

(بِسْمِ اللَّهِ الرَّحْمَنِ الرَّحِيمِ)

بَعْدَ الْعَمَلِ الدُّوْرِبِ وَالْمُتَابِرَةِ فِي شَرِكَةِ طَاقَةِ الشِّمَالِ، تَوَصَّلْنَا إِلَى بَعْضِ النَّتَائِجِ وَالتَّصَامِيمِ وَذَلِكَ عَلَى قَدْرِ إِسْتِطَاعَتِنَا وَضِمْنِ الْمَعْلُومَاتِ الَّتِي اسْتِطَعْنَا مَعْرِفَتَهَا وَتَعَلُّمَهَا.
خِلَالَ الْعَمَلِ، وَاجَهْنَا بَعْضَ الْمَشَاكِلِ حَيْثُ قَامَ أَسَاتِذَةُ الْجَامِعَةِ بِالمُسَاعَدَةِ لِإِكْمَالِ مَا بَدَأْنَا بِهِ.
إِنَّ غَايَتِي الْأُولَى بَعْدَ هَذَا كُلُّهُ هِيَ النِّجَاحُ لِكِي أُثَبِتَ جِدَارَتِي أَمَامَ الْجَمِيعِ وَأَمَامَ نَفْسِي. أَرْجُو أَنْ يَكُونَ الْعَمَلُ عَلَى الْمُسْتَوَى الْمَطْلُوبِ وَشُكْرًا لِكُلِّ مَنْ أَعْطَى مِنْ وَقْتِهِ لِإِنْجَاحِ هَذَا الْمَشْرُوعِ .

After diligent and continuous work in the North Lebanon Alternative Power company, we reached some results and designs to the best we can within the knowledge and information that we could know and learn.

Through the work, we encountered some problems that the doctors helped us to resolve and finish what we started.

After all, my first purpose is the success to prove my worth to everyone and to myself. We hope that my report will be up to the required level. Thanks to all those who gave their time for the success of the work.

Dedication

This is dedicated to my parents who encouraged and supported me in each and every step, and helped me overcome all obstacles and problems we faced. It's their support that encouraged me to continue and never give up, and made me believe in myself.

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Nomenclature

- T: Temperature ($^{\circ}\text{C}$);
- S: Entropy (kJ/kg.K);
- h: Enthalpy (kJ/kg);
- P: Pressure (bar);
- V: Specific volume (m^3/kg);
- X: Percentage of steam into a mixed substance “steam & water”;
- η_t : Turbine’s efficiency;
- \dot{m}_1 : Percentage-extraction of steam from the first part of turbine in point 2 (in figure of installation)
- η : Thermal efficiency;
- P_{el} : Electric power (Watts);
- P_m : Mechanic power (Watts);
- P_{th} : Thermal power (Watts);
- Q_m : Mass flow rate (kg/s);
- Q_v : volume flow (m^3/s);
- ΔH : Difference of enthalpy (between entry and out of steam turbine) (kJ/kg);
- Δh_f : Enthalpy for fixed blades (kJ/kg);
- Δh_m : Enthalpy for moving blades (kJ/kg);
- D: the diameter of the blade;
- N: the number of revolutions per unit time;
- g_c : conversion factor, $32.2 \text{ lb}_m \cdot \text{ft} / (\text{lb}_f \cdot \text{s}^2)$ or $1 \text{ kg} \cdot \text{m} / (\text{N} \cdot \text{s}^2)$
- v_{s_1} : Absolute velocity of fluid leaving nozzle (m/s);
- v_{s_2} : Absolute velocity of fluid leaving blade (m/s);
- v_B : Blades’ velocity (m/s);
- v_{r_1} : Relative velocity of fluid (as seen by an observer riding on the blade) (m/s);
- v_{r_2} : Relative velocity of fluid leaving the blade (m/s);
- θ_1 : Nozzle entrance angle (degree);
- ϕ_1 : Blade entrance angle (degree);
- γ_1 : Blade exit angle (degree);
- δ_1 : Fluid exit angle (degree);
- l: height of the blade;
- D_1 : diameter of shaft;
- D_2 : diameter limited by the blades;
- P: perimeter of the shaft;
- c: distance per blades;
- d: width of blades;
- St: total section;
- n: number of blades around the shaft.

Abstract

The object is making a simple design of large steam turbine in an incineration power plant, which is producing a power of 1.5 MW, regulation and test for it.

So for that, many steps are done as calculation of efficiency, length for blades...then designed it in a 3D program.

This work must have much time for make it. But hoping that what was done is well! A steam turbine should have many engines for regulation as governor for its speed and a steam regulation to regulate the entry quantity of steam. And it is sensitive and may be broken if a wet steam enters to it.

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General introduction

The major issue in Lebanon is pollution; it worries every citizen of the country. It can be classified in different forms which has an impact on everyone's health.

Nowadays, Lebanon faces a big problem, which is land pollution, also known as trash. It leads to other pollutants (in air, water ...). The amount of waste is rising day after day. It is caused by increasing of daily consumption for more population.

We can eliminate garbage by recycling, composting, gasification, sanitary landfill... Waste incineration is one of solutions, so for the installation of incineration power plant (TEMO-IPP), where my internship was done, it is need a simple manufacture of a steam turbine that produces 1.5 MW of power, and then improve results by control it. This company has already a steam turbine that produces 40 kW. So it is being my task to do that larger one.

One of the primary requirements of all steam turbine manufacturers is to standardize their products in order to meet the cost and delivery time targets while, at the same time, providing a high level of flexibility to their customers. This also helps to obtain optimum performance levels and product quality. But our primary is the benefit for our country and don't purchase from outside.

This book includes several chapters, in beginning talking about general information of steam turbine. Then, all steps of calculation to obtain detailed dimensions for each part of turbine, after that make all parts 'design. Finally, the control system and how it works.

Chapter I: Basics

In general, to know well about anything, it is needed to read many documents briefly concern the topic which is studied. In case, the world of steam turbines includes specific key words as blade. In the next paragraphs, it is written about types, parts, materials of a steam turbine.

1.1 State of art

The first reaction blades used by Parsons were short brass cantilevers inserted around the circumference of a drum rotor. Up to 20 rows of such blades were used, with matching stationary blades. An early Parsons turbine from 1891 is shown in Figure I-1. The impulse blade sections sketched by De Laval in Figure I-2 were shaped to redirect the steam flow after the converging nozzle had accelerated this flow to a high velocity. Parson's turbines and their Westinghouse counterparts grew quickly in output between 1891 and 1900. This growth was soon followed by that of machines from other manufacturers around the world. By 1920, turbines of 20 to 30 MW output were considered 'huge' machines.

The first serious blading problems appear to have developed around 1917. These were associated with the "bursting" of bladed turbine disks during operation. By that time, blade shapes had evolved from the relatively short blades of the 1880's to longer more slender profiles of more or less constant section, around the circumference of a thin disk: see Figure I-3. This slender blade and disk design led to the bursting failure problem discussed by Campbell. As a result of Campbell's work on bladed disk oscillations, and his explanation that the disk failures resulted from resonance between backward travelling waves and a standing pressure disturbance in the steam flow, this type of bladed disk design was revised in favor of thicker disks and blades of greater axial width. Figure I-4 shows the presentation of Campbell's famous diagram.

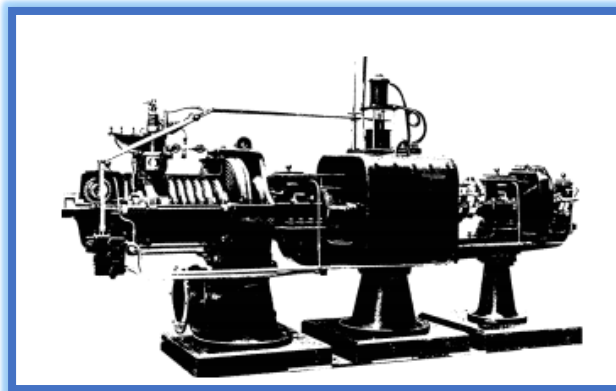


Figure I-1: The First Parsons Condensing Steam Turbine (1891), Courtesy of Siemens-Parsons Co. Ltd

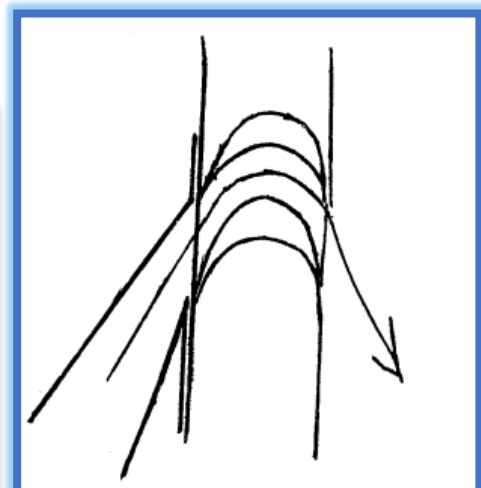


Figure I-2: De Laval's 1888 Sketch of Nozzle and Impulse Blade Profile.



Figure I-3: Blades Damaged by Disk Oscillations. Campbell

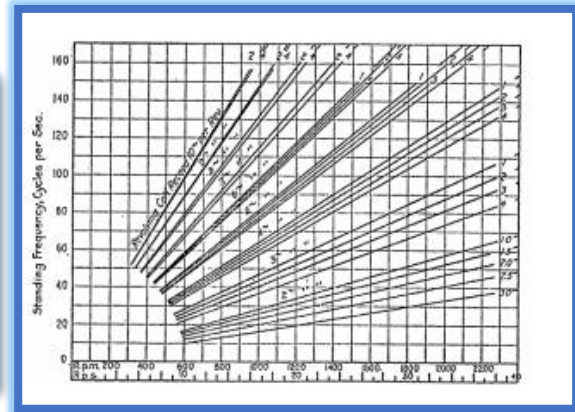


Figure I-4: Campbell Diagram Showing per-rev Lines for Different Blade Lengths. Campbell

The earliest type of blade vibration analysis procedure appears to have involved use of the simple single-degree frequency formula cited by Stodola. An allowance for the stiffening effect of centrifugal force was later introduced using Lamb and Southwell's method, though this procedure appears to have been known earlier, judging by the dates involved. This "allowance" method is mentioned both by Campbell and by Stodola who gave a theoretical explanation for its validity. Apart from the design comments in Stodola's book, Campbell's paper apparently represents the first discussion of blading problems in the literature. The emphasis of Campbell's paper was however on explaining the disk "bursting" problems that he encountered between 1917 and 1921. The same axial vibrations of the disk also caused the broken blades shown by Campbell in Figure I-3. Interest in blading revived in the mid-1930's as machine output continue to increase and blade lengths grew longer. Kroon (1934) described studies of the strength properties of lashing wires, in which Kroon's calculated results were confirmed by measurements of deflections of laboratory beam models. In 1940, Kroon performed a further series of experiments on first stage blades in a turbine operating under high temperature steam inlet conditions. He used an optical method and a mirror system within the disk and rotor bore to observe blade vibrations as the turbine operated. Other damaging vibrations of blade groups were studied by Smith in 1936-37 (but not reported until 1948), by which time similar problems had been encountered in the U.S. by Prohl. Studies were also made during the years 1941-1945 into the vibrations and blade failures encountered in the development of jet engine blades, in the U.K., in the U.S., and probably (unreported) in Germany: see Shannon. (1)

Waste incineration power plant is consider as a thermal power plant in which the prime mover is steam driven. Water is heated, turns into steam and spins a steam turbine which drives an electrical generator. In our country, usually turbines are gotten from outside companies, and they are much complicated and expensive, so our company has some practical ideas to avoid this choice and makes our special and simple design.

1.2 Introduction to turbo-machinery

A turbo- machine is a device in which energy transfer occurs between a flowing fluid and a rotating element due to dynamic action, and results in a change in pressure and momentum of the fluid. Mechanical energy transfer occurs inside or outside of the turbo-machine, usually in a steady-flow process. Turbo-machines include all those machines that produce power, such as turbines, as well as those types that produce a head or pressure, such as centrifugal pumps and compressors.

Chapter I: Basics

The turbo-machine extracts energy from or imparts energy to a continuously moving stream of fluid. However in a positive displacement machine, it is intermittent. The turbo-machine as described above covers a wide range of machines, such as gas turbines, steam turbines, centrifugal pumps, centrifugal and axial flow compressors, windmills, water wheels, and hydraulic turbines. In this text, we shall deal with incompressible and compressible fluid flow machines.

1.3 Types of turbo-machines

There are different types of turbo-machines. They can be classified as:

1. Turbo-machines in which:
 - (i) work is done by the fluid
 - (ii) And work is done on the fluid.

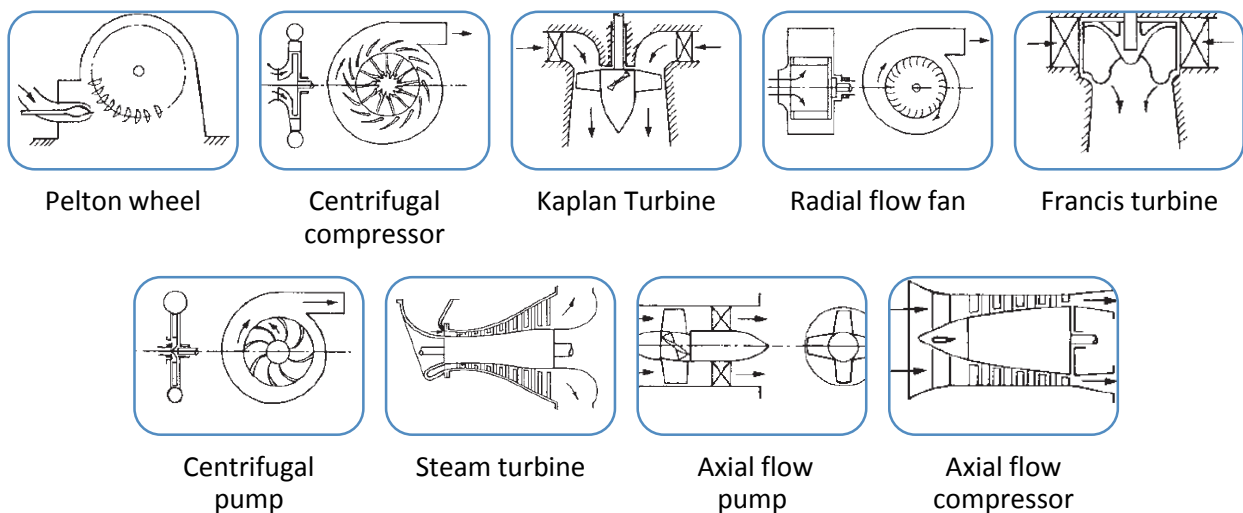


Figure I-5: Types and shapes of turbo-machines.

2. Turbo-machines in which: fluid moves through the rotating member in axial direction with no radial movement of the streamlines. Such machines are called axial flow machines whereas if the flow is essentially radial, it is called a radial flow or centrifugal flow machine.

Two primary points will be observed: first, that the main element is a rotor or runner carrying blades or vanes; and secondly, that the path of the fluid in the rotor may be substantially axial, substantially radial, or in some cases a combination of both. Turbo-machines can further be classified as follows:

Turbines: Machines that produce power by expansion of a continuously flowing fluid to a lower pressure or head.

Pumps: Machines that increase the pressure or head of flowing fluid.

Fans: Machines that impart only a small pressure-rise to a continuously flowing gas; usually the gas may be considered to be incompressible.

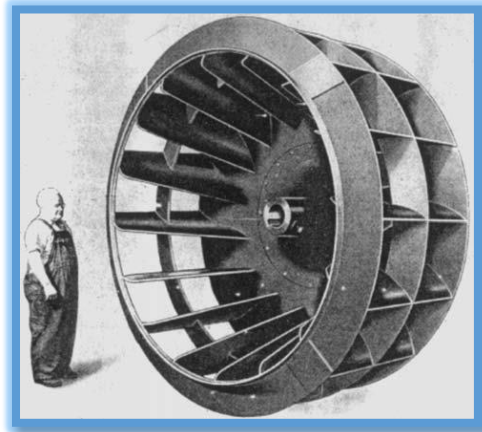


Figure I-6: Radial flow fan rotor. (Courtesy of the Buffalo Forge Corp.)

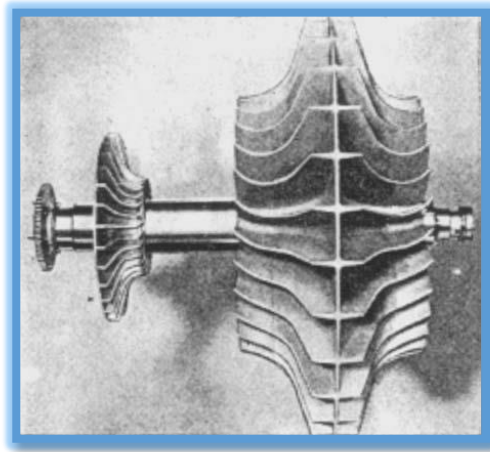


Figure I-7: Centrifugal compressor rotor (the large double-sided impeller on the right is the main compressor and the small single-sided impeller is an auxiliary for cooling purposes). (Courtesy of Rolls-Royce, Ltd.)

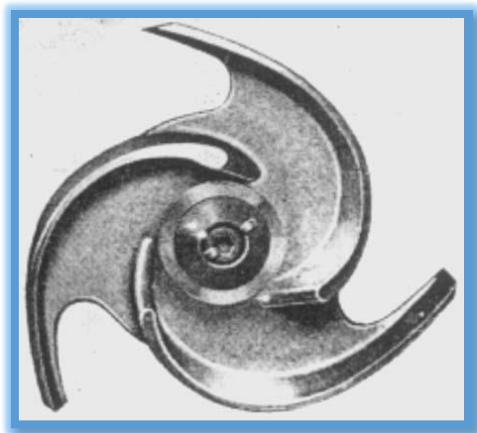


Figure I-8: Centrifugal pump rotor (open type impeller). (Courtesy of the Ingersoll-Rand Co.)

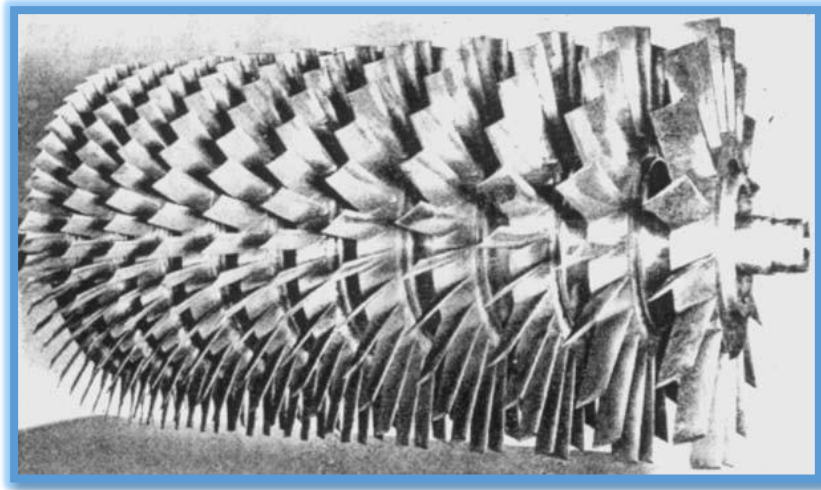


Figure I-9: Multi-stage axial flow compressor rotor. (Courtesy of the Westinghouse electric corp.)

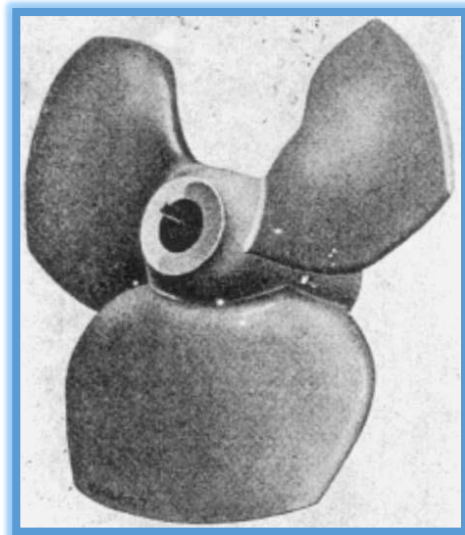


Figure I-10: Axial flow pump rotor. (Courtesy of the Worthington Corp.)

Compressors: Machines that impart kinetic energy to a gas by compressing it and then allowing it to rapidly expand.

Compressors can be axial flow, centrifugal, or a combination of both types, in order to produce the highly compressed air. In a dynamic compressor, this is achieved by imparting kinetic energy to the air in the impeller and then this kinetic energy is converted into pressure energy in the diffuser. (2)

1.4 Steam turbines

Industrial steam turbines represent one of the largest populations of prime movers in the world. They are found in many industries and utilized in a variety of applications. In a steam turbine the stored energy of high temperature and high pressure steam is converted into mechanical (rotating) energy, which then is converted into electrical energy in the generator. The original source of heat can be a furnace fired by fossil fuel (coal, gas, or oil) or biomass. A steam turbine is a device that extracts thermal energy from pressurized steam and uses it to do mechanical work on a rotating output shaft. Because the turbine generates rotary motion it is particularly suited to be used to drive an electrical generator.

The steam turbine is a form of heat engine that derives much of its improvement in thermodynamic efficiency through the use of multiple stages in the expansion of the steam, which results in a closer approach to the most efficient reversible process. An ideal steam turbine is considered to be an isentropic process or constant entropy process, in which the entropy of the steam entering the turbine is equal to the entropy of the steam leaving the turbine. No steam turbine is truly isentropic, however, with typical isentropic efficiencies ranging from 20–90% based on the application of the turbine. The interior of a turbine comprises several sets of blades, or buckets as they are more commonly referred to. One set of stationary blades is connected to the casing and one set of rotating blades is connected to the shaft. The sets intermesh with certain minimum clearances, with the size and configuration of sets varying to efficiently exploit the expansion of steam at each stage. (3)

Steam turbines consist essentially of a casing to which stationary blades are fixed on the inside and a rotor carrying moving blades on the periphery. The rotor is fitted inside the casing with the rows of moving blades penetrating between the rows of fixed blades. Thus steam flowing through the turbine passes alternately through fixed and moving blades with the fixed blades directing the steam at the right angle for entry into the moving blades. Both casings and rotors must be constructed to minimize damaging thermal stresses and the moving blades must be fitted to the rotor securely to withstand the high centrifugal forces.

Where the shaft of the rotor passes through the ends of the casing, a seal is required to prevent steam leakage. Also within the casing, seals are required to prevent steam from leaking around the blades rather than passing through them. Turbine seals are of the labyrinth type where there is no mechanical contact between the fixed and rotating parts. Leakage is thus not really eliminated but merely controlled to minimal amounts. The shafts of the rotors are carried on bearings and are linked together and to the electrical generator. Bearings must be properly aligned to accommodate the natural gravitational bending of the shaft. Allowance must also be made for differential expansion between the rotors and the casings during thermal transients. Both must be free to expand without upsetting the alignment, while allowing the rotors to expand more quickly and to a greater degree than the casing. Lubrication is required for the bearings. Multiple pumps driven by alternative power sources assure adequate lubrication under all operational circumstances. (4)

The advantages of steam turbines are: comparatively low initial cost, low expense for maintenance, small floor space, large overload capacity, exhaust steam is free of oil contamination as no internal lubrication is needed and high efficiency over a wide range of load conditions. The steam turbine can be built in a unit of much greater capacity than is practical with the reciprocating steam engines. (5)

1.4.1 History

The first device that may be classified as a reaction steam turbine was little more than a toy, the classic Aeolipile, described in the 1st century by Greek mathematician Hero of Alexandria in Roman Egypt.

More than a thousand years later, in 1543, Spanish naval officer Belasco de Gray used a primitive steam machine to move a ship in the port of Barcelona. In 1551, Taqi al-Din in Ottoman Egypt described a steam turbine with the practical application of rotating a spit. Steam turbines were also described by the Italian Giovanni Branca (1629) and John Wilkins in England (1648). The devices described by al-Din and Wilkins are today known as steam jacks.

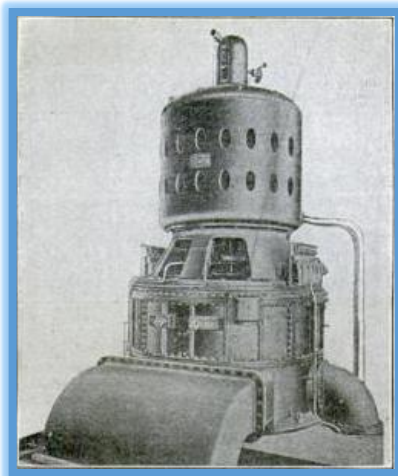


Figure I-11: 2000 KW Curtis steam turbines circa 1905.



Figure I-12: Parsons Turbine from the Polish destroyer ORP *Wicher*.

The modern steam turbine was invented in 1884 by the Englishman Sir Charles Parsons, whose first model was connected to a dynamo that generated 7.5 kW (10 hp) of electricity. The invention of Parson's steam turbine made cheap and plentiful electricity possible and revolutionized marine transport and naval warfare. His patent was licensed and the turbine scaled-up shortly after by an American, George Westinghouse. The Parson's turbine also turned out to be easy to scale up. Parsons had the satisfaction of seeing his invention adopted for all major world power stations, and the size of generators had increased from his first 7.5 kW set up to units of 50,000 kW capacity. Within Parson's lifetime the generating capacity of a unit was scaled up by about 10,000 times, and the total output from turbo-generators constructed by his firm C. A. Parsons and Company and by their licensees, for land purposes alone, had exceeded thirty million horse-power.

A number of other variations of turbines have been developed that work effectively with steam. The de Laval turbine (invented by Gustaf de Laval) accelerated the steam to full speed before running it against a turbine blade. Hence the (impulse) turbine is simpler, less expensive and does not need to be pressure-proof. It can operate with any pressure of steam, but is considerably less efficient.

One of the founders of the modern theory of steam and gas turbines was also Aurel Stodola, a Slovak physicist and engineer and professor at Swiss Polytechnical Institute (now ETH) in Zurich. His mature work was *Die Dampfturbinen und ihre Aussichten als Wärmekraftmaschinen* (English The Steam Turbine and its perspective as a Heat Energy Machine) which was published in Berlin in 1903. In 1922, in Berlin, was published another important book *Dampf und Gas-Turbine* (English Steam and Gas Turbines).

The *Brown-Curtis* turbine which had been originally developed and patented by the U.S Company International Curtis Marine Turbine Company was developed in the 1900s in conjunction with John Brown & Company. It was used in John Brown's merchant ships and warships, including liners and Royal Navy warships. (6)

1.4.2 Steam Turbine Blading

Steam turbine blading have different shapes which are referred to as either impulse blading or reaction blading. Impulse blading is characterized by high velocity fluids entering the turbine blade, by a blade profile that efficiently turns the direction of the fluid with little pressure change, and by decreasing the velocity of the fluid as it leaves the blade to extract energy. Typical impulse blades are crescent or U-shaped and may not always be symmetrical.

Reaction blading is characterized by high velocity fluids entering the turbine blade, but not as high as impulse velocity levels, by a blade profile that efficiently allows the fluid to expand while passing through the blade, and by decreasing both the velocity and pressure of the fluid as it exits from the blade to extract energy. Typical reaction blading has tear-drop shaped leading edges with a tapered thickness to the trailing edge. The blades may have twist to their shape which may range from low amounts of twist or reaction at the base of the blade to high twist or reaction at the tip of the blade.

Impulse type blading is typically utilized in the high pressure or front sections of the steam turbine while reaction blading is utilized in the lower pressure or aft sections of the turbine. Many of today's new steam turbines, however, are utilizing reaction blading in all stages of the turbine including the high pressure sections. Regardless of the blading type, the blade tips may be covered with bands peened to their tips which connect several blades together in groups, or the blades may have integral shrouds which are part of the blades, or may have no tip cover bands or shrouds (free standing). The blade shrouds and cover bands are utilized to keep the passing steam from leaking over the tip of the blades which reduces efficiency and power output and to reduce or dampen the vibration characteristics of the blading. Both stationary and rotating blading can have shrouds or covers depending on the turbine design. The number of blades in a group that are covered by shrouds is dependent upon the vibration characteristics of the specific machine. For some designs, thick wires (called tie wires) are brazed into or between blades to dampen the vibration levels of the blades or groups of blades. In other cases, the tie wires are installed in the blade tips particularly in large blades in the last stages of turbines. And for some blade designs, interlocking tip shrouds (z-shaped) and midspan snubbers (contact surfaces) are utilized to dampen blade vibration, particularly for long last stage turbine blades.

1.4.3 Discs, Rotors, Shafts, Blade Rings, Shells, and Diaphragms

To transmit the torque produced in each stage of the turbine, the rotating blading is fastened to discs or wheels through a specially designed attachment shape at the blade base or root. The root shape may be fir-tree, T-slot, or semi-circular fir-tree shaped or may use multiple pins to hold the blades to the discs. The turbine discs may be shrunk fit onto a shaft with an anti-rotation key or the discs may have been forged with the shaft as an integral assembly. The output shaft from the shrunk fit or integral disc rotor is then connected to the driven equipment through a flange connection or flexible coupling.

Similarly, stationary blading roots may be attached to slots in shells, casings, or blade rings or where the stationary blading is welded to support rings to create a stationary blading assembly referred to as a diaphragm. Depending on the pressure and temperature of the steam to the turbine, there may be dual sets of shells or casings; an inner shell which holds the stationary blading and an outer shell which acts as pressure boundary for the turbine as well as accommodating attachment of blade rings.

The mass and thermal inertial of steam turbine rotors and shells can be quite large. As such, the temperature gradients the rotors and shells can encounter during starting and transients need to be controlled carefully otherwise there can be serious rubs between the rotating and stationary parts and/or there can be extensive distortion of rotors and/or shells when the gradients are too large or occur too fast. (7)

1.4.4 Classification

▪ General

Different criteria can be used for the classification of steam turbines. Two main criteria can be used, namely the principle of work and direction of the vapor stream.

- According to the principle of work can be classified turbines into two major groups:
 - impulse turbines (or action)
 - And reaction turbines.

a) The impulse turbine (or action)

This is the simplest form of a steam turbine. In this type of turbine, the steam expands only in the nozzle. It therefore falls to its total enthalpy in the nozzle. The jets are fixed in the lower part of the casing of the turbine side fins (or vanes) placed on the edge of rotating wheels mounted on a central shaft. This steam moving in a fixed nozzle passes over the curved vanes, which absorb part of the kinetic energy of the expanded steam, thereby rotating the wheel and the shaft on which they are mounted. This turbine is designed so that steam entering through one end of the turbine expands through a succession of nozzles until it has lost most of its internal energy.

b) Reaction turbine

In the reaction turbine, a portion of the mechanical energy is obtained by the impact of the steam on the fins. The most important part is obtained by accelerating the steam during its passage through the turbine wheel. Where it expands a turbine of this type consists of two sets of wings, one fixed and one mobile. These fins are so arranged that each pair acts as a nozzle. It through which the steam expands as it passes. On each floor, a small amount of thermal energy is converted into kinetic energy. The steam expands in the stator vanes. Then causes the moving blades arranged on the wheel or drum of the turbine. The enthalpy drop thus occurs in two stages, first in the nozzle and a second in the rotor.

- Depending on the direction of the stream can be distinguished:

-axial turbine

-And radial turbines.

c) Axial Turbines

In this type, the steam flow is essentially parallel to the axis of the turbine. Axial turbines are mainly composed of an admission torus that channels fluid to the inlet and a stator carrying vanes or distributors (wherein the thermal kinetic energy of the fluid becomes fully (in the case of an impulse turbine) which partially (the case of reaction turbine) into kinetic energy. the rotor carries blades where the kinetic energy and the remaining thermal energy is transformed into mechanical energy.

d) Radial Turbines

The steam flow enters this case perpendicular to the rotor axis. It functions as a centrifugal compressor with a reverse flow (centripetal) and rotation in the opposite direction. It is generally used for low-power and for applications where longer axial turbine (thus more cumbersome) cannