle	a1	9		Absolute flow angle	α2	9	
Density	p1	1223	kg/m <sup>1</sup>	Density	ρ2	1223	kg/m
Static pressure	p1	63.6	-	Static pressure	p2	66.6	bar
Temperature	TI	-200	•c	Temperature	T2	-200	°C
Total density	pt1		kg/m <sup>1</sup>	Total density	pt2	1223	kg/m
Total pressure	pt1		bar	Total pressure	pt2	97	bar
Total temperature	Tel	-200		Total temperature	Tt2	-200	*C
Volume flow	Q1	188.4	m²/h	Volume flow	Q2	188.4	m²/h
Mass flow	m1	64	kg/s	Mass flow	m2	64	kg/s
Swirl	\$1	4.384	-	Swirt	s2	4.384	m²/s
× Cancel 2 Help	<ul> <li>No messages</li> </ul>	r	<b>२</b> ≡	× Cancel ? Help	<ul> <li>No messages</li> </ul>	ſ	2

Picture 201 The values of all velocities, pressures and flow angles at the inlet of the stator CFturbo program Picture 202 The values of all velocities, pressures and flow angles at the outlet of the stator CFturbo program



Picture 203 Meridian contour and B-spline configuration CFturbo program



Picture 204 Start of set up of volute - diffuser CFturbo program

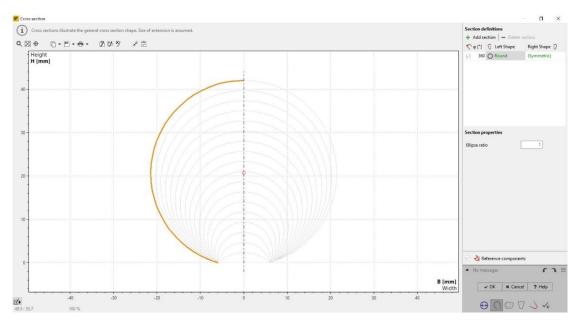
Setup + Inlet	t						×
0 9	Setup	🕑 Inlet d	etails			leridian	
Neighborin	g componer	nt outlet			Schematic sketch for illustr	ration only	
Coupling:	1 In flow d	lirection (Fixed by U	pstream Outle	t)	radial coordina	te	
Hub	z =	103 mm	r =	63 mm	72 – <b>r [mm]</b>		+
Shroud	z =	91 mm	r =	63 mm			
				~	-		
Inlet					68 -		
/ Center	line // H	ub, Shroud					
Offset	Δz	0 mm	Δr	0 mm	64		/
Absolute	z	97 mm	r	63 mm	04	b	d
			d	126 mm	1		
			b	12 mm	60		1
Angle	- 1	$\wedge \times +$	Y	180.0 •	-		
					56		
					-		
					52 -		z [mm] axial coordinate
					90		100
	877			. 01		▲ No mes	sages f 🤉 🗄
	O V	J VH		OK XC	ancel ? Help		

Picture 205 Entrance dimensions CFturbo program

The thickness was formed a little higher, with 2.5 mm, and the method designed is with the constant vorticity (x=1), whereas with this method, we have a high hydraulic efficiency. By estimation, the hydraulic grade manually entered 95%. The diameter and height were adjusted so that we have an outlet pressure of about 86 bar.

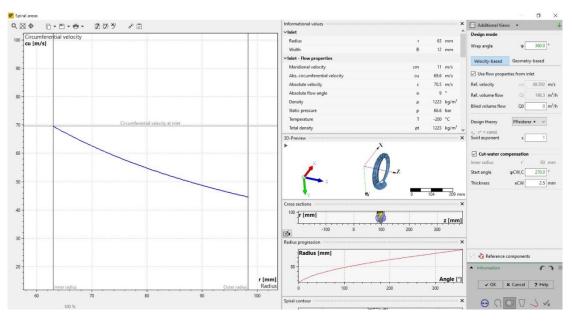
Swirl	S	4.384	m²/s
Mass flow	ń		kg/s
Volume flow	Q	188.4	m³/h
Total temperature	Tt	-200	°C
Total pressure	pt	97	bar
Total density	pt	1223	kg/m
Temperature	Т	-200	°C
Static pressure	р	66.6	bar
Density	ρ	1223	kg/n
Absolute flow angle	α	9	٠
Absolute velocity	c	70.5	m/s
Abs. circumferential velocity	cu	69.6	m/s
Meridional velocity	cm	11	m/s
✓Inlet - Flow properties			
Spiral Inlet width ratio	b-Ratio	100	%
Spiral Inlet diameter ratio	d-Ratio	100	%
Ratios to previous component	t		
Internal volume flow	Qi	198.3	m³/h
Values Meridia	n		

Picture 206 The values of all velocities, pressures and flow angles at the inlet of the volute - diffuser CFturbo program



Picture 207 Configuration of spiral tube shape and size CFturbo program

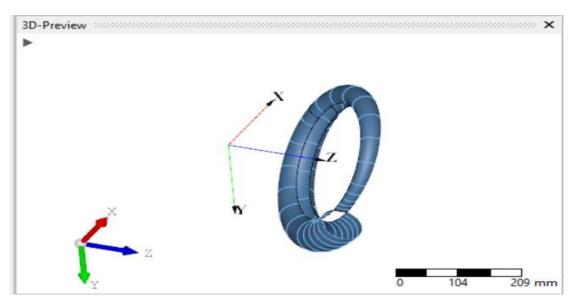
It is noted that the stator added before the diffuser is considered part of the diffuser. For this reason, d4/d2 = 105% and not 100% is mentioned in the set up.



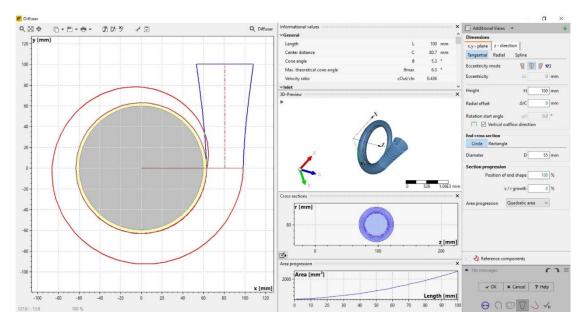
Picture 208 Spiral design space based on some criteria CFturbo program

Informational values				×	Informational values				
∽Inlet				^	Width	В	36.44		
Radius	r	63	mm		Side ratio	H/B	97.2		
Width	В	12	mm		Area	A		mm²	
✓Inlet - Flow properties					Area/Radius	A/rc	12.88	mm	
Meridional velocity	cm	11	m/s		✓Last spiral section - Flow properties				
Abs. circumferential velocity	cu	69.6	m/s		Meridional velocity	cm	0	m/s	
Absolute velocity	c	70.5	m/s		Abs. circumferential velocity	cu	50.6	m/s	
Absolute flow angle	α	9	•		Absolute velocity	c	50.6	m/s	
Density	ρ	1223	kg/m <sup>3</sup>		Absolute flow angle	α	90	•	
Static pressure	р	66.6	bar		Density	ρ	1223	kg/m <sup>3</sup>	ł.
Temperature	т	-200	°C		Static pressure	P	75.9	bar	
Total density	pt	1223	kg/m <sup>3</sup>		Temperature	т	-200	°C	
Total pressure	pt		bar		Total density	pt	1223	kg/m <sup>3</sup>	ł.
Total temperature	Tt	-200	°C		Total pressure	pt	91.5	bar	
Volume flow	Q	188.4	m <sup>3</sup> /h		Total temperature	Tt	-200	*C	
Mass flow	m	64	kg/s		Volume flow	Q	188.4	m³/h	
Swirl	5	4.384	-		Mass flow	m	64	kg/s	
✓Last spiral section					Swirl	5	4.063	m²/s	
Inner radius	1	63	mm		∼Losses				
Outer radius	r	98.4	mm		Sizing parameter	SP	1.079		
Equivalent diameter	D	36.3	mm		Meridional loss coefficient	km	0.024		
Min. axial coordinate	z	78.8	mm		Tangential loss coefficient	ku	0.054		
Height	н	35.42			Wall loss coefficient	kw	0.102		
Width	В	36.44			Overall loss coefficient	k	0.181		
Side ratio	H/B	97.2		~	Total pressure loss	Δpt	5.49	bar	

Picture 209 The values of all velocities, pressures and flow angles at the inlet and the last spiral section CFturbo program Picture 210 The values of all velocities, pressures and flow angles at the inlet and the last spiral section CFturbo program



Picture 211 The spiral CFturbo program



Picture 212 Configuration of conical diffuser at the outlet CFturbo program

Informational values				Informational values				>
∀General			,	Density	ρ	1223	kg/m <sup>3</sup>	
Length	L	100	mm	Static pressure	P	75.9	bar	
Center distance	С	80.7	mm	Temperature	т	-200	*C	
Cone angle	9	5.3	•	Total density	pt	1223	kg/m <sup>3</sup>	
Max. theoretical cone angle	θmax	6.3	•	Total pressure	pt	91.5	bar	
Velocity ratio	cOut/cln	0.436		Total temperature	Tt	-200	*C	
∨Inlet				Volume flow	Q	188.4	m³/h	
Equivalent diameter	D	36.3	mm	Mass flow	m	64	kg/s	
Area	A	1035	mm²	Swirl	s	4.063	m²/s	
vOutlet				✓Outlet - Flow properties				
Equivalent diameter	D	55	mm	Absolute velocity	c	22	m/s	
Area	A	2376	mm²	Density	ρ	1223	kg/m <sup>3</sup>	
Diffuser center position	Cx	80.7	mm	Static pressure	P	85.8	bar	
Diffuser center position	Су	100	mm	Temperature	т	-200	*C	
Diffuser center position	Cz	97	mm	Total density	pt	1223	kg/m <sup>1</sup>	
Last spiral section - Flow properties				Total pressure	pt	88.8	bar	
Meridional velocity	cm	0	m/s	Total temperature	Tt	-200	*C	
Abs. circumferential velocity	cu	50.6	m/s	Volume flow	Q	188.4	m³/h	
Absolute velocity	c	50.6	m/s	Mass flow	m	64	kg/s	
Absolute flow angle	α	90	•	≁Losses				
Density	ρ	1223	kg/m <sup>3</sup>	Cone loss coefficient	kc	0.164		
Static pressure	р	75.9	bar	Wall loss coefficient	kw	0.011		
Temperature	Т	-200	°C	Overall loss coefficient	k	0.175		
Total density	pt	1223	kg/m <sup>3</sup>	Total pressure loss	Δpt	2.74	bar	

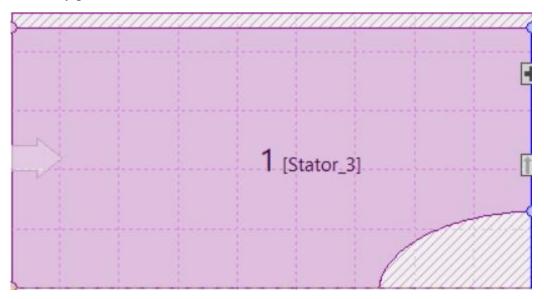
Picture 213 The values of all velocities, pressures and flow angles at the outlet of diffuser CFturbo program

Picture 214 The values of all velocities, pressures and flow angles at the outlet of diffuser CFturbo program

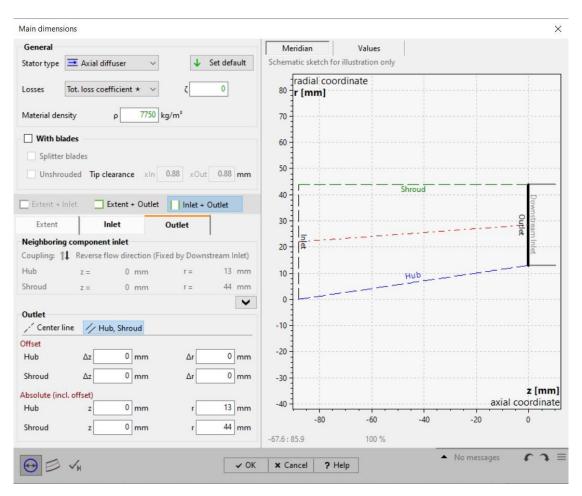
③◆ □・巴・冉・ 凶以汐 孑立	Q Cut-water	Informational values		×	Additional Views	
y [mm]		✓Inlet (before cutwater)			Design mode	
y [mm]		Equivalent diameter	D	36.3 mm	Simple Fillet Sh	arp
		Area	A	1035 mm <sup>2</sup>		
		✓Cutwater (throat)			Fillet radius	R 2 mr
		Equivalent diameter	D	41.72 mm		
		Throat area	A	1367 mm <sup>2</sup>	Diffuser base form facto	r 50 %
		Angular position	φC,1	36 *	Automatic spiral star	
		Inner angle	ain	6.6 *	Mutomatic spiral star	
		Outer angle	αOut	31.7 *	Spiral start position	φC,0 22.3 *
		Average angle	αAvg	18 *		
		Inner diameter	din	142.6 mm	Surface transition	Non-tangential * 🕤
		Outer diameter	dOut	150 mm		
		Average diameter	dAvg	145.3 mm		
		Minimal diameter	dMin	142.6 mm		
		<ul> <li>✓Outlet (diffuser discharge)</li> </ul>				
		Equivalent diameter	D	55 mm		
		Area	A	2376 mm²		
		3D-Preview		×		
			2			
		•		8 535 mm		
		Cross sections		s 335 mm		
			-246 /1			
		100 <b>r [mm]</b>				
		-		z [mm]	Reference comp	
		0	100 200			
					<ul> <li>No messages</li> </ul>	53
		Area progression		X		
	x [mm]	2000 - Area [mm <sup>2</sup> ]		ength [mm]	✓ OK × C	ancel ? Help

Picture 215 Curvature shaping space between conical diffuser and spiral CFturbo program

#### 4.2.5 Rotary part of the inducer



Picture 216 Rotary inducer nose & inducer input space CFturbo program

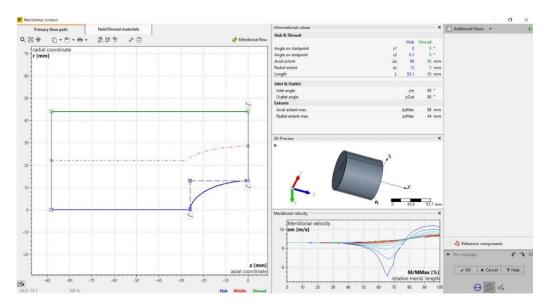


Picture217 Set up the input area of the inducer CFturbo program

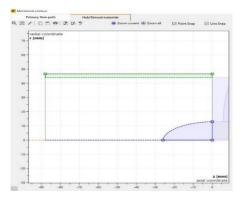
			Meridian Values			
			∼Outlet			
dm1	57	mm	Average diameter	dm2	57	mm
b1	44	mm	Width	b2	31	mm
A1	7880	mm²	Area	A2	5550	mm
			✓Ratio to downstream inlet			
cm1	6.6	m/s	Diameter ratio	d-Ratio	1	
cu1	0	m/s	Width ratio	b-Ratio	1	
c1	6.6	m/s	Area ratio	A-Ratio	1	
α1	90	•	✓Outlet - Flow properties			
ρ1	1223	kg/m <sup>3</sup>	Meridional velocity	cm2	9.4	m/s
p1	6.73	bar	Abs. circumferential velocity	cu2	0	m/s
T1	-200	°C	Absolute velocity	c2	9.4	m/s
pt1	1223	kg/m <sup>3</sup>	Absolute flow angle	o2	90	•
pt1	7	bar	Density	ρ2	1223	kg/i
Tt1	-200	°C	Static pressure	p2	6.46	bar
Q1	188.4	m³/h	Temperature	T2	-200	*C
m1	64	kg/s	Total density	pt2	1223	kg/r
s1	0	m²/s	Total pressure	pt2	7	bar
			Total temperature	Tt2	-200	*C
			Volume flow	Q2	188.4	m²/
			Mass flow	m2	64	kg/s
			Swirl	\$2	0	m2/3
	b1 A1 cu1 c1 α1 p1 p1 T1 pt1 Tt1 Q1 m1	b1 44 A1 7880 cm1 6.6 cu1 0 c1 6.6 a1 90 p1 1223 p1 6.73 T1 -200 pt1 1223 pt1 7 Tt1 -200 q1 188.4 m1 64	b1 44 mm A1 7880 mm <sup>2</sup> cm1 6.6 m/s cu1 0 m/s c1 6.6 m/s α1 90 ° ρ1 1223 kg/m <sup>3</sup> ρ1 6.73 bar T1 -200 °C ρt1 1223 kg/m <sup>3</sup> ρt1 7 bar Tt1 -200 °C Q1 188.4 m <sup>3</sup> /h m1 64 kg/s	b1     44 mm     Width       A1     7880 mm²     Area       cm1     6.6 m/s     Area ratio       cu1     0 m/s     Width ratio       cu1     0 m/s     Width ratio       cu1     0 m/s     Area ratio       cu1     200 °C     Absolute rlow angle       pt1     1223 kg/m³     Absolute flow angle       pt1     7 bar     Density       Tt1     -200 °C     Static pressure       Q1     188.4 m³/h     Temperature       m1     64 kg/s     Total density       s1     0 m²/s     Total pressure	harmAverage dameterdum2b144 mmWidthb2A17880 mm²AreaA2 $cm1$ 6.6 m/s $Area$ b1cu10 m/s $Midth$ ratio $A-Ratiocu10 m/sAreaAreacu10 m/sMidth ratioA-Ratiocu10 m/sArea ratioA-Ratiocu10 m/sMidth ratioA-Ratiocu10 m/sMidth ratioA-Ratiocu190 *-Outlet - Flow propertiesCu2p11223 kg/m³Meridional velocitycu2p11223 kg/m³Absolute flow anglec2p117 barDensityc2p117 barDensityp2p117 barTemperaturep2p1164 kg/sTotal densitypt2s10 m²/sTotal densitypt2total pressurept2total temperaturett2$	hand       A mm       Width       b2       31         hand       7880 mm²       Area       A2       350 $A$ 1       7880 mm²       Area       A2       350 $C$ 6.6 m/s $\sim$ Ratio to downstream inlet $\sim$ Ratio to downstream inlet       1 $C$ 0 m/s $\sim$ Ratio to downstream inlet       1 $C$ 6.6 m/s       Area ratio $\sim$ Ratio to downstream inlet       1 $C$ 6.6 m/s       Area ratio $\sim$ Ratio to downstream inlet       1 $C$ 1       1223 kg/m²       Meidional velocity $C$ 9.4 $\rho$ 1       1223 kg/m²       Meidional velocity $C$ 9.4 $\rho$ 1       1223 kg/m²       Absolute velocity $C$ 9.4 $\rho$ 1       1223 kg/m²       Absolute flow angle $O$ $O$ $O$ $\rho$ 1       1223 kg/m²       Absolute flow angle $O$ </td

Picture218 The values of all velocities, pressures and flow angles at the inlet CFturbo program

Picture 219 The values of all velocities, pressures and flow angles at the end of the rotary nose CFturbo program

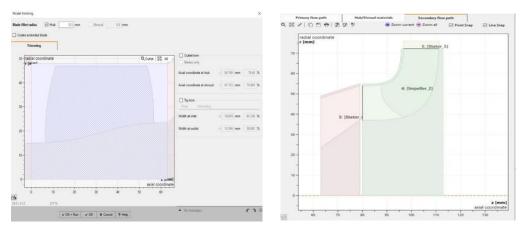


Picture 220 Meridian contour and B-spline configuration CFturbo program



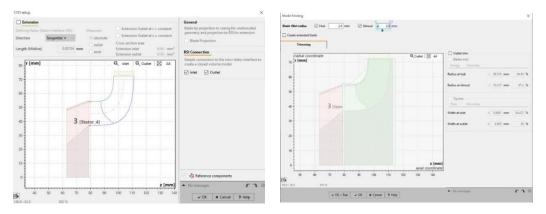
#### 4.3 CFD analysis

Since the design has been completed, appropriate preparation was made for its export to the Simscale program. First, the housing where you find the impeller was deleted to simplify the CFD. Then, a 0.5mm fillet and a fillet in the impeller, equal to 1.5 times the thickness of the fin [36], were made in the inducer. Then, the flow extension of the diffuser was set 4 times the exit diameter. Finally, the flow volume was defined and with the grid extension of the Rotating Zone 0.5mm. The pictures below show the actions taken for the hydrogen pump. Exactly the same happened with the oxygen pump.



Picture 222 Fillet of inducer CFturbo program

Picture 223 Impeller without the curve CFturbo program



Picture 224 CFD impeller setup CFturbo program

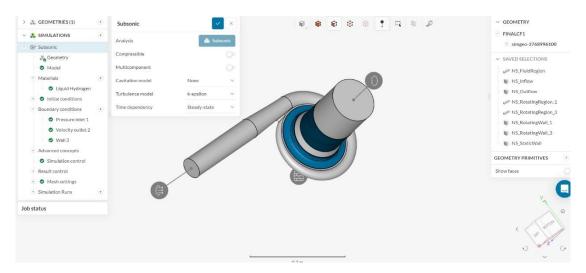
Picture 225 Fillet of inducer CFturbo program

Exter	sion				
Add exter	sion with	h constant cr	oss sec	tion (ext	rusion)
Length			180	mm	
Factor			4	x D6	
Equiv. dia	meter	D6 =	45	mm	
No mes	isages			r	7

Picture 226 CFD extension to diffuser outlet CFturbo program

#### 4.3.1 Simulation settings for CFD analysis

Initially, before exporting to Simscale ,the flow volume was set automatically by the CFturbo program and a 0.5 mm mesh extension was made in the rotating zones of the inductor and impeller. Then, using an API key from Simscale, the model was exported in STEP file format. Finally, the model was imported into the Simscale program.



Picture 227 Simscale program environment Simscale program

In the next step, all the necessary settings for the simulation were made. The following images show all the necessary simulation settings. The material, the rotating walls, the rotating zone, the inlet pressure, the outlet flow, the simulation control, the output data measurements, total pressure difference, torque, force and the mesh. Exactly the same settings were made for the oxygen pump for an inlet pressure of 7 bar, outlet flow of 64 Kg/s and number of revolutions of 20000 rpm. The flow chosen is Subsonic because in this case the velocities are very high. Also, to simplify the analysis we assume that there is no heat flow from the walls and the flow is incompressible.

Liquid Hydrogen		×	Velocity outlet 1	$\sim$	×
Fluids	Material	~	Boundary conditions	Velocity outlet	~
Fluid type	Liquid	~	Velocity type	Flow rate	~
Viscosity model	Newtonian	~	Flow rate type	(ṁ) Mass flow	~
(v) Kinematic viscosity	2.292e-7	m²/s $\sim$	Flow rate	20 k	kg∕s ∨
(ρ) Density	73.72	kg/m² $\sim$		<u>**</u>	Reset
Assigned Volumes (1)		Clear list	Assigned Faces (1)	C	lear list
NS_FluidRegion		×	NS_Outflow@NS_FluidReg	ion	×
-5 🗊			Ŵ		

Picture 228 Properties of material Simscale program

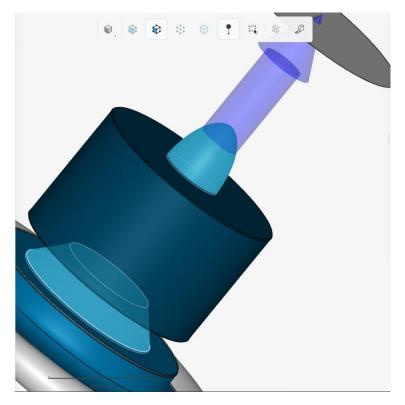
Picture 229 Output mass flow Simscale program

Boundary conditions	Pressure inlet	~
Pressure type	Total pressure	~
(P <sub>t</sub> ) Gauge total pressure	7	bar 🗸
	<u>[27</u>	Rese
Assigned Faces (1)		Clear lis
NS_Inflow@NS_FluidRegion		×

Picture 230 Total inlet pressure Simscale program

> 10	GEOMETRIES (1)	+	Wall 3		~ >
	SIMULATIONS	+	Boundary conditions	Wall	~
	Geometry		(U) Velocity	Rotating wall	~
	Model		✓ Point on axis		• 6
-	Materials	+	x	0	m ~
	<ul> <li>Liquid Hydrogen</li> </ul>				- 100
+	Initial conditions		У	0	m∨
Ξ		+	Z	0	m ~
	Pressure inlet 1		✓ Rotation axis		
	<ul> <li>Velocity outlet 2</li> </ul>		4		
	🖉 Wall 3		x	0	m∨
+	Advanced concepts		у	0	m ~
	Simulation control		z	-1	m ~
+	Result control				
+	Mesh settings		(ω) Rotational velocity	8e+4	RPM ~
+	Simulation Runs	(+)	-	5	Reset
Job st	atus		Assigned Faces (3)		Clear list
			NS_StaticWall@NS_Fluid	Region	×
			NS_StaticWall@NS_Fluid	Region	۲
			NS_StaticWall@NS_Fluid	Region	×
			Ô		

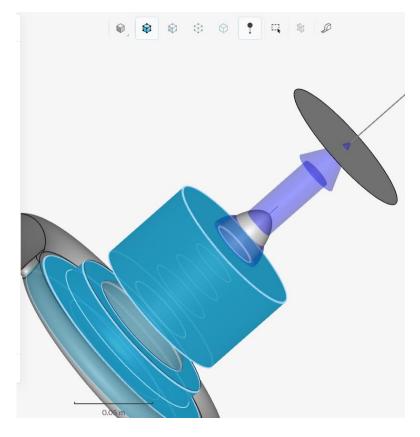
Picture 231 Setting up rotating walls Simscale program



Picture 232 Rotating walls Simscale program

E E	Rotating zones ~ origin x y z ~ Axis	MRF rota 0 0	ating zone v • • • m v m v m v
	x y z	0	m V
	y z	0	m ~
	z		
Ð		0	m ~
Ð	× Avie		
	6A13		
	x	0	m ~
	У	0	m ~
	z	-1	m V
Ð	$(\omega)$ Rotational velocity	8e+4	RPM ~
			Rese
	Assigned Volumes (2)		Clearlis
	NS_RotatingRegion_3 NS RotatingRegion 1		×
Ð			Ŭ
		y z (ω) Rotational velocity  Assigned Volumes (2) NS_RotatingRegion_3 NS_RotatingRegion_1	y 0 z -1 (ω) Rotational velocity 8e+4 Assigned Volumes (2) NS_RotatingRegion_3 NS_RotatingRegion_1

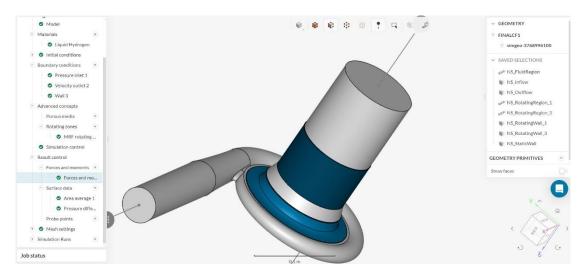
Picture 233 Setting up rotating zones Simscale program



Picture 234 Rotating zones Simscale program

Simulation control		/ ×
Number of iterations	3000	
Write control	Iterations	~
Write interval	3000	
Convergence criteria	0.001	
Number of processors	32 Cores	~
Maximum runtime	3e+4	s∨

Picture 235 Simulation control setup Simscale program

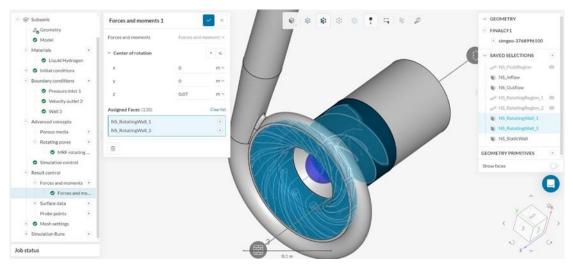


Picture 236 The model and enter result control settings Simscale program

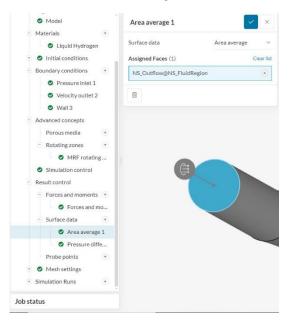
Forces and moments 1		
Forces and moments	Forces and	l moment: ~
<ul> <li>Center of rotation</li> </ul>		+ 6
x	0	m ~
У	0	m ~
z	0.07	m ~
Assigned Faces (138)		Clear lis
NS_RotatingWall_1		×
NS_RotatingWall_3		×

Picture 237 Force and torque measurement setup Simscale program

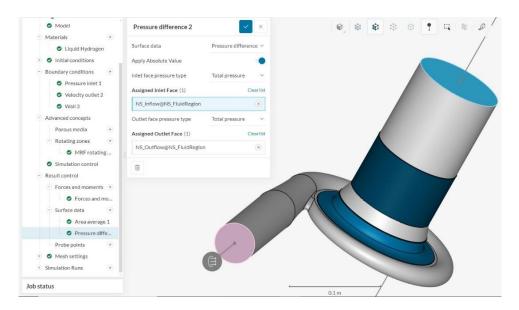
The center of mass was roughly defined. However, experimentally, it was found that it does not affect the results to a large extent.



Picture 238 The center of mass of the model and the rotating walls from the rotation zones Simscale program



Picture 239 Setting average value measurements at the output Simscale program



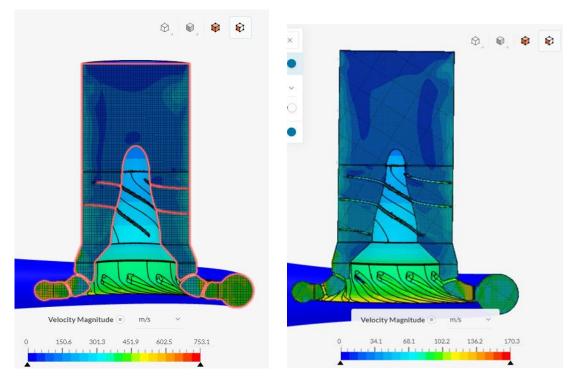
Picture 240 Setting up total pressure difference measurements Simscale program

Mesh settings		
Mesh settings	Manual ~	
Cell size specification	Relative to CAD $\sim$	
Minimum cell size	1e-5	
Maximum cell size	0.002	
Cell size on surfaces	0.001	
Merge CAD surfaces	•	

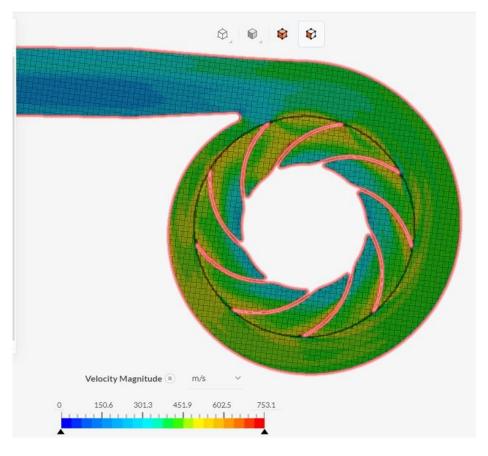
Picture 241 Manual setting of mesh Simscale program

### 4.3.2 Results of CFD analysis

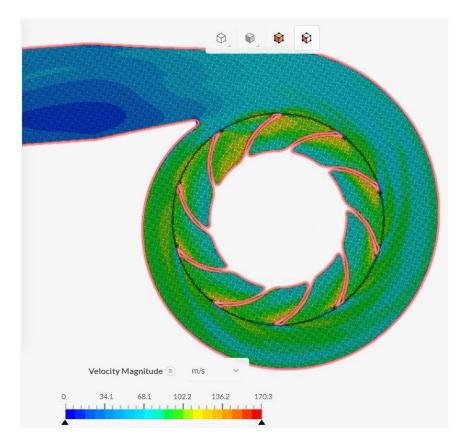
The figures below show the effects of speed and total pressure on the impeller and commutator on the liquid hydrogen and liquid oxygen pumps.



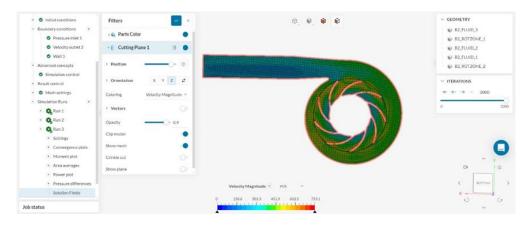
Picture 242 Velocity Magnitude on the XZ level of pump LH<sub>2</sub> Simscale program Picture 243 Velocity Magnitude on the XZ level of pump LO<sub>2</sub> Simscale program



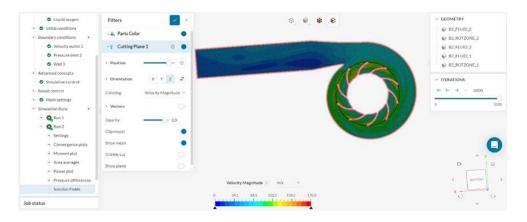
Picture244 Velocity Magnitude on the pump impeller LH2 Simscale program



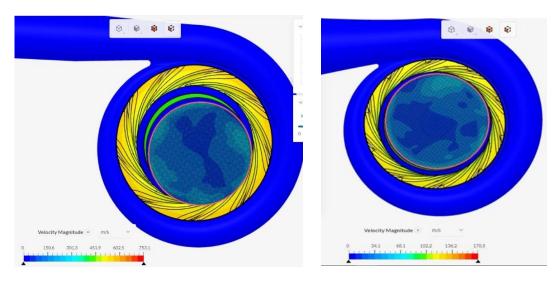
Picture 245 Velocity Magnitude on the pump impeller  $\mathrm{LO}_2$  Simscale program



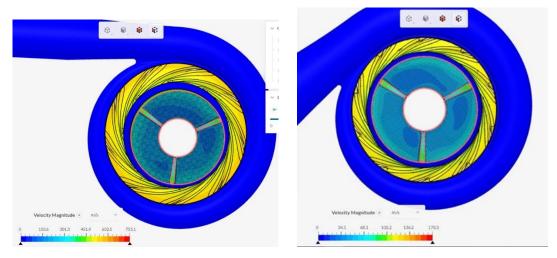
Picture 246 Velocity Magnitude on the pump impeller  $LH_2$  (the whole picture) Simscale program



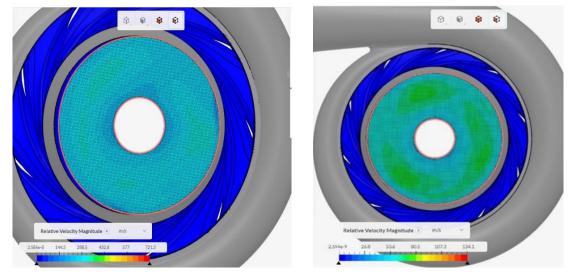
Picture 247 Velocity Magnitude on the pump impeller  $LO_2$  (the whole picture) Simscale program



Picture 248 Velocity Magnitude on the pump LH<sub>2</sub> at the input Simscale program Picture 249 Velocity Magnitude on the pump LO<sub>2</sub> at the input Simscale program

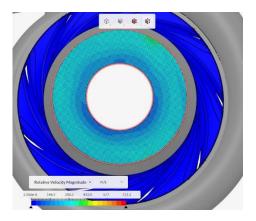


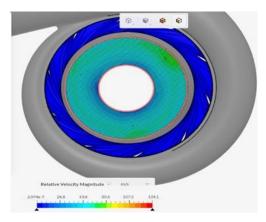
Picture 250 Velocity Magnitude on the pump inducer LH<sub>2</sub> Simscale program Picture 251 Velocity Magnitude on the pump inducer LO<sub>2</sub> Simscale program



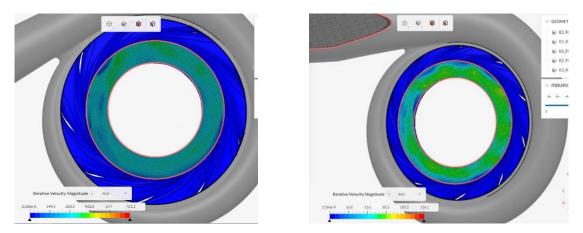
Picture 2523 Relative Velocity Magnitude at the inducer input LH<sub>2</sub> Simscale program

Picture 253 Relative Velocity Magnitude at the inducer input  $\mathrm{LO}_2$  Simscale program

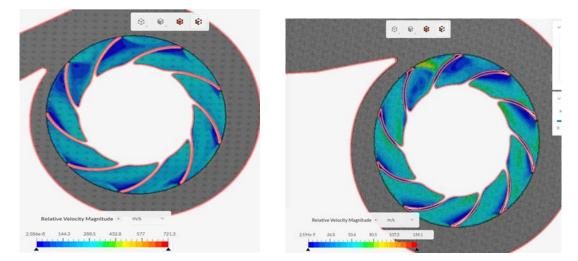




Picture 254 Relative Velocity Magnitude at the inducer output LH<sub>2</sub> Simscale program Picture 255 Relative Velocity Magnitude at the inducer output LO<sub>2</sub> Simscale program

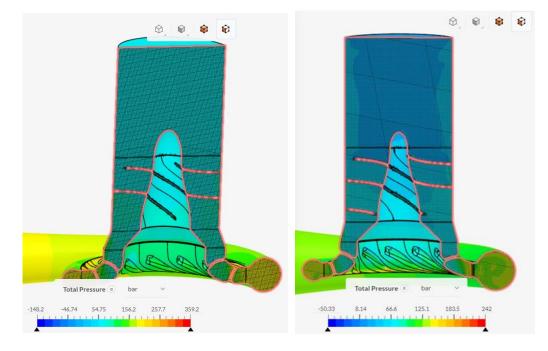


 $\label{eq:product} Picture \ 256 \ Relative \ Velocity \ Magnitude \ at the impeller \ input \ LH_2 \ Simscale \ program \\ Picture \ 257 \ Relative \ Velocity \ Magnitude \ at the impeller \ input \ LO_2 \ Simscale \ program \\ Picture \ 257 \ Relative \ Velocity \ Magnitude \ at the impeller \ input \ LO_2 \ Simscale \ program \\ Picture \ 257 \ Relative \ Velocity \ Magnitude \ at the impeller \ input \ LO_2 \ Simscale \ program \\ Picture \ 257 \ Relative \ Velocity \ Magnitude \ at the impeller \ input \ LO_2 \ Simscale \ program \\ Picture \ 257 \ Relative \ Velocity \ Magnitude \ at the impeller \ Simscale \ Picture \ Simscale \ Picture \ Simscale \ Picture \ Simscale \ Picture \ Simscale \$ 

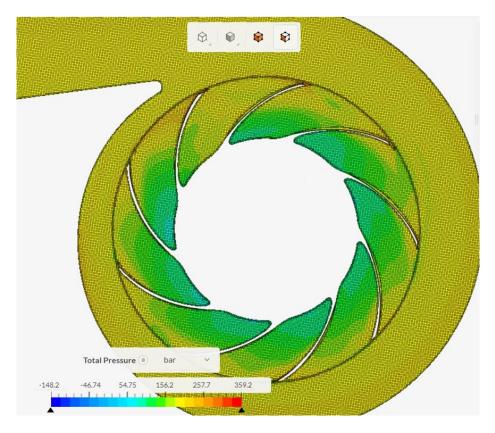


Picture 258Relative Velocity Magnitude at the impeller LH<sub>2</sub> Simscale program Picture 259 Relative Velocity Magnitude at the impeller LO<sub>2</sub> Simscale program

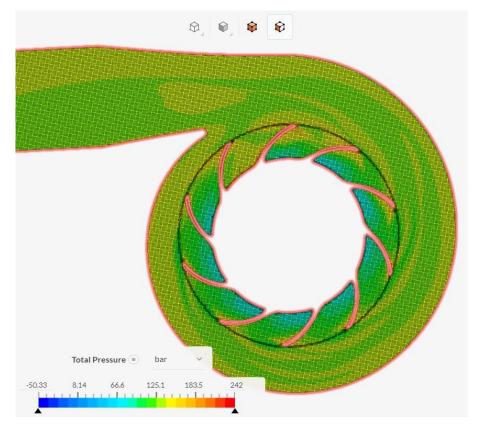
From the above photos of the velocities, strong eddies and flow reversals are observed at the inlet. The speed increase in the inducer and impeller is also observed. This is followed by normalization and velocity reduction in the diffuser. Finally, there are the relative speed decelerations that occur in the inductor and the impeller. Below are the pressures.



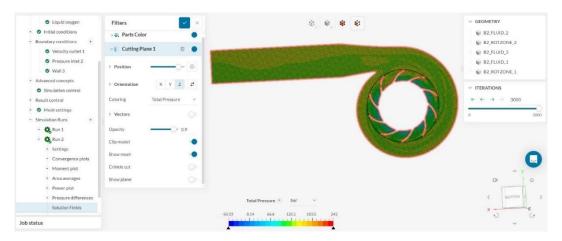
Picture 260 Total Pressure on the XZ level of pump  $LH_2$  Simscale program Picture 261 Total Pressure on the XZ level of pump  $LO_2$  Simscale program



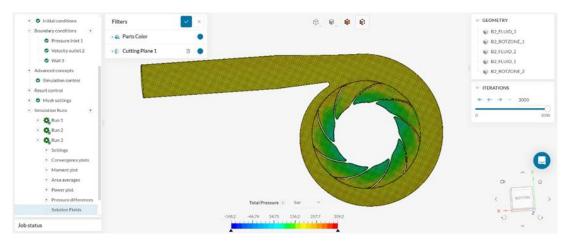
Picture 262 Total Pressure on the pump impeller LH<sub>2</sub> Simscale program



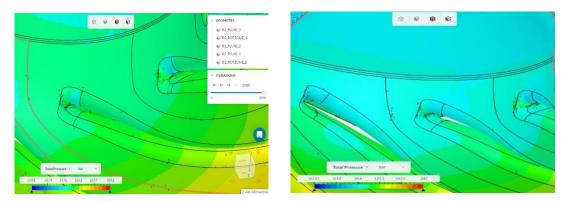
Picture 263 Total Pressure on the pump impeller LO2 Simscale program



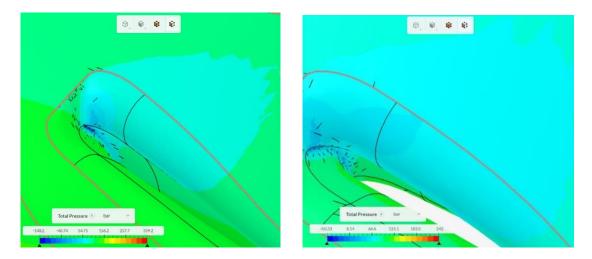
Picture 264 Total Pressure on the pump impeller  $LH_2$  (the whole picture) Simscale program



Picture 265 Total Pressure on the pump impeller  $\mathrm{LO}_2$  (the whole picture) Simscale program

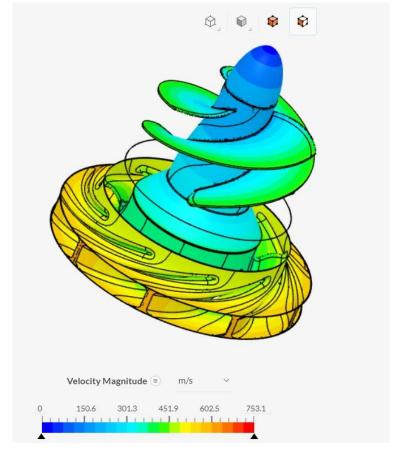


Picture 266 Development of small cavitation at the hub of the wings (pump LH<sub>2</sub>) Simscale program Picture 267 Development of small cavitation at the hub of the wings (pump LO<sub>2</sub>) Simscale program

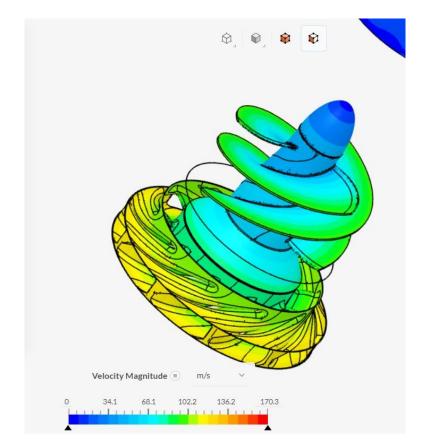


Picture 268 The blade in which intense cavitation develops (pump LH<sub>2</sub>) Simscale program Picture 269 The blade in which intense cavitation develops (pump LO<sub>2</sub>) Simscale program

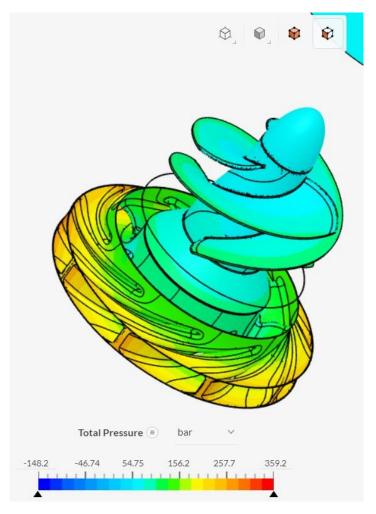
From the results, it can be seen that the cavitation is developing locally. However, intense cavitation was detected only on one wing locally on the leading edge. In addition, the cavitation is more pronounced in the hydrogen pump. Here a small development of cavitation at the base of the wing is observed. This may be due to hydraulic pressure losses between the rotating conical shaft and the walls. It can also be caused by the excessive inlet speed of the impeller. Excessive cavitation development is observed locally on the front side of the inducer blade. This is shown in the results below where the images are 3D.



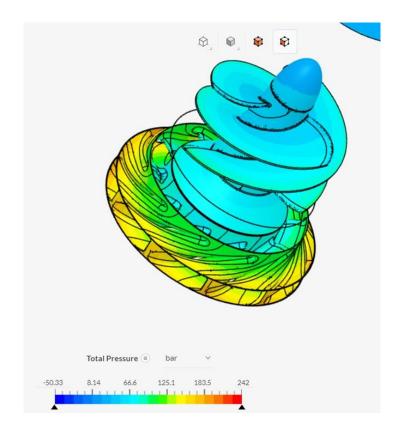
Picture 270 Velocity Magnitude 3D on the inducer and impeller of pump  $LH_2$  Simscale program



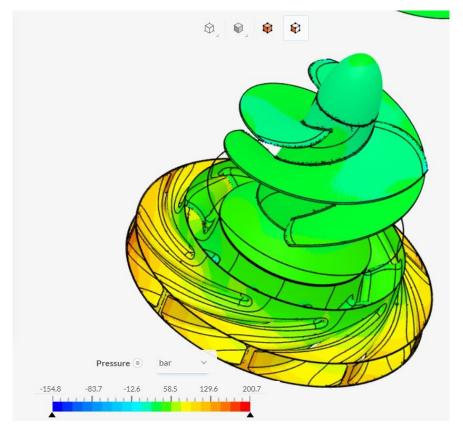
Picture 271 Velocity Magnitude 3D on the inducer and impeller of pump  $\mathrm{LO}_2$  Simscale program



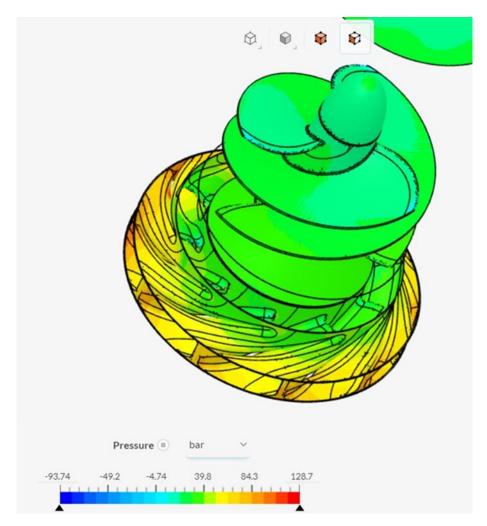
Picture 272 Total pressure 3D on the inducer and impeller of pump  $L\mathrm{H}_2$  Simscale program



Picture 273 Total pressure 3D on the inducer and impeller of pump  $\mathrm{LO}_2$  Simscale program

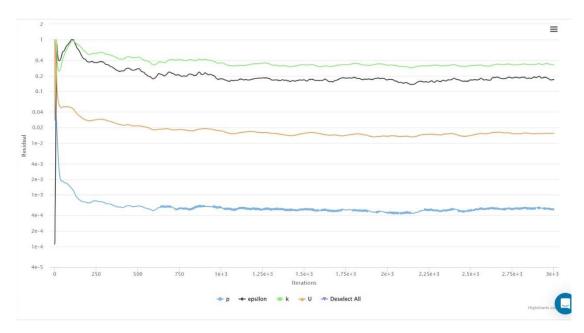


Picture 274 Static pressure 3D on the inducer and impeller of pump  $LH_2$  Simscale program

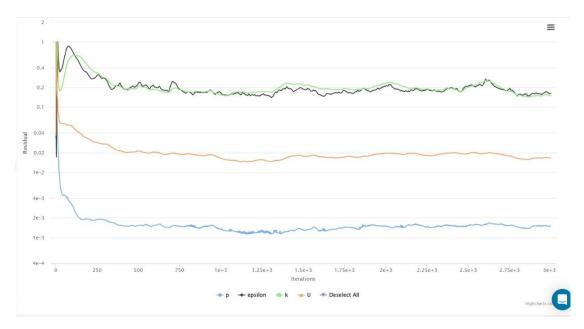


Picture 275 Static pressure 3D on the inducer and impeller of pump  $\mathrm{LO}_2$  Simscale program

Below are the results of the output values of the two pumps, the mesh characteristics, the efficiency and the power of the pumps.

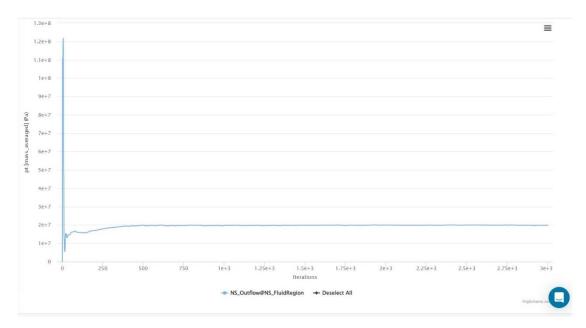


Picture 276 The turbulence model coefficients (pump LH<sub>2</sub>) Simscale program

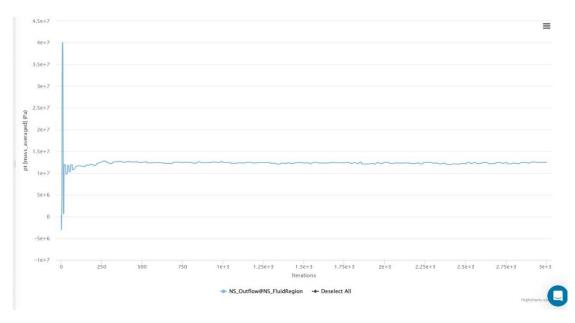


Picture 277 The turbulence model coefficients (pump LO<sub>2</sub>) Simscale program

Here we notice that the coefficients of the K-e turbulent model are balanced.



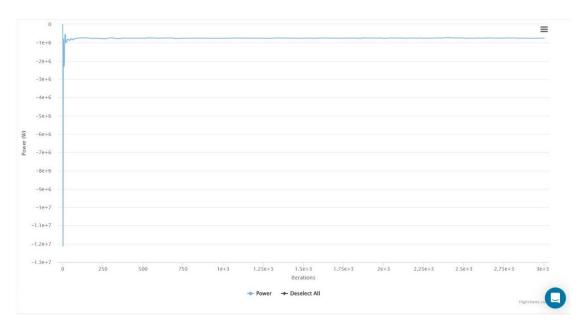
Picture 278 Average outlet pressure of pump  $LH_2$  Simscale program



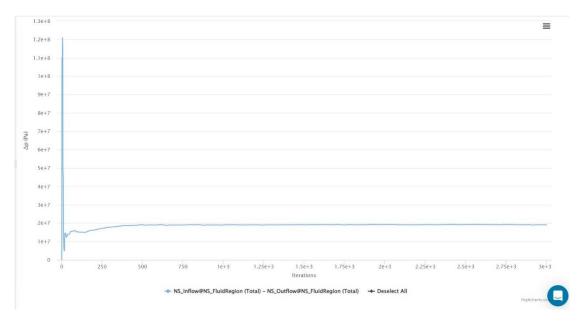
Picture 279 Average outlet pressure of pump  $LO_2$  Simscale program



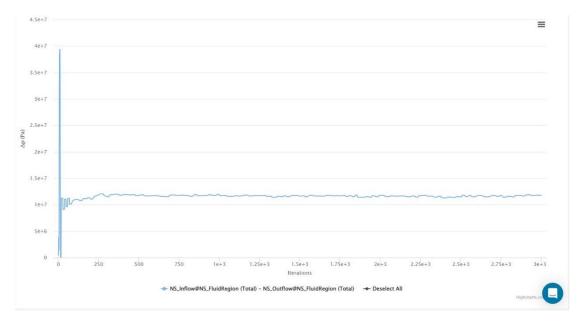
Picture 280 Input power of pump  $LH_2$  Simscale program



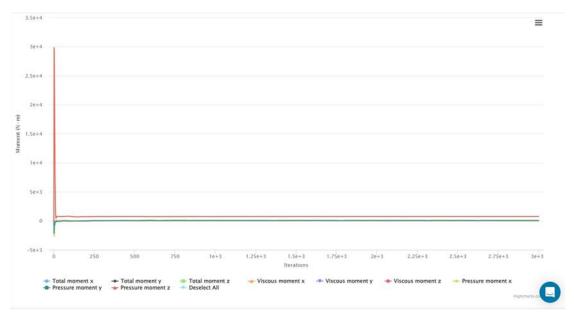
Picture 281 Input power of pump LO2 Simscale program



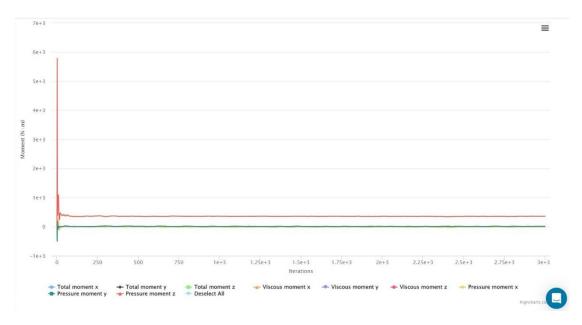
Picture 282 Total pressure difference of pump LH<sub>2</sub> Simscale program



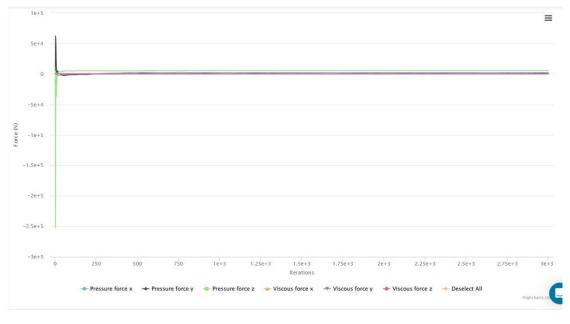
Picture 283 Total pressure difference of pump LO2 Simscale program



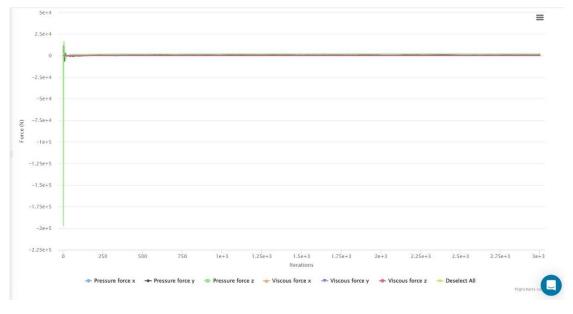
Picture 284 Torque of pump  $LH_2$  Simscale program



Picture 285 Torque of pump LO<sub>2</sub> Simscale program



Picture 286 Force of pump LH<sub>2</sub> Simscale program



Picture 287 Force of pump LO2 Simscale program

From the results it is found that the outlet pressure and the pump torque are above the design point. The tables below show the characteristics of the two pumps, the conditions under which they operated and the grid characteristics.

TABLE 1 PUMP TECHNICAL CHARACTERISTICS		
Size	Pump $LH_2$	Pump LO <sub>2</sub>
Inlet pressure (bar)	7	7
Outlet pressure total (bar)	198	124,6
Input Power (MW)	6,25	0,761
Mass flow (Kg/s)	20	64
Volume flow (m3/s)	0,27129	0,05223
Pressure totaldifference (bar)	191,5	117,6
Number of revolutions (rpm)	80000	20000
Hydraulic efficiency (%)	83	80,7

TABLE 2 CHARACTERISTICS OF MESH		
Size	Pump LH <sub>2</sub>	Pump LO <sub>2</sub>
Minimum (relative to CAD)	0,00001	0,00001
cell size		
Cell size on surface (relative	0,001	0,001
to CAD)		
Maximum (relative to CAD)	0,002	0,002
cell size		
Number of vertices	9726819	8158698
Number of cells	7398816	6221598
Number of volumes	5	5
Number of cell sets	5	5
Number of faces	25037810	21028755
Number of face sets	172	200

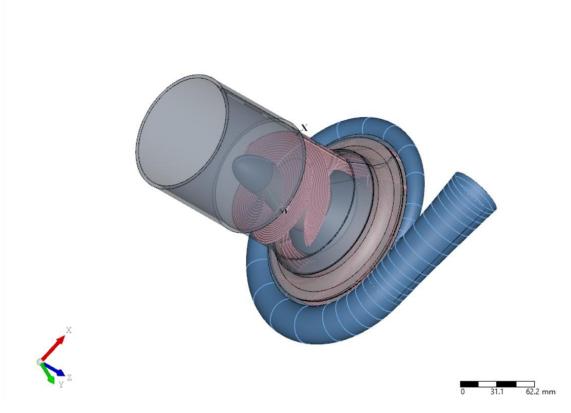
The hydraulic efficiency is very large and is very close to the corresponding efficiency curves with respect to the specific speed number nq.

Preferred construction materials are shown in the table below. They are two common materials used to make these pumps. Many examples are cited from NASA reports of the use of these materials.

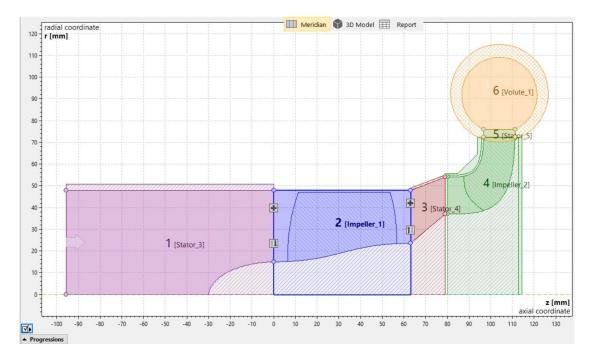
PUMP	MATERIAL
LH <sub>2</sub>	✓ Ti-5AL-2,5Sn
LO <sub>2</sub>	✓ Inconel-718

What remains to be designed with more stringent requirements is the material, the canopy and the disk at the back of the impeller in order to be tested in a stressful analysis. This part is not considered in this thesis.

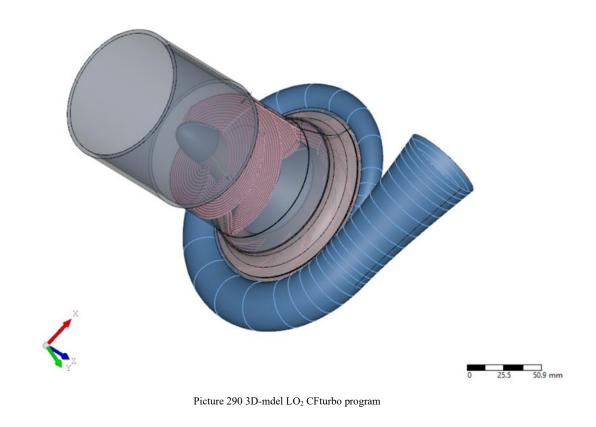
## **FINAL MODELS**

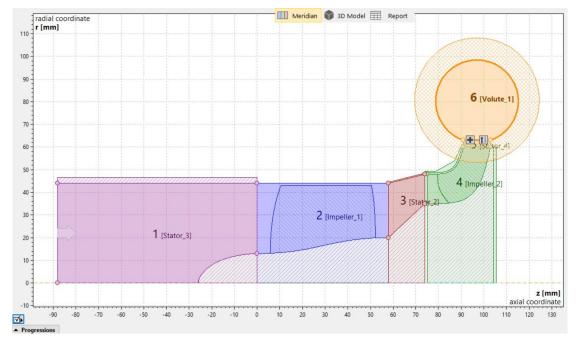


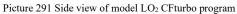




Picture 289 Side view of model LH2 CFturbo program







## **Unit 5 Observations and conclusions**

The design of the two pumps for liquid hydrogen and liquid oxygen was completed and successfully tested in a simple CFD simulation. If data were available on the cryogenic fluids at that temperature, a more accurate cavitation simulation could be made. However, there is a lot of room for improvement and corrections for the design of the pumps.

- Start designing the pumps with smaller mass flow. Thus, the hydraulic losses are reduced and the flow would be more controlled.
- Set the design point so that the inducer outlet and the impeller inlet intersect geometrically. By itself, there would be no need for the conical rotor with the conical stator which makes the construction more complicated, and brings hydraulic losses
- Design a finned diffuser between the inducer and the impeller so that the fluid that exits from the inducer, is the same as the fluid inlet of the impeller.
- Use different criteria for the design of the impeller such as those based on the inlet angle.
- Use the automatic model of the deviation angle of the inducer, rather than manually.
- Thin the inducer fin at the leading edge. In this way, the flow is smoother, hydraulic losses are reduced and cavitation possibilities are reduced.

In general, the design of a turbopump for both normal and cryogenic rocket engines is very difficult. However, software has been developed with which engineers cannot only design the components very quickly but they can also redesign the entire pump-turbine system if the desired criteria are not met. Nevertheless, the prediction of unstable cavitation has not been fully understood by engineers. Also, there is no universal method to avoid cavitation during design. Therefore, cavitation remains a big problem for engineers, which they try to tackle with advanced design software, with CFD simulations, with real experiments and with their experience.

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