Rocket Engines: Turbomachinery

Hans Mårtensson
Sonny Andersson
Stefan Trollheden
Staffan Brodin
VOLVO Aero Corporation
S-461 81 TROLLHÄTTAN
Sweden

1. INTRODUCTION

A turbopump in a rocket engine consists of a pump that delivers fuel or oxidizer to the thrust chamber where the propellants are brought to react and increase in temperature. Since the combustion process takes place under constant pressure, the chamber pressure is the net result of the turbopump system. Turbomachinery for liquid rocket propulsion shares many of the design features and challenges found in gas turbines. To an even greater extent than in jet-engines used for aircraft the emphasis is on delivering very high power in a small machine. Pumps and turbines are classical subjects of engineering and are in wide spread use in many areas. The constraints and requirements of the application is what make the design of turbopumps for liquid rocket engines unique. In this lecture the pumps will be discussed mostly in the context of load on the turbine. The main reason for doing so lies in the background of the authors, working at a company that designs and manufactures turbines for space propulsion. We shall look into what governs the design of turbines for liquid rocket engines. This is done by example following a simplified design loop, discussing as we go.

Many important steps were taken from after the war and to the 70's in the design of turbopumps for rocket engines. A historical review of the development is very enlightening and highly recommended for interested engineers [1]. Following a period of very active development NASA issued a collection of documents that summarizes design experience in terms of rules and criteria [4]-[7]. These are still very valid in terms of providing guide lines to what works and what does not. Also very important is that they provide insight to solutions adopted for several machines and by several different companies. Much of what we will describe in terms of design will be based on these. This provides a coherent basis for the lecture, and does not infringe on proprietary information. In engineering practice each company that develops turbines has in-house practices and experience that strongly affects the final design. Modern computational tools have provided for great changes to engineering work and advancement of machines since the 70's. Much of this is used in detailed design and replaces to some extent testing on component level, and has become indispensable in modern engineering.

New materials and computational tools allow the designer to improve the performance of the machine. With improved tools for design the machine becomes more optimal, which is good. At the same time as the designer approaches the limits of what is physically possible the numbers of failure modes multiply and the accuracy required in verification of the analysis increases. It has always been argued that computational tools such as CFD and FEM reduce development time and cost. It appears more that the actual effect is that as optimization becomes possible with better tools, the machines get better but the development time and cost is equal, at the best. This is maybe not to surprising after all since new designs are often made in order to deliver better performance. At the end of the day this is still a balance act of risk management, customer input, innovation, economy and hard work.

Mårtensson, H.; Andersson, S.; Trollheden, S.; Brodin, S. (2008) Rocket Engines: Turbomachinery. In *Advances on Propulsion Technology for High-Speed Aircraft* (pp. 5-1 – 5-28). Educational Notes RTO-EN-AVT-150, Paper 5. Neuilly-sur-Seine, France: RTO. Available from: http://www.rto.nato.int.

2. CYCLE SELECTION EFFECTS ON TURBOPUMP CHARACTERISTICS

Selection of the engine cycle determines the characteristics of the turbine to a large extent. The cycle a choice made on engine level, balancing complexity with performance for some flight and load. On the development side this is made quite some time before the detailed design of turbomachinery is made. In order to optimize the system the engine designer will need to know what to expect in terms of performance from the different components that make up the engine. The engine designer will want to know in a variety of configurations and cycles what can be achieved in terms of cost, weight and efficiency potentials.

Trying to describe how the cycle selection affects the turbomachinery we will first look at the main performance numbers describing the propulsion system as a hole. This gives some appreciation of what the controlling parameters of the engine means to the turbopump. Since we want to discuss the turbomachinery aspects simplifications are taken as necessary. First of all, the measure of the rocket engine performance can be expressed in the specific impulse Isp:

$$Isp = \frac{F_n}{g \cdot \dot{m}} \qquad \begin{array}{c} F_n \text{ is the thrust} \\ g \text{ is the gravitation constant} \\ \dot{m} \text{ is propellant mass flow} \end{array} \qquad (1)$$

For our purposes of studying turbomachinery we will accept the notion that the higher the Isp the better the performance is. The first we can notice is that the higher the chamber pressure is the higher the thrust Fn will be. This of course immediately goes back to the pump exit pressure which must be higher than the chamber pressure. The second component that can be discussed is the mass flow. If nothing else is changed we would expect the thrust to be proportional to the mass flow. This is easiest seen in the trivial example of putting 2 engines side by side resulting in twice the thrust for twice the mass flow and Isp equal.

The choice of cycle can be described in 3 main categories, which have great implications for the turbomachines. The main cycles used for larger modern liquid rocket engines are, as shown I figure 1, gas generator, expander and stage combustion.

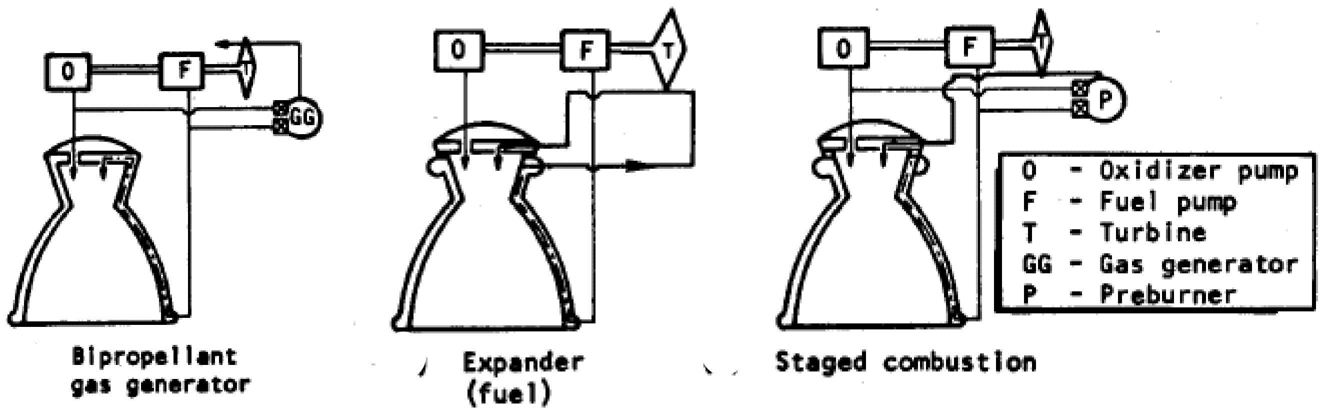


Figure 1 Principle schematics of major liquid bi-propellant rocket engine cycles from NASA SP-8107

Out of these 3 cycles the gas generator cycle is the easiest to control and least sensitive since it works against atmospheric or low pressure not very dependent on the turbine itself. The turbine for this cycle is a low flow high pressure ratio machine. The discharge pressure of the pump is slightly above chamber pressure. For the expander cycle the heat source is the nozzle cooling, which means that the amount of heat is limited by heat transfer processes. All the flow is pass through the turbine, and the available chamber pressure depends directly on the on how much pumping power that can be delivered given the heat input. This makes it sensitive to the turbine efficiency. The discharge pressure is high compared to the chamber pressure, enough to allow expansion through the turbine before being injected to the chamber. A staged combustion cycle allows for extremely high chamber pressures. As a consequence the discharge pressure from the pumps will be needed to be extremely high and multiple pumps are used in some cases. From NASA SP-8107 a summary graph is given in figure 2 for reasonable choices of cycle for different chamber pressures along with the resulting pump discharge pressure.

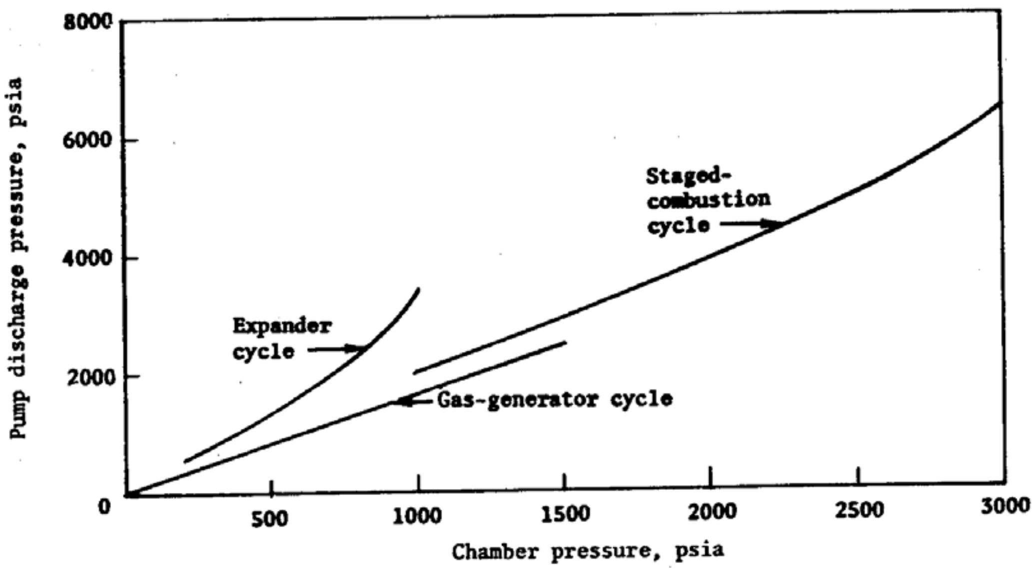


Figure 2 Relation between chamber pressure and pump discharge pressure for the cycles (1000 psi =70 bar)

The mass flows of fuel and oxidizer respectively will vary with the choice of propellants. The most important property for the pump is the density of the fuel. We can compare RP-1 with liquid hydrogen LH2 in this respect. For a 100 bar discharge pressure the pump head is for the propellants summarized in table 1. The lower the density therefore the higher the pump power becomes. The differences are large, for the same mass flow of LH2 the pump will take a factor of 10 times the power.

Table 1 Effect of density on pump head

	Density * [kg/m3]	Pump head at 100 bar [m]
LH2	75	13600
RP-1 (Kerosene)	810	1260
LO2	1200	849

* Approximate numbers. Density varies with temperature and pressure

Notably the pump head for liquid hydrogen is 13.6 km, a huge number that if applied as a fountain would send the water column up above the Mount Everest, and the most extreme discharge pressure would go up to above 600 bar exit pressure from a pump. Other effects to consider are the lubricating and cooling properties for bearings. The burner gases are of very different quality, they are going to be either fuel rich or oxygen rich. For LH2/LO2 the choice is always hydrogen rich, and LH2 is used for cooling with excellent properties. For RP-1/LO2 there is some variation. For Oxygen rich gases means that the partial pressure of oxygen is extremely high with associated fire risk and risk for deterioration of structural surfaces by oxidation. In order to reach 1000K on the rich side an O/F=13 (or equivalence ratio 26) would be needed. Such very rich mixtures may cause problems with residue as the carbon compounds meet surfaces. Using kerosene for cooling can also easily cause coking.

For the turbopump assembly some different arrangements have been used. Gears have been in use early on in the development of American engines. These have been replaced largely by pumps driven directly by the turbine. We shall therefore not discuss geared configurations. There may also be one turbine driving fuel and oxidizer pump on a single shaft. The choice is made as a trade between cost/complexity and efficiency. On the pump side both axial and radial pumps are used in different configurations. The majority would appear to be radial pumps with an inducer fitted to avoid cavitation. An example of an axial pump for LH2 is shown in figure 3 from the J2 engineLH2 fuel pump. Further down examples of centrifugal pumps are shown.

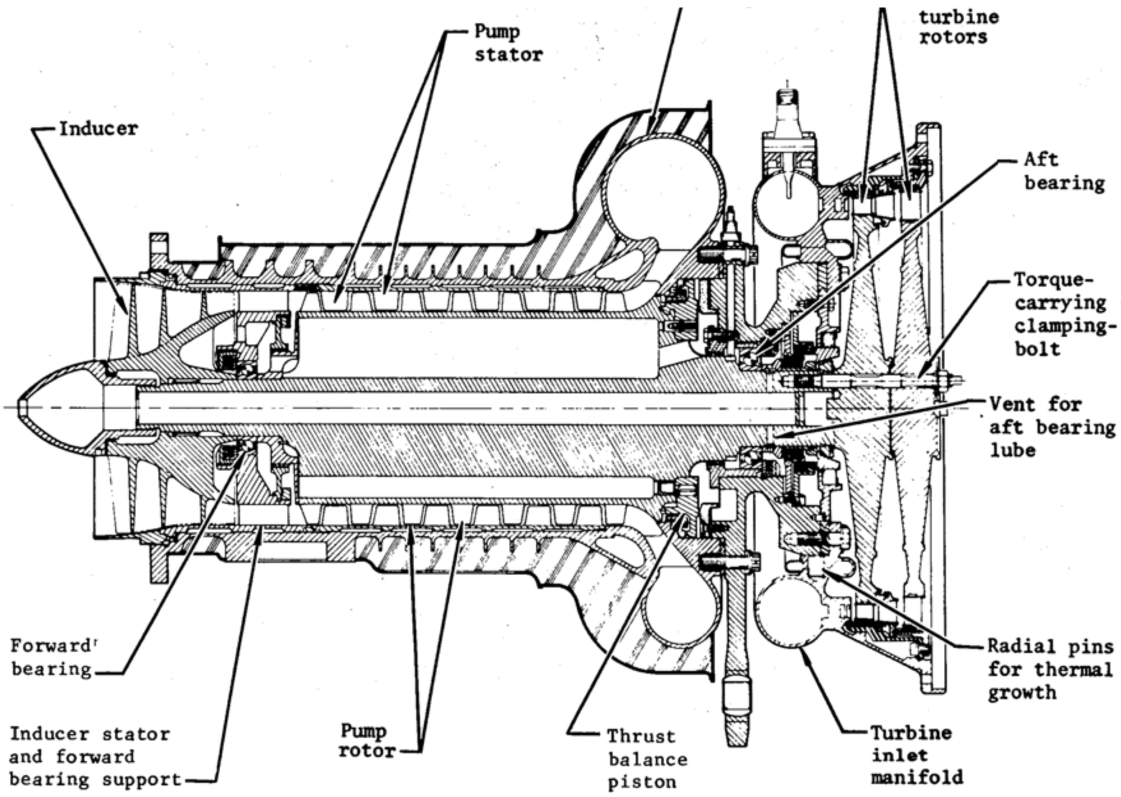
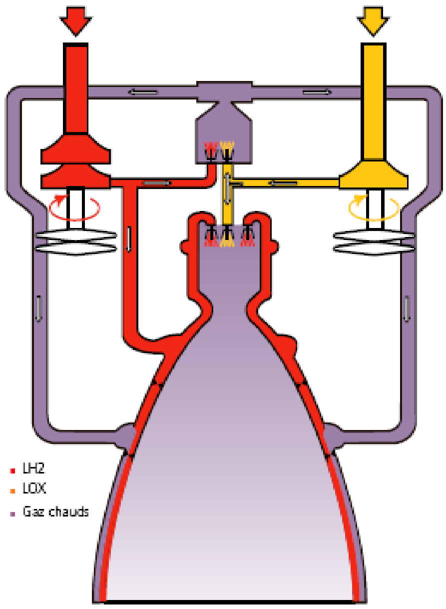


Figure 3 J2 axial fuel turbopump assembly from [3]

2.1 Gas generator

In a gas generator cycle the turbine exhaust is de-coupled from the main flow of propellant thus giving an available pressure ratio that is chamber pressure to atmosphere.

The gas is produced by burning some of the fuel/oxidizer. The first goal is to limit the loss of Isp inherent in expanding some of the gas to lower pressures than the chamber pressure, the mass flow used should be limited. Increasing the inlet temperature beyond ~1000K would mean that the vanes and blades must be cooled. In terms of turbomachinery efficiency it is difficult to use a pressure ratio in the order of 100 efficiently over the single turbine; therefore the gas is exhausted to atmosphere through separate nozzles contributing to the thrust, or possibly introduced in the main nozzle. The latter principle is used on the Vulcain 2 engine, shown in figure 4. This arrangement uses the turbine exhaust gas both to generate thrust and to help in the cooling of nozzle wall by using the turbine exhaust as a film. Weight is then gained by avoiding a large surface that need fuel cooling near the hot flame.



VULCAIN®2

Le moteur cryotechnique Vulcain®2 est dérivé du moteur Vulcain® de l'étage principal d'Ariane 5. Il a été développé dans le cadre du programme Ariane 5 Evolution.

Lancé en octobre 1995, ce programme a permis d'augmenter les performances du lanceur. Ainsi sur une Ariane 5 équipée de l'étage supérieur ESC-A, le moteur Vulcain®2 permet une augmentation de la charge utile en orbite de transfert géostationnaire de 1 150 kg par rapport au moteur Vulcain® qu'il remplace. Snecma est maître d'oeuvre du programme Vulcain®2 et pilote dans ce cadre une coopération d'industriels européens comprenant notamment Techspace Aero, société du groupe SAFRAN, EADS ST GmbH, Avio et Volvo Aero Corporation.

Vulcain®2 a été optimisé en améliorant, en particulier, trois paramètres :
• le rapport de mélange oxygène liquide/hydrogène liquide est passé de 5,2 à 6,1

CARACTÉRISTIQUES

• Poussée dans le vide (kN)	1 340
• Impulsion spécifique (s)	431
• Pression de combustion (bar)	115
• Ergols	LOX- LH2
• Débit d'ergols (kg/s)	320
• Rapport de mélange	6,10
• Vitesse de rotation TP (tr/min)	LOX : 12 300 - LH2 : 35 800
• Puissance turbines (kW)	LOX : 5 - LH2 : 14
• Hauteur (m)	3,45
• Diamètre sortie tuyère (m)	2,10
• Masse du moteur (kg)	2 100

Figure 4 Schematic and main data of Vulcain2 engine developed by SNECMA and used for the European Ariane5 heavy launcher (Source: fichier technique available at www.snecma.com)

The start transients need to be very fast in order not to consume propellants on the ground, in this case a pyrotechnic device is used. This means that a hot gush at high pressure hits the turbine after the start "explosion", the turbine acceleration is completed in seconds giving an extremely rapid load transient. This start transient is a major challenge to the mechanical design, as it leads to extremely severe thermal gradients and thereby stresses in the material. The stop transients can be as severe a load case on the engine. This may appear strange for a dispensable engine, but the engine is test fired at least once before flight. The hot engine is flushed with cold gases when it shuts down which produces a shock cooling.

In order to minimize the mass flow consumed the turbine and pump should operate efficiently. In oreder to achieve this the blade speed and flow velocities should generally be high. The blade speed is limited by mechanical constraints to the order of 500-600 m/s. For the Oxygen pump the shaft speed may limit the practical mean speed to values lower than this. In the Vulcain 2 engine separate pumps are used for LH2/LOX both with radial pumps. A cross section of the LOX pump in figure 5 shows the arrangement using an inducer on the pump side and an overhung 2-stage axial turbine on the turbine side. The overhung turbine arrangement is generally preferred if it is possible. Outboard bearings cost weight and but allows higher rotor stiffness, and higher rotor speed which is necessary for the LH2 turbopump. Such an arrangement is used for instance on the LH2 turbine for the Vulcain engine.

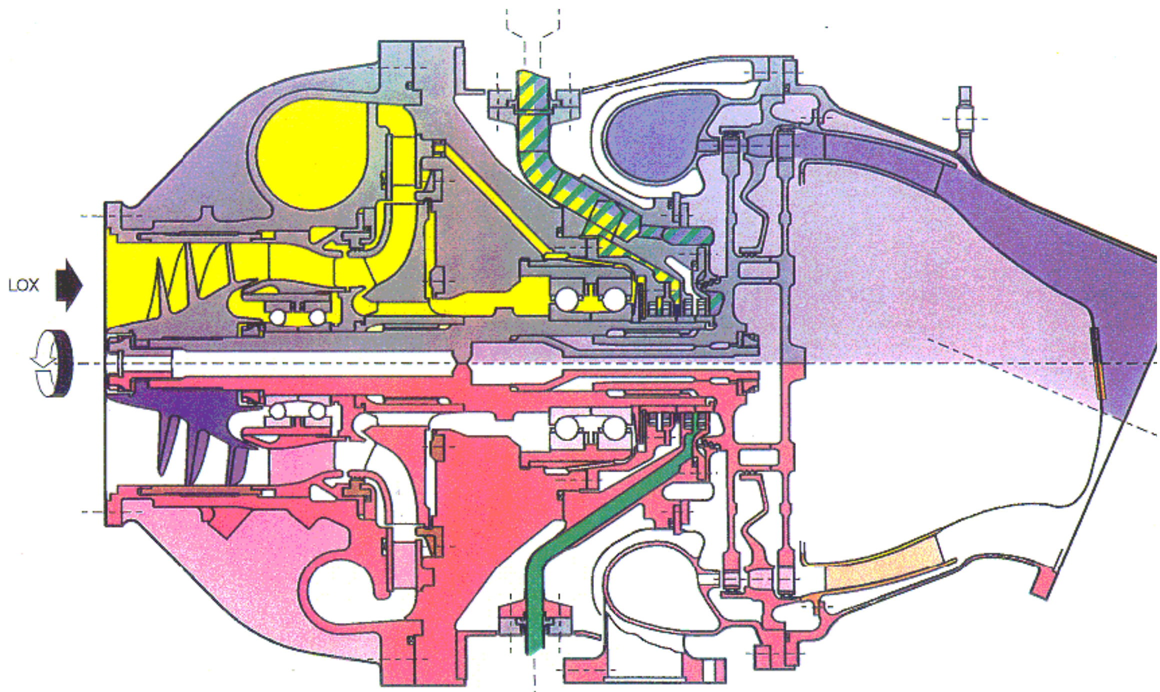
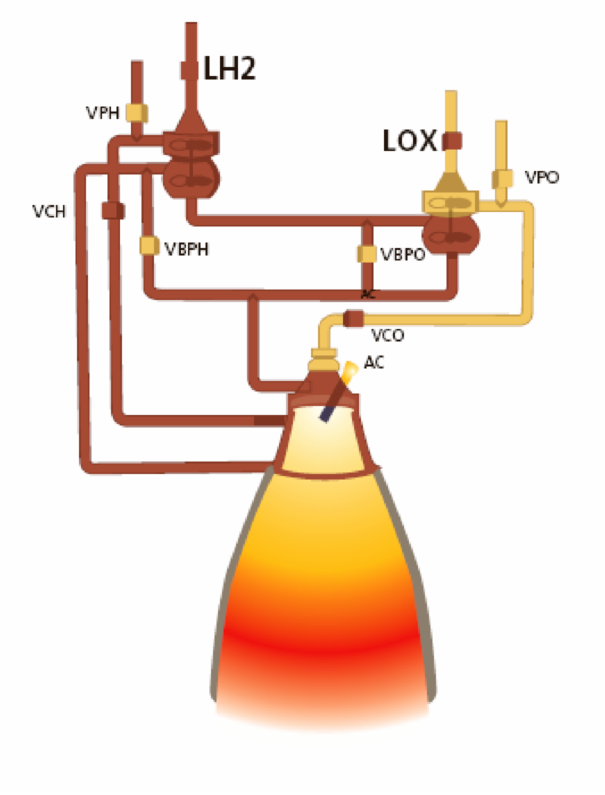


Figure 5 Cross section of Lox pump developed by Avio togheter with VOLVO Aero for the Vulcain 2 (from [9].)

2.2 Expander cycle

The attainable chamber pressure is immediately affected by the turbomachinery efficiency since the available heating power is limited by the heat exchanger process on the thrust chamber. On the upside for this cycle is the fact that a separate combustor is not needed in this cycle. The efficiency of the turbomachine is critical to operation and the engine system integrator has to work very closely with the component designers. The engine can start from tank overpressure, especially if it is an upper stage engine with vacuum counter pressure. The start produces a very cold dip before the heating of the fuel starts following ignition in the chamber very cold dip on the start transient. The initial LH2 flow is at ~30K from which temperatures rise fast. A modern example of an engine using the expander cycle is the SNECMA VINCI upper stage engine, shown in figure 6.



CARACTÉRISTIQUES

• Type	Cycle expander
• Poussée dans le vide (kN)	180
• Impulsion spécifique (s)	465
• Pression de combustion (bar)	60
• Rapport de section	240
• Ergol	LOX - LH2
• Débit d'ergols (kg/s)	LOX : 33,70 - LH2 : 5,80
• Rapport de mélange	5,80
• Vitesse de rotation TP (tr/min)	LOX : 18 000 - LH2 : 90 000
• Puissance turbines (kW)	LOX : 350 - LH2 : 2 800
• Hauteur (m)	4,20
• Diamètre sortie tuyère (m)	2,20

Figure 6 Characteristics of the VINCI upper stage expander cycle engine by SNECMA (Source: fichier technique available at www.snecma.com)

In the expander cycle all the fuel passes the thrust chamber in order to pick up heat. LH2 is the fuel most easily adapted to the cycle. It has a high heat capacity and does not risk to coke or contaminate the pipes. Oxygen is mostly avoided as heat carrier as it is aggressive and easily causes fires. Hydrocarbon fuels have a coking problem and works less well as heat carriers that the hydrogen. Figure 7 shows a cros-section through a VINCI LH2 pump for an expander cycle.

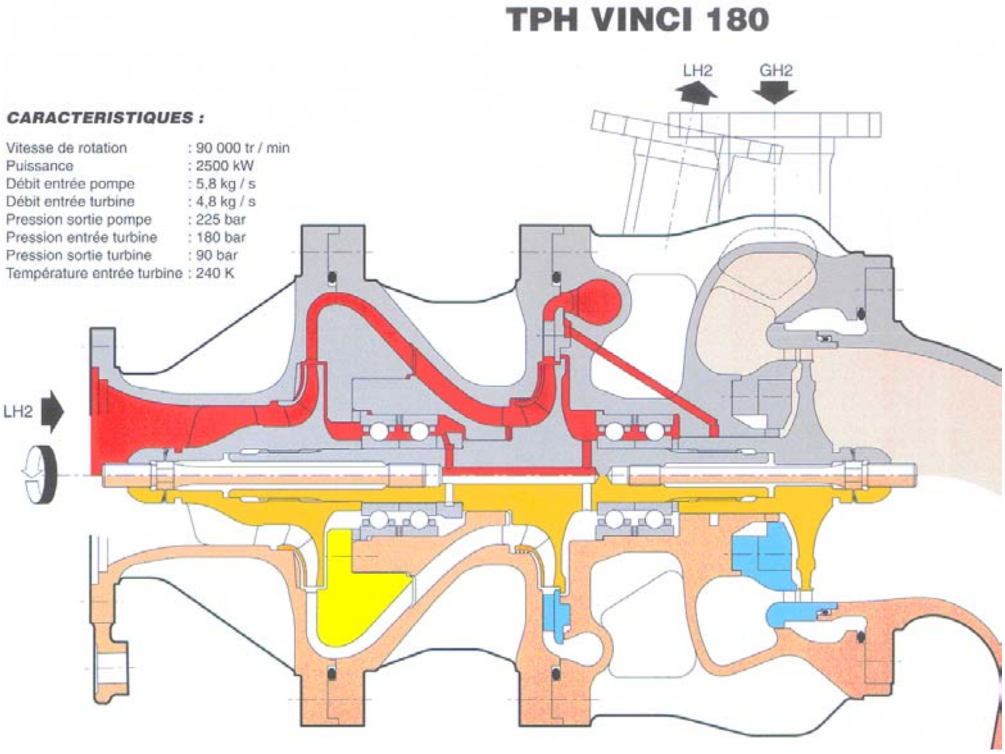


Figure 7 showing the VINCI LH2 turbopump developed by SNECMA with VOLVO aero as responsible for the turbine [13]

2.3 Staged combustion

The most typical feature is the extremely high pump discharge pressure, the SSME (Space Shuttle Main Engine) at 470 bar for a chamber pressure of 223 bar and Russian RD-170 has discharge pressures above 600 bar for a chamber pressure of 250 bar. The SSME has dual turbopumps making the system more complex in number of machines and sensitive to the success of the design of components. Under such extreme pressures the mechanical integrity of the machines becomes the overwhelming issue. This is shown in figure 8 where both low pressure and high pressure turbopumps can be seen.

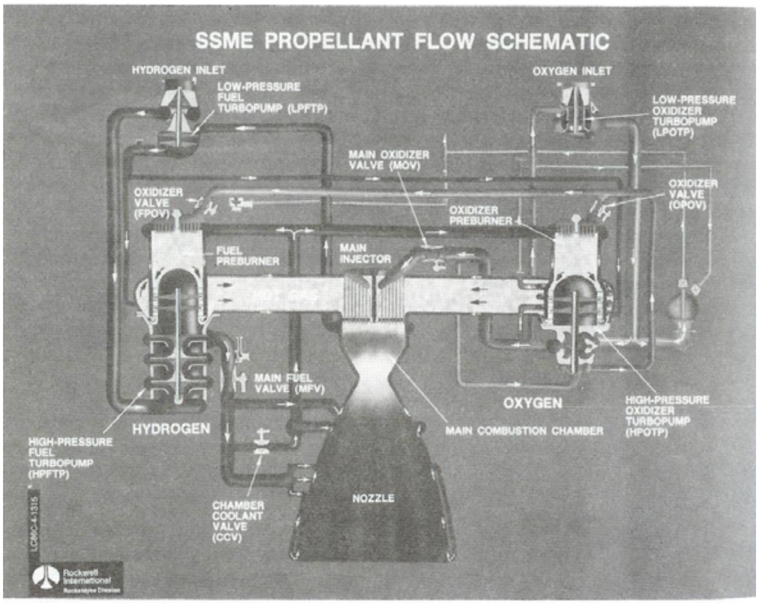


Figure 8 SSME chart showing engine flows

The turbine of the RD-170 feeds all turbopumps, with a total shaft power of and 192 MW over a single stage, the inlet pressure in 519 bar and the flow 2400 kg/s [2]. The pressure ratio over the turbine is 1.92 and inlet temperature 770K. I the engine RP-1/LOX is used with the choice of oxygen rich gas in the turbine. For this engine development a remarkable note is that over 200 development engines was used for testing.

3. EXAMPLE DESIGN OF TURBOPUMP

The actual development of the turbine starts from the basic architecture and specification of the engine performance parameters. A specification in this context is a set of data and rules agreed with the engine designer. This will set goals on performance, weight and cost. The development phase can span over a decade starting with a specification and ending with a functional product. In this process progressively more people become involved and more money is spent each week. At the same time as time progresses there will be less and less things that can be changed since more details become frozen.

3.1 Preliminary Design of pump for an imaginary gas generator cycle

In order to provide some understanding of how design features relate we will go through a sample design loop using relatively simple tools. The example will be a design for a gas generator cycle using LH2 and LO2. The basic cycle parameters for the cycle that are of relevance to the turbomachinery are picked arbitrarily:

Propellant mass flow rate 300 kg/s at O/F=6, Pchamber 100 bar, Pinlet 1 bar.

Table 2 Summary of Pump flow data for example turbopump

Pinlet	Pexit LH2	Massflow LH2	Pexit LO2	Mass flow LO2
1 bar	132 bar	42.9 kg/s	120	257.1 kg/s

In order to start the design we begin with the sizing of the pump. The efficiency potential of the pump will be maximized for the engine and the sizes and speed are selected. These numbers hen form the primary requirements on the turbine. As the turbine is designed it could happen that it is difficult to find a good solution which could force us back one step and re-design the pump. The main tools for selecting the pump size are 2 non-dimensional numbers, classical in turbomachinery:

Specific speed
$$n_s = \frac{\Omega\sqrt{Q_{in}}}{(gH)^{3/4}}$$

Specific diameter
$$d_s = \frac{D(gH)^{1/4}}{\sqrt{Q_{in}}}$$

The red circle in figure 9 marks the optimal choice of specific speed and specific diameter. Commonly American literature does not use SI-units and comes out differently in terms of size. Here SI-units will be used, and conversions made when using diagrams or numbers quoted from American literature.

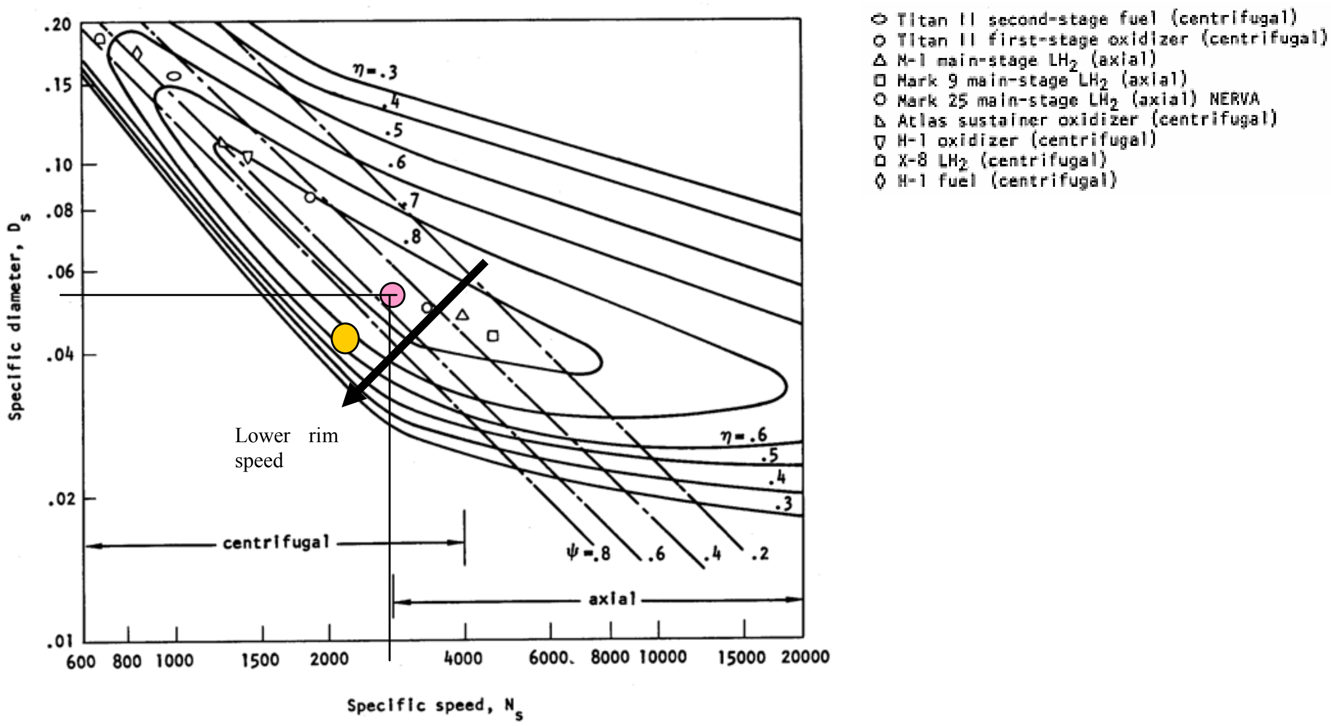


Figure 9 ns-ds diagram with from NASA SP-8109

The selected point at maximum efficiency gives (converted to SI units):

Specific diameter $d_s = 0.055 * 50.5 = 2.77$

Specific speed $n_s = 3000 * 3.66 * 10^{-4} = 1.1$

The design point for a single stage pump the diameter and shaft speed for the machines.

The LO2 pump

The LO2 pump must be chosen with a relatively low tip speed but gets a rather large diameter. The pumped power is about 1/3 of the LH2 pump, and hence it will be less critical to performance of the cycle. In terms of a system trade the weight could be brought down by decreasing the tip diameter and increasing the shaft speed. The weight range of the LO2 pump in this class is 150-200kg. Table 3 compiles the data for the pumps.

Table 3 Solutions for pumps using design charts

	Shaft speed N [RPM]	Tip diameter D [mm]	Tip speed Utip [m/s]	Efficiency potential η [-]	Power needed P [kW]
LO2	22980	132	158	0.8	3500
LH2 1-stage	122100	104	665	0.8	10200
LH2 1-stage reduced ns,ds	106140	83	462	0.7	11700
LH2 2-stage	72600	124	469	0.8	10200

The LH2 pump

Trying first with a single stage LH2 pump gives and design that would have a high rim speed, 665 m/s, SP-8107 considers pump rim speeds (unshrouded) above 800 m/s, but here we will take a careful approach. We also have a low inlet pressure that leads to cavitation risks. It is then reasonable to consider this too high and look for other solutions. The options that we have are to reduce rim speed by lowering ds and/or ns. The choice indicated by the yellow dot in figure 9 gives a rim speed below 500 m/s, but also a lower efficiency. In the diagram the loss is as much as 10 % a rather large number that must be compensated by turbine shaft power. Another option is to use a second stage and stay on the optimal performance point, table 3 includes those options.

This example of pump sizing serves to show the effects of constraints and to provide operating conditions that will be used for sizing a turbine. Another constraint is formed by the risk for cavitation in the pump. The pressure ahead of the pump must be large enough so that the lowest pressure on the suction side of the impeller does not fall below a critical value for cavitation. This is usually measured as NPSH (Net Pressure Suction Head) and will often affect the allowable limits of the machine. In [5] these limits are detailed and compared in several ways. Other constraints that may limit the speed is, rotor dynamics and the bearing inner race speed may become limiting as well as the turbine blade stresses in a later stage of the design.

At this point, however, we will be happy for now with the sizing of the pumps so that we can go on with the turbine.

3.2 Turbine performance

The most important performance data can be thought of as a budget for the turbine designer. In order to start working out the details of the turbine these must be known. We shall try to give an example that is non-unique but reasonable way to work through the choices that can be made. The start is made from the basic set of data given on one aerodynamic design point (ADP). When the engine cycle and architecture has been selected and laid out the pump load is known and a suitable turbine can be designed to meet those requirements. The pump size and reduced speed depends on the working media and delivery pressure. The speed on the shaft will be considered governed by the pump, until other limitations must be accepted.

3.3 Turbine duty and assessment of efficiency potential

We are going to use 1000 K as the inlet temperature to the turbine, which can marginally be managed with super alloys with no cooling. By comparison modern aeroengines are at 1900K, using advanced cooling technology. Introducing cooling to the blades would increase complexity of the system, in particular for the relatively small blades and also require bleeds from the pump with increased plumbing as a result. On the oxygen side cooling must be drawn from somewhere else since the turbine is mostly fuel rich. We have a temperature and pressure at the inlet to work with, after allowances for pipe and valve losses and burner. The exit pressure would be selected to be something sufficiently high to allow safe evacuation of the hot exhaust from the aft of the rocket. The higher the exit pressure the more thrust can be generated; on the other hand more gas will be consumed. This trade must be done at engine level but is probably fixed when the turbine shall be designed. The spouting velocity $C0 = \sqrt{2C_p\left(T_{00} - Tis\right)} = \sqrt{2C_pT_{00}\left(1 - \frac{P_{out}}{P_{00}}^{\frac{\gamma-1}{\gamma}}\right)}$ =2655m/s gives some indication of what we can expect to find in terms of the velocity ratio and the possible turbines that would go with it. For the LH2 turbine we can expect the following: More than 500 m/s in Um will not be allowed for stress reasons. Um/C0<0.2 results and we can see in diagram , figure 10, that for a single stage machine an impulse design could give 45% efficiency, in diagram figure 10 that a 2 stage design could improve this to 55% 3 stage will probably be too heavy on the rotor dynamics and is not admissible. For the LOX we know that the pump is slow, if we expect to end up at Um=300 m/s then we have UM/C=0.12. This gives us 38% for the singe stage and 45% for the 2-stage machine. Cost constraints in the wants us to save on hardware so we accept a 38 % efficiency in the spec in order to make allowances for a 1 stage machine.

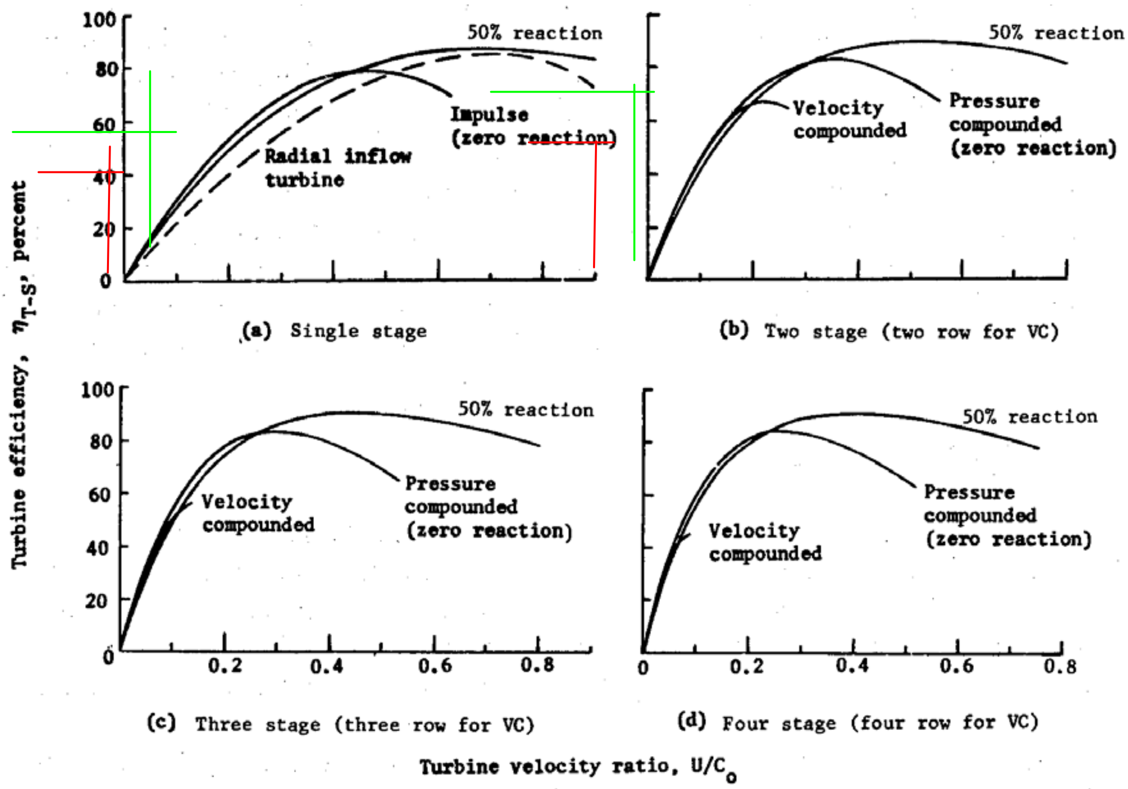


Figure 10 Efficiency potentials for supersonic turbines, red LO2 green LH2 points

As is seen from table 4 the mass flow consumption will still be twice as high for the LH2 turbine, which motivates the choice of saving cost on picking a single stage machine. 8 kg/s is below 3% of the flow to the chamber. Table 4 data will now form the specification that the turbine design will be set to match.

Table 4 Given data to match for the turbine ADP

	Ptin [bar]	Ttin [K]	Shaft speed [RPM]	Power [MW]	Pout [bar]	Massflow* [kg/s]	Efficiency* [-]
	100	1000	72600	10.2	10	5.27	55
	100	1000	22980	3.5	10	2.6	38

* Overdetermined less massflow is “always” better, as well as higher efficiency

This set is over determined in the sense that power can be computed given the other parameters. Efficiency could however be taken as a minimum requirement, the same way as the mass flow could be taken to be a maximum requirement. In a gas generator cycle the engine designer will probably be able to improve his engine if these parameters are better than budget. A temperature of 1000K is reached by burning at an O/F of 1.02, which gives values of γ=1.357 and a heat capacity Cp=7757 that will be used as constants through the turbine.

3.4 Selection of diameter and blade height

The first steps in the design are to fix some of the more general parameters for the turbine. The turbine gas paths mean diameter will be the first to be selected. This is done in more or less the same way as for the pump. Figure 10 gave some indications on how stages can be selected for a supersonic design. One of the most general design charts is given in figure 11 from [3]. Optimization of turbine performance in general is described in classical literature, and we will not go through all the possibilities given there. For each product there are unique constraints that affect the design. Applying these will force the designer away from the optimal selection of specific speed and specific diameter in order to satisfy load or mechanical requirements. The goal will be to find the best design considering a combination of weight, cost, efficiency and risk. The turbine parameters are normally given using exit parameters and this is the way they are used in our diagrams, so we will follow that way of presenting data.

Specific speed for the turbine

$$n_s = \frac{\Omega\sqrt{Q_{exit}}}{(gH)^{3/4}}$$

Specific diameter for a turbine

$$d_s = \frac{D(gH)^{1/4}}{\sqrt{Q_{exit}}}$$

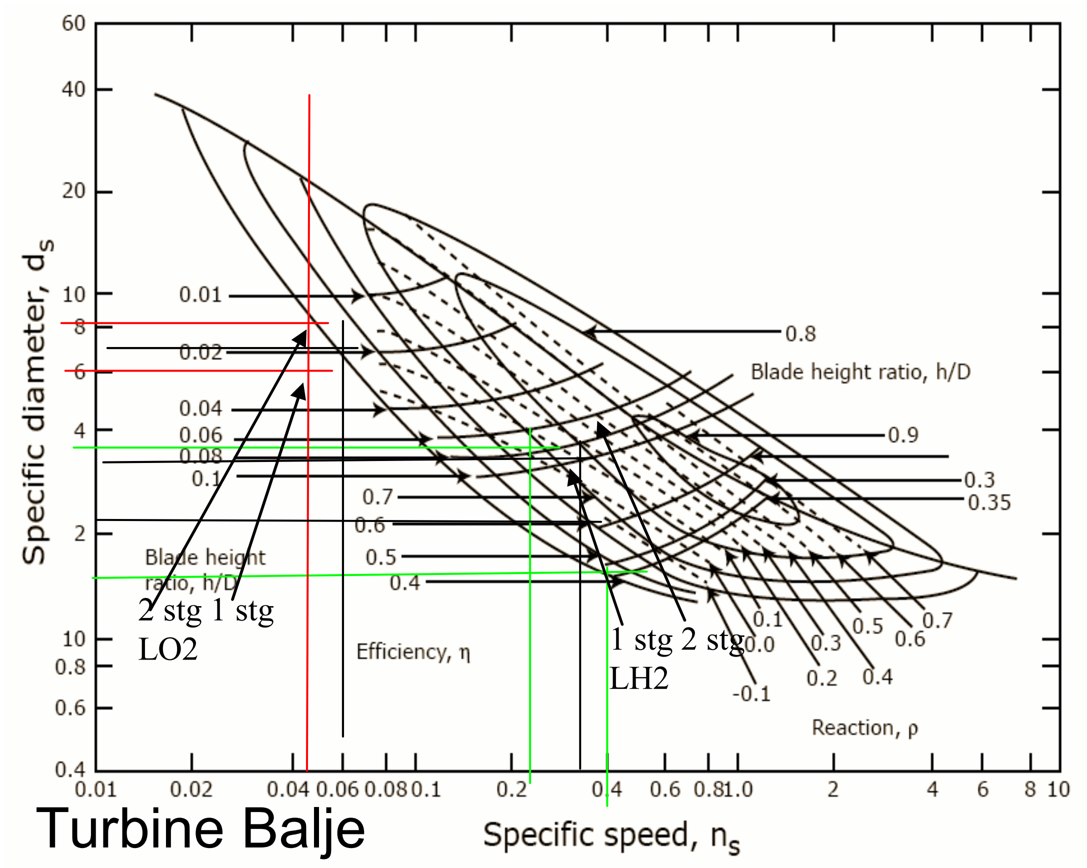


Figure 11 ns-ds diagram for turbines with selected points for the LO2 red and LH2 green options

We will now take the given shaft speed, which immediately gives the specific speed. The shaft speed was given by the pump, the isentropic enthalpy drop by inlet temperature and pressure ratio and the volume flow by exit density and mass flow.

The LO2 turbine

In figure 11 the red line mark the LO2 turbine choices for a 2 stage and a single stage turbine. In both cases ns=0.05. ds is 6.37 for the single stage and 8,37 for the 2-stage machine at a diameter Dm of 250 mm.

The diameter for the LO2 turbine was increased compared to the pump in order to increase the specific diameter. The blade height however for the LO2 turbine is becoming small, 1-2% of the diameter giving a blade that is only 5 mm high following the correlation in figure 11. The selection of blade height is probably the most open choice that the designer has to deal with. Reducing the diameter in order to increase the blade height would lower the efficiency again and make the effort useless. The blade height can allow for some blockage in the passage. In Sp-8107 a lower limit of 4mm is given for blade height. This would of course be dependent on whether a tip shroud is used or not, but can serve as a warning sign until it can be shown that the leakage can be handled. The height will have to be negotiated at a later stage in more complete loss analysis, where the aspect ratio of the blade can be calculated.

The LH2 turbine

The green lines in figure 11 shows the LH2 2-stage machine choices, and the black line the LH2 1 stage.

Using an intelligent guess that we will need all speed we can get, and assuming that the max we can have is a blade speed at mean diameter Um=500 m/s. The diameter can be picked out immediately at Dm =132mm. Optimal it would be ds=8, but this takes us to 1000 m/s which is hopeless. In terms of attainable efficiency the 1 stage is at 55% in figure 11. For the LH2 the first check shows that a stage count of 2 for the LH2 turbine in our example gives good improvements in efficiency >60 % and will be worth considering. A blade height of 10-13 mm is perfectly acceptable to start with.

3.5 Velocity or pressure compounding

Velocity compounding provides the extreme in obtaining power for a given pressure ratio and mass flow. However in our example we have chosen to fix the pressure ratio over the entire turbine. The reason for doing so practical, the engine designer will have assumed a pressure for designing exhaust of the turbine gas, whether in separate nozzles or if it is introduced in the main nozzle. For the turbine to be efficient at high pressure ratios the tip speed must be high. Here the propellant properties become important, for the LO2 pump the shaft speed will be low, so without a gear the specific speed will have to lower than desirable for turbine performance. The degree of reaction should be low at low isentropic velocity ratios, say lower than 0.25 the degree of reaction should be 0 and increasing to 50% as the isentropic velocity ratio goes to 0.45. This is here settled in the layout step where we look at the velocity triangles.

3.6 Layout

At this stage it is appropriate to introduce analysis using the notion of velocity triangles and flow areas. We will use analysis on the mean diameter of the turbine at design point, in order to get a better view of loss distributions and area development. More constraints can be added on structure angles etc. Also we work up to selecting chord and blade count. Some definitions of velocity triangles and notation are given in figure 12 below.

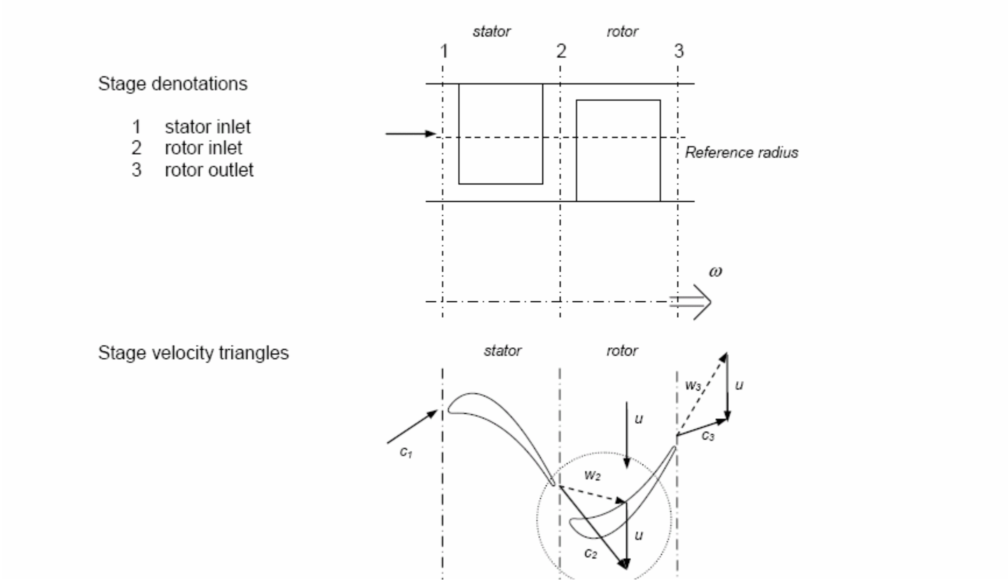


Figure 12 Turbine stage notations

Stress limits are assessed by using the AN2 parameter that relates to blade root mean stress. $AN^2 = \pi\left(r_O^2 - r_I^2\right)\cdot\Omega^2$. Figure 13 gives a NASA correlation for the parameter. Notably for INCO 718 for example is that the data falls sharply after 1000K. At that point cooling must be added or the material improved.

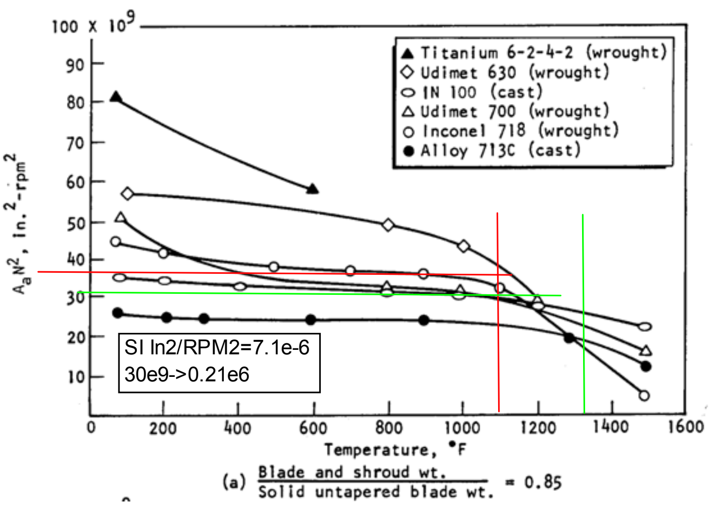


Figure 13 AN2 versus temperature stress limitation from [6]

Now the design points will be given on the mean line in order to show main data that helps to make final decisions on the turbine layout.

3.7 LH2 turbine single stage

For the LH2 turbine we first try to get a single stage design to work. Beginning Layout of a single stage turbine for the LH2 will require height substantial height increase through the rotor in order to avoid choking. This piece of information is not found in the ns-ds diagram, but could be vital as the AN2 parameter gets out of admissible range. This can not be seen immediately in the ns-ds diagram. Opening up the gas path means that the blade becomes much longer which first increases the root stress, the AN^2 parameter goes up above admissible values, in final version 0.635E6 which is above the admissible 0.21E6 (in SI units). In order to improve this situation shaft speed could be reduced, with some implications for the turbine efficiency.

Table 1

LH2	Station	S1 inlet	S1 outlet R1 inlet	S2 inlet R1 outlet
Total pressure (abs)	[bar]	100.0	80.0	16.4
Total pressure (rel)	[bar]		28.6	20.0
Static pressure	[bar]	94.1	12.1	12.5
Total temperature (abs)	[K]	1000	1000	818
Total temperature (rel)	[K]		863	863
Static temperature	[K]	984	608	763
Density	[kg/m^3]	4.69	0.97	0.81
Speed of sound	[m/s]	1651	1298	1453
Mach No. (abs)	-	0.30	1.90	0.64
Mach No. (rel)	-		1.19	0.86
Axial velocity	[m/s]	350	680	816
Tangential velocity (abs)	[m/s]	350	2370	-439
Tangential velocity (rel)	[m/s]		1868	-941
Flow angle (abs)	[deg]	45.0	74.0	-28.3
Flow angle (rel)	[deg]		70.0	-49.1

With these problems reported and that there seems to be little hope to improve the single stage LH2 is abandoned. It is desirable in early design to have some margins to work with later in the detailed design in order to make up for off design points or shifts in design point needed by the engine designer. At this point we are bouncing between choking and stress limits. Increasing the diameter implies reducing shaft peed, which can be done but preferably not.

LH2 turbine 2-stage

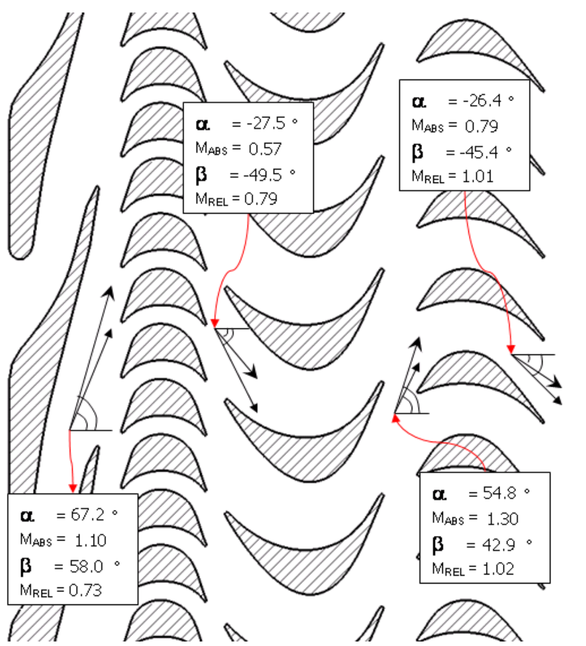
The 2 stage machine as seen in th ns-ds diagram has a higher efficiency potential to start with. The initial potential estimate has an efficiency of 55% giving a mass flow consumption of 5.27 kg/s. In the layout estimates we arrive at an efficiency of 63% which is comforting.

The overall data for this design is compiled and shows good margins. The exception is AN2 for stage 2 which is marginal. Some more work should be added here, possibly trading away some efficiency to gain on robustness. This can be done at pump level now since we have some margins on the turbine.

Station data for the 2 stage LH2

LH2	Station	S1 inlet	S1 outlet R1 inlet	S2 inlet R1 outlet	S2 outlet R2 inlet	S3 inlet R2 outlet
Total pressure (abs)	[bar]	100.0	88.0	44.4	39.1	19.1
Total pressure (rel)	[bar]		59.1	53.1	27.5	24.0
Static pressure	[bar]	94.1	41.8	35.7	14.4	12.7
Total temperature (abs)	[K]	1000	1000	875	875	750
Total temperature (rel)	[K]		917	917	797	0
Static temperature	[K]	984	822	826	672	674
Density	[kg/m^3]	4.69	2.49	2.12	1.05	0.92
Speed of sound	[m/s]	1651	1509	1512	1364	1366
Mach No. (abs)	-	0.30	1.10			
Mach No. (rel)	-		0.73	0.79	1.02	1.01
Axial velocity	[m/s]	350	643	772	1022	971
Tangential velocity (abs)	[m/s]	350	1530	-401	1450	-481
Tangential velocity (rel)	[m/s]		1028	-903	948	-983
Flow angle (abs)	[deg]	45.0	67.2	-27.5	54.8	-26.4
Flow angle (rel)	[deg]		58.0	-49.5	42.9	-45.4

The overall blade data comes out of the analysis so that now we have sufficient data to start drawing the turbine cross section. Figure 14 details a view over the turbine blades the way the profile may look. From left to right we have Stator1 (nozzle) Rotor1 stator 2 rotor2



LH2	Stage 1	Stage 2
Efficiency	65.0%	71.5%
Reaction (static pressures)	10%	7%
Zweifel,Stator	1.25	1.25
Pitch, Stator	0.022	0.0134
Chord, Stator	0.018	0.0232
Aspect ratio h/c,Stator	0.44	0.423
Slope angle	0.3	5.1
Zweifel,Rotor	1.25	1.25
Pitch, Rotor	0.007	0.0075
Chord, Rotor	0.009	0.0112
Aspect ratio h/c	0.841	1.056
Slope angle	-0.5	5.8
AN2	1.88E+05	3.12E+05

Figure 14 View of the example LH2 turbine with velocity triangles

LO2 turbine single stage

For the LO2 turbine the stress limitation is not a problem. Here the High Mach number is a problem for the efficiency. We find also here that a 2 stage machine would be beneficial to efficiency we shall make the best of a single stage machine, if a robust solution can be found.

Table LO2 single stage main data

LOX	Station	S1 inlet	S1 outlet R1 inlet	S2 inlet R1 outlet
Total pressure (abs)	[bar]	100.0	75.0	13.6
Total pressure (rel)	[bar]		25.7	19.2
Static pressure	[bar]	94.1	8.3	3.1
Total temperature (abs)	[K]	1000	1000	827
Total temperature (rel)	[K]		907	907
Static temperature	[K]	984	560	561
Density	[kg/m^3]	4.69	0.72	0.27
Speed of sound	[m/s]	1651	1245	1246
Mach No. (abs)	-	0.30	2.10	1.63
Mach No. (rel)	-		1.40	1.86
Axial velocity	[m/s]	350	610	701
Tangential velocity (abs)	[m/s]	350	2542	-1909
Tangential velocity (rel)	[m/s]		2241	-2210
Flow angle (abs)	[deg]	45.0	76.5	-69.8
Flow angle (rel)	[deg]		74.8	-72.4

LOX	Stage 1
Efficiency	42.2%
Reaction (static pressures)	6%
Zweifel,Stator	1.25
Pitch, Stator	0.029
Chord, Stator	0.007
Aspect ratio h/c,Stator	0.73
Slope angle	22.8
Zweifel,Rotor	1.25
Pitch, Rotor	0.013
Chord, Rotor	0.007
Aspect ratio h/c	1.009
Slope angle	33.9
AN2	5.68E+04

At this stage the flow angle should be kept below 75 degrees, because above this angle control of the flow area is difficult. We have a lot of margin t stress and rotor exit choke in this design so that it can be redesigned to lower the angle of the start. The degree of reaction would then go up slightly.

3.8 Summary of the designs

Now we have a design that has been defined to some level in one pass. There is definitely more work to be done on the example. If a better total design is to be generated there are at least 2 things that we may rework on the turbopump. NPSH criteria should be introduced in order to give realistic cavitation limit behaviour, and some thought should be given to tank pressure in this context. The second step is to check whether we could find better total solutions by increasing the diameters and reducing speed in order to find better efficiency on the turbine side. The efficiency on the pump side is relatively flat around the maximum, whereas the turbine has a larger gradient, so there is good hope to find a decent solution. This could also resolve some of the choking margins that we have on the turbine without increasing stresses.

The preliminary design choices will also affect the pump in terms of the axial thrust force on the shaft. The degree of reaction determines the pressure difference over a rotor disk. In our example we have discussed single stage impulse turbines which nominally do not have an axial thrust on the shaft, as well as weak reaction blades that have a significant pressure difference. In our example a 10-15% reaction gives up to a 2 kN axial thrust. This is important in terms of the system function and must be considered when designing seals as well as it may later affect the aerodynamic design. The total axial force on the bearing is not allowed to change sign, as the bearings will be designed to have the force going one way. This point must be analyzed carefully at off-design conditions.

3.9 Secondary flow system

Control of the thermals and leakage flows in the turbopump is an important design task. Provisions must be made for purging the cavities between rotors and stators. In gas turbines and aero engines this amounts to providing cooler compressor air to the turbine rotor stator cavities in sufficient amount to avoid the penetration of hot gases into the cavity. Also coolant for the turbine blade and shroud are provided for in this way. In the LH2 turbo pump leakage flows of liquid hydrogen can be purged through the turbine cavities and entered to the main gas path. The main particularity of this is that LH2 comes very cold; the temperature can be in the range of 30-40K. LH2 is also used in the system to cool and lubricate the bearings. In the oxidizer pump the situation is very different. If oxygen leaks into the turbine the risk for explosion or failure is large. The turbine is driven by a hot fuel rich mixture together with which the oxygen will react very rapidly. This has caused several failures in development as well described in [1]. It is preferred that hot turbine gas flows down through the cavity, with implications in particular for the disk temperature. Alternative or complementary schemes used involve purging with Helium or other inert gases.

3.10 Pitch line or stream line analysis

At this point we have a proposed baseline turbine that should satisfy most constraints and have a good efficiency potential. We can turn this around and start analyzing the performance at design point and at other operating points, as well as start to consider the blade profile shape, angles thickness distributions et.c.. The correlations at this stage will relate the loss in total pressure to the flow velocity or loading. Typically this is given in terms of the total pressure loss over a blade or vane row divided by the exit dynamic pressure. For turbines the common form is:

$$Y_{xx} = \frac{\Delta P_t}{P_{t,out} - P_{s,out}}$$

In the literature and practice a number of different sets of losses are referred in order to describe the losses due to different design features. Ainley-Mathiesen or Kacker-Okapuu are two well known such systems. We shall not go into a detailed discussion on these, only refer to the more important losses in these turbine blades. In [9] a classic text on turbine performance.

- Shock loss – A dominant loss for supersonic turbines such as the LO2 turbine in our example. A normal shock at M=2 relative speed causes a loss of almost 30% of the total pressure. This can increase the total loss dramatically.
- Profile friction loss – The boundary layer friction causes the classical wet surface loss. At high pressures and velocities the Reynolds number is large despite the small size and the wet surface loss therefore relatively small. $Y \propto \mathrm{Re}^{-\frac{1}{5}}$
- Trailing edge – The base surface gives a drag that incurs significant losses, in particular for small blades and mechanically challenged where the thickness to chord ratio is large. Normally $Y \propto \frac{t}{c}$ but can be worsened at high exit Mach numbers or when the thickness/throat ratio becomes large. The aerodynamic designer will want to have 0 trailing edge thickness, while the structural designer will want to have a large radius in order to reduce stress concentration and thermal differences, add to this the manufacturing responsible who wants a large radius and allowances for tolerances and the ingredients of a good discussion are all there.
- Tip leakage loss – For unshrouded blades this can be a dominant loss factor. Thermal and speed transients in the start/stop sequences cause uneven growth of the casing and tip diameter. Tangency or rub is not allowed since debris may result or the rotor can be thrown into whirl. The resulting tip gap must be large enough so that tangency never happens, which may require a tip clearance/blade height. Sealed tip shrouds reduce the problem greatly, especially for low reaction blades where the axial pressure difference driving the leakage is small.
- Secondary flow loss – As the blade aspect ratio becomes low the secondary flows formed when incoming boundary layers roll up or are migrated over the blade surface fills up the passage. Classically the desire has been to stay at h/c 3-4 but at least above 1. This is a very open subject today on how to handle. CFD and 3D design has a large impact on the secondary flow loss.

3.11 CFD supported detailed design

Current trends and efforts in design is to draw advantages of CFD methods becoming better and faster. All designs are analyzed in 3D CFD today. The results are most often allowed to affect the design in a sort of cut and try fashion, where solutions are modeled geometrically meshed and the results of the modification analyzed again. One such area is in optimizing the blade profile rather than designing the profile in a family. This way the optimization comes in late and can be used on detailed design decisions, unless large deviations in performance are discovered that leads to re-work of the preliminary design. This is however undesirable as it involves more people and often causes extra work and problems on other places. 3D analyses are systematically used and often still complemented by testing in turbine rig, the accuracy is today good in 3D CFD as used by experienced engineers that use good CFD codes and procedures. Figure 15 shows Mach number plots in 3d from CFD analysis of supersonic blades.

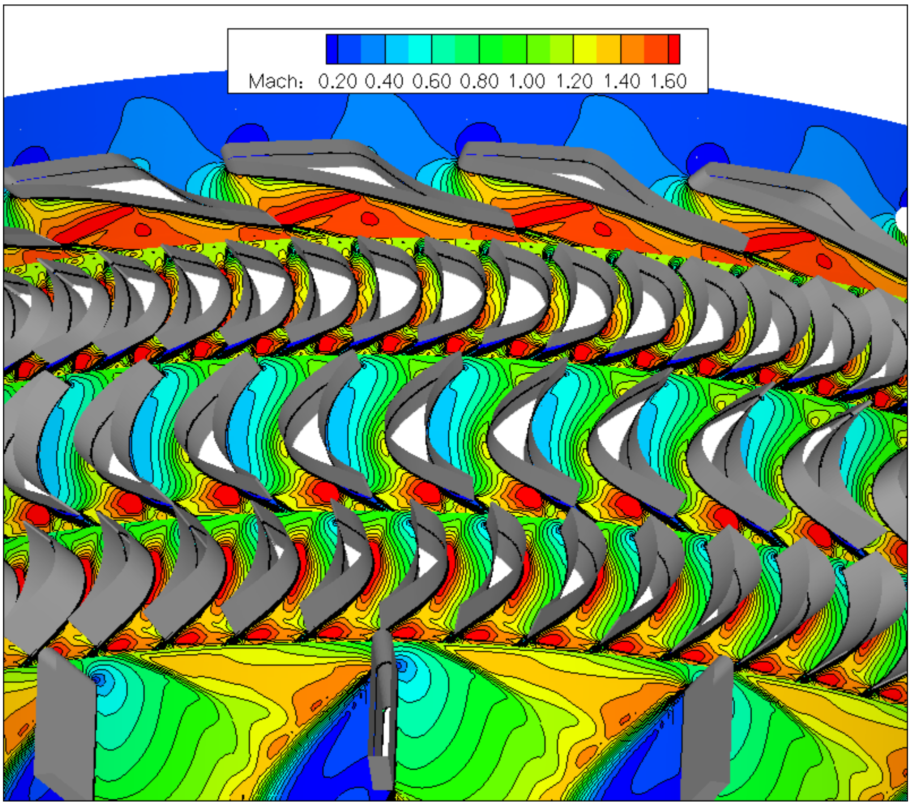


Figure 15 Mach number profiles at mid-span for an LH2 turbine design

The drive is however towards moving the CFD analysis upstream in the design chain so that we can have more parameters open. One such effort has been made in a recent diploma thesis [12]. Here the blade parametric - CFD analysis chain is automated and then used in a optimization loop such that the efficiency can be maximized under the constraints that power and mass flow are constant. Optimization of the rotor blade was tried with a given nozzle first comparing a single design point at 10% reaction. A comparison of Mach number distributions is provided in figure 16. Clearly, the optimized design could be improved over the old design. The matching of the inlet angle and area is better as evidenced by the strong reduction of the bow shock. Also on the exit side the tendency toward overexpansion vanishes and the matching is better. The improvement in profile loss corresponded to 7% in isentropic efficiency.

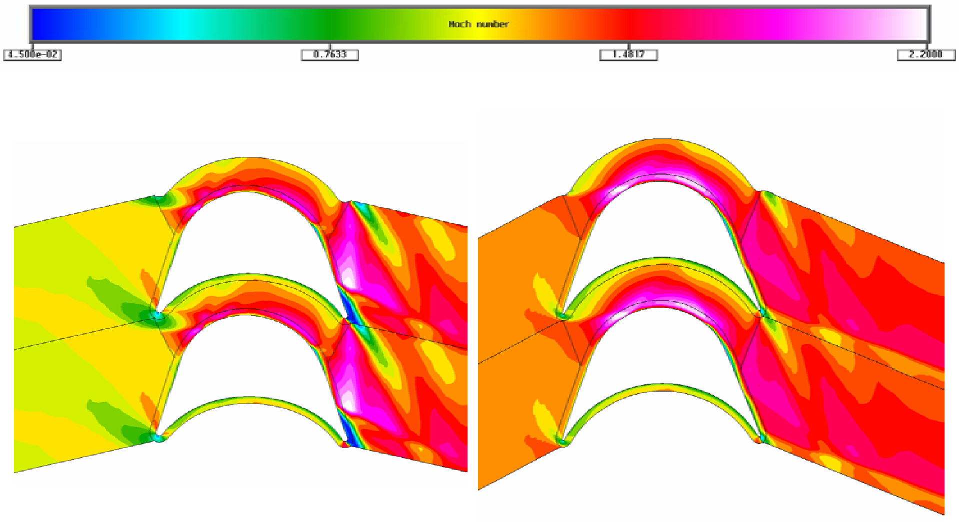


Figure 16 Comparison at10% reaction. Reference blade left, optimized (Blade 1206) right

Perhaps more challenging is to use a range of operation points representative of an operating envelope and trying to find a design giving both high efficiency and low variability in efficiency under different operating conditions. Also this turned out to be possible; Figure 17 shows the Mach number distributions on a reference operating point and a worst case point. Clearly the blade picked out by optimization is very better adapted both operating conditions.

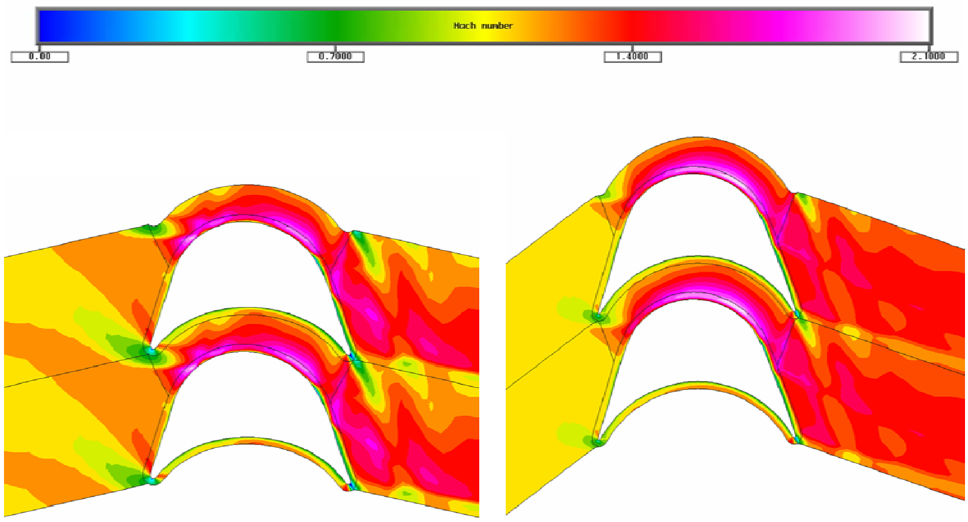


Figure 17a Comparison of Mach numbers at reference blade left optimized blade right, reference operating condition

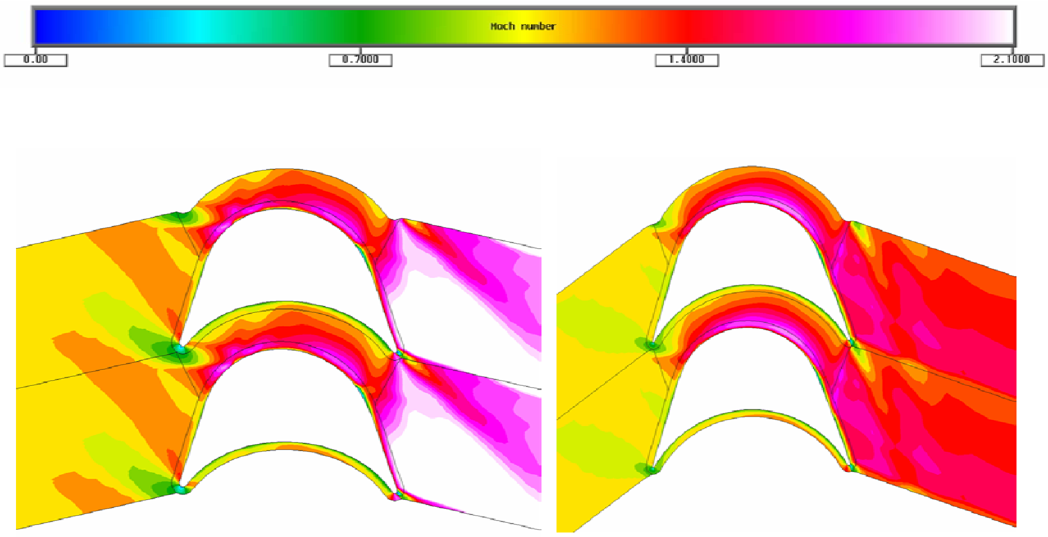


Figure 17b Comparison of Mach numbers at reference blade left optimized blade right, 1206 right off-design point

In fact over 5 operating conditions the variability was very low, and the efficiency improved on all operating points over the reference blade, figure 18. This provides a promising start for a hopefully very good engineering design methodology.

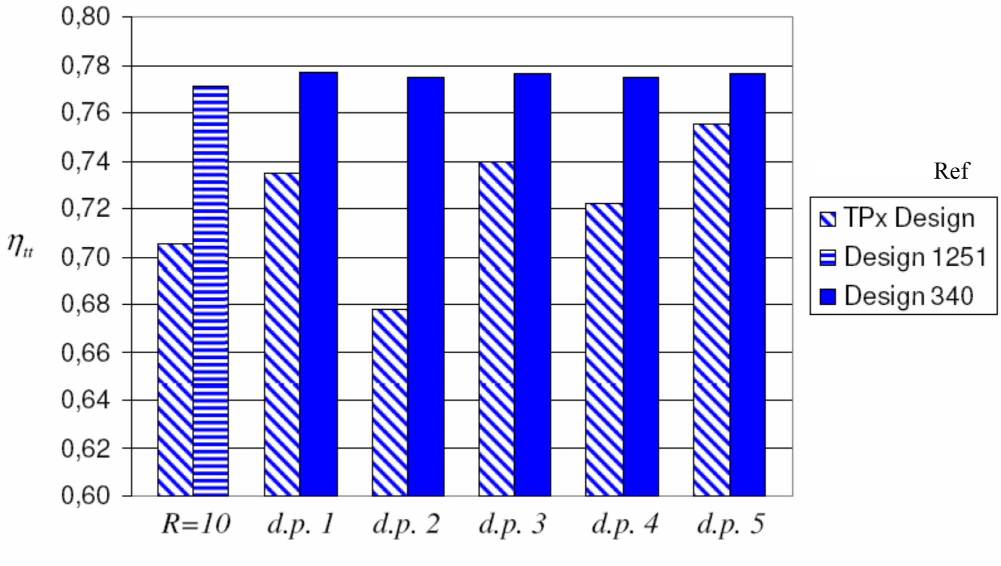


Figure 18 Efficiencies (2D CFD) for optimized profiles

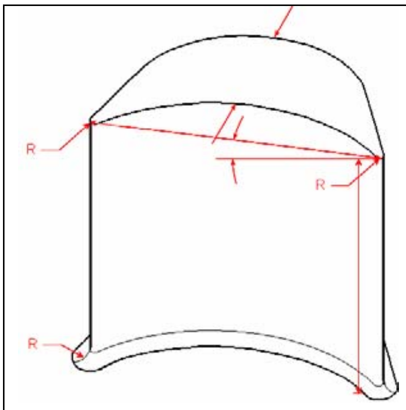
3.12 Experience input and robustness

Minimizing development risk factors are important to the success of a project. These are less intuitive to an engineering student but influence industrial design in the common attitude that we do things the same way as last time. This can lead to the selection of an axial turbine where a radial turbine may appear more appropriate following standard selection charts. This occurs relatively often and is a principal reason why machines tend to look alike with stepwise improvements.

In terms of robustness to tolerances the design should be able to function predictably and not show a large scatter within the manufacturing tolerances. This is a very general statement but often machined processes become difficult if they require better than 0.1 mm accuracy. This could be thought of as form tolerance, thickness of a blade leading edge radius, blade height et.c.. The use of such a measure is very arbitrary but to the designer it could help to think in terms of:

"If this thickness is 0.1 mm larger what is the effect on the flow area?" If the answer is worrying then maybe the solution should be considered once again. A blade with pitch 5 mm and at an angle of 70° will have a gap less than 1.7 mm; with 0.1 mm error this changes the area by more than 5%. A 5% critical area is a rather large perturbation that should be considered carefully. The outcome may shift the design point in order to optimize cost or provide the manufacturing responsible with stricter tolerances and affect the process selection.

In this area as well CFD/FE computations are and will become important tools coupled with statistical methods such as six sigma. An example of this is given in [15] where sensitivities of due to a number of parameterized departures from nominal drawing are analyzed. The relation between tolerances that can be generated in an EDM process and the resulting efficiency can be given.



parameter	dimension	tolerance
leading edge radius	0.209 mm	± 0.1
trailing edge radius	0.157 mm	± 0.1
chord length	11.3 mm	± 0.2
stagger angle	1.786 °	± 2
fillet radius	0.55 mm	± 0.15
blade height	12.5 mm	± 0.2
blade thickness	4.8 mm	± 0.2

Figure 19 Geometric sensitivity parameters studied

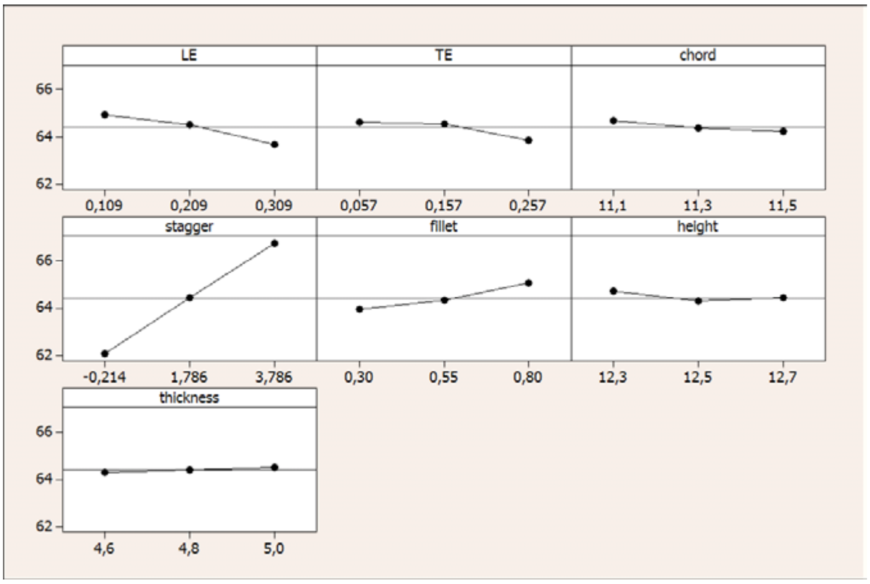


Figure 20 Sensitivity plots for the efficiency due to tolerances

In another application to manufacturing tolerances [10] shows how this is used in order to give allowances for steps in the blade profile and ensuring that the unsteady loads do not increase due to this.

3.13 Mechanics

The mechanical design is highly complex both in static and rotating parts. The main difficulty in the static parts is the severe thermal transients encountered on start and stop transients. Parts designed for burst become thick walled due to high pressures suffer from the thermal gradients encountered during transients. This causes a conflict in the designs that are hard to solve and demands a lot of work and design experience. The environment is more aggressive than in normal atmosphere due to hydrogen embrittlement in the hydrogen rich environment. The demands on specific materials data are great in highly loaded machines. Especially data in hydrogen rich atmosphere or other difficult environments require special equipment and is expensive to acquire.

In order to be efficient and have high power output the blades must move at high speed, as was described in the performance design paragraphs. Limitations in tip speed come from the mechanical strength of the blade, expressed as a burst speed in tip speed. Having chosen the tip speed as near to the experience limit as possible the detailed mechanical design can start. All details must sustain the number of load cycles prescribed for the engine. This is for the rocket engine in comparison to an aero engine is relatively few.

Order of magnitude comparison of cycles for turbomachines

	Number of cycles	Operating time
Power generation gas turbine / Steam turbine		1E5-1E6
Commercial jet aero engine	10000-50000	30000-100000 hours
Fighter engine	1000-10000	1000-10000 hours
Rocket	4-10	1-10 hours

The significance of the difference in cycle count and operating time is relatively large in terms of stress limits allowed and how material data can be used. In an aero engine and gas turbine maintenance cycles are defined from the start and inspections are made that allows identification of possible fatigue problems. Fleet leaders, engines with the highest operating time, can be used in order to find and correct problems. In rocket engines this is not an option as in almost every case the engine only flies once, (the Space Shuttle excepted).

3.14 Blade vibration

Blade vibration is most definitely one of the major design concerns for the turbine. Blade vibration problems in general constitute the probably largest source of difficult problems late in the design project. Both flutter and forced response concerns are difficult issues for the turbine design. The turbine blade will be exposed to high level excitation from the stator wakes and shocks in the aerodynamic system. Figure 21 shows the highly unsteady flow pattern in a supersonic turbine. The level of the forcing at the principal stator passing frequency $\Omega \cdot m$ can be of the same order of magnitude as the steady state load.

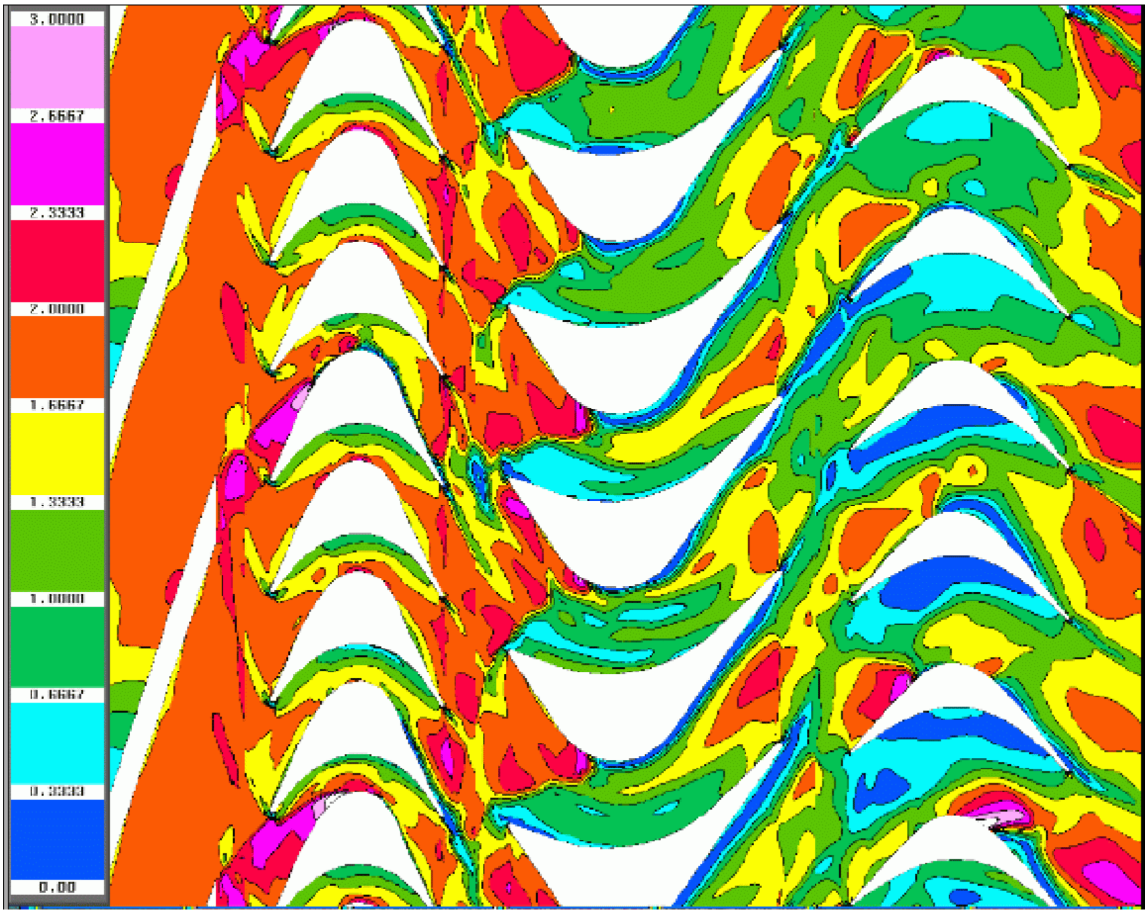


Figure 21: Snapshot from unsteady CFD showing the instantaneous Mach numbers in a supersonic turbine

Forced response is mainly caused by the varying load imposed by flow imperfections leaving the stator and causing a harmonic load onto the rotor. In order to avoid problems the resonances where flow imperfections will excite blade or disk modes the first approach is to work with the Campbell diagrams. For our sample design these are constructed and shown in figure 22.

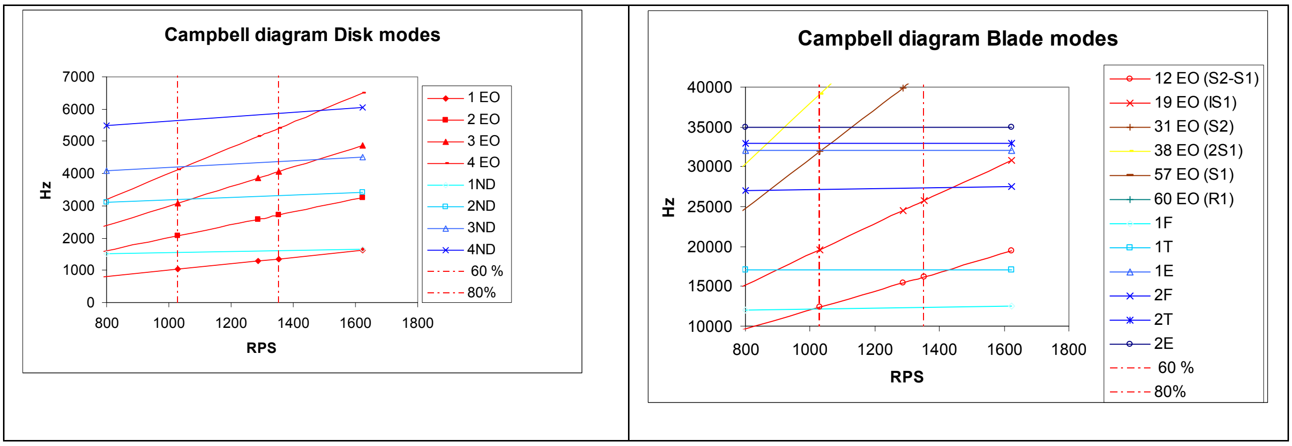


Figure 22 Campbell diagrams for the Rotor 1 blade of the example turbine

The Campbell diagram should be used to identify possible problems. Here experience is of very high importance. There will probably be some crossings where possible excitation sources cross the mode in the operating range. When crossings occur, it is one has to prioritize and use past experience, and then analyze further using a forced response analysis with CFD/FEM. However the possibility that vibrations occur must be taken seriously. Therefore, development programs usually schedules vibrations testing and mostly dampers are developed in the case that high vibrations should occur. An example of damper implementation is given in [11], where a good figure is used, figure 23 to describe the blade platform damper.

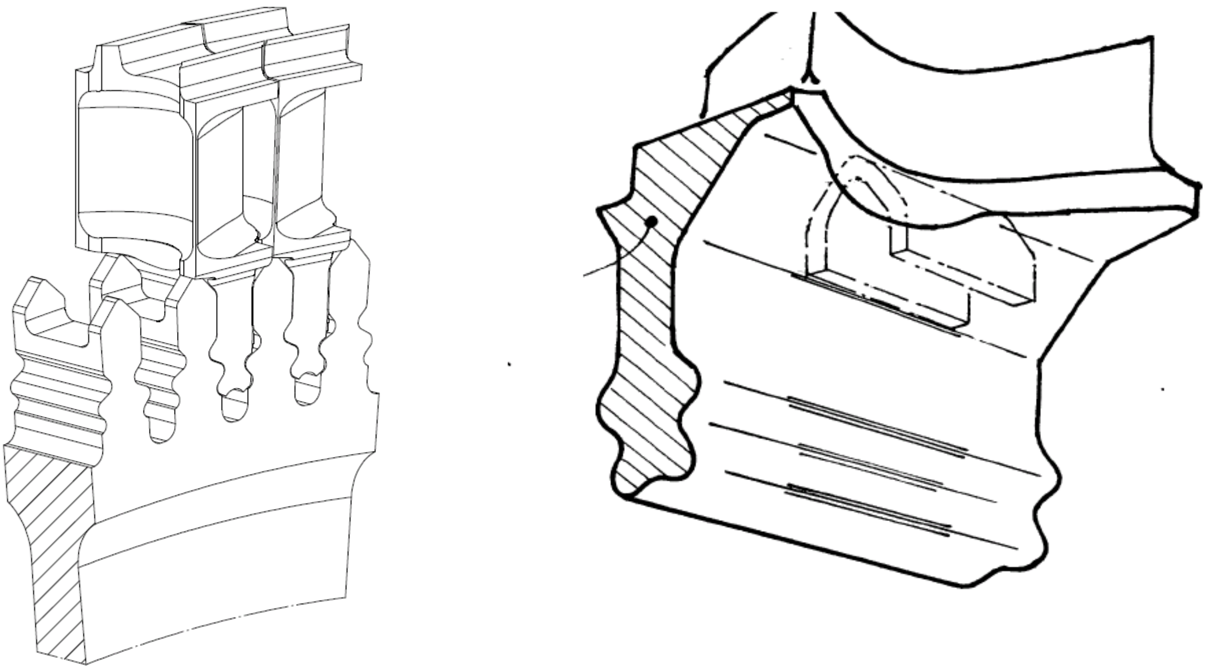


Figure 23 Turbine blade sketch with cottage roof patform damper from [11]

Sometimes dampers are designed for a variety of modes and later used for contingency in the case that high vibrations are encountered in testing.

Flutter is an aeroelastic instability in which a vibration appears at high aerodynamic load as growing amplitude that can cause a failure in a few seconds. [14] Describes this for supersonic turbines, which is particularly difficult in this aspect due to the very high flow velocities in the machine. The most commonly used dimensionless number is the reduced frequency

$$k = \frac{\pi \cdot Chord \cdot f}{V_{rel}}$$

For our examples the LH2 turbine would have for the 1st flex mode (1F) k=0.2 whereas the LO2 single stage design would have k=0.08 which is very low, and must be further considered.

4. REFERENCES

[1] Doyle S., History of liquid rocket engine development in the united states 1955-1980., AAS history series, volume 13, 1992, ISBN 0-97703-349-8

[2] Sutton G., Rocket propulsion elements, 1992, John Wiley and Sons, ISBN 0-471-52938-9

[3] Balje O., A guide to design selection and theory, 1981, John Wiley and Sons, ISBN 0-471-06036-4

[4] NASA SP-8107

[5] NASA SP-8109

[6] NASA SP-8110

[7] NASA SP-8125

[8] Brennen C., Hydrodynamics of pumps, 1994, Oxford University Press ISBN 0-19-856442-2

[9] Horlock J., Axial Flow Turbines, 1866, Butterworths

[10] Bogdan Marcu, Ken Tran , Brad Wright , “Prediction of Unsteady Loads and Analysis of Flow Changes due to Turbine Blade Manufacturing Variations during the Development of Turbines for the MB-XX Advanced Upper Stage Engine” , AIAA-2002-4162, 38th AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit, 7-10 July 2002, Indianapolis, Indiana

[11] K. Holmedahl , ANALYSIS AND TESTING OF THE VULCAIN 2 LOX TURBINE BLADES FOR PREDICTION OF HIGH CYCLE FATIGUE LIFE , AIAA-2000-3680, AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit, 36th, Huntsville, AL, July 16-19, 2000

[12] Abelsson, C (2007) Parametric Turbine Blade Design for Supersonic Applications, Diploma thesis from School of Engineering Sciences, Aerospace Engineering, University of Southampton, UK.

[13] S. Brodin J. Steen, I. Ljungkrona, N. Edin, B. Laumert, Vinci Engine Development Testing, a Summary of Turbine Testing Results, AIAA-2005-3949, 2005

[14] Mårtensson H, Flutter Free Design of Aerodynamically Unstable Supersonic Turbines , AIAA joint propulsion conference, Sacramento CA, 2006

[15] Török, A, Analysis of manufacturing tolerances for a supersonic blade, 2006, M.Sc. Thesis from VAC and Institute foe aero engines (ILA), Univ. Stuttgart, Germany.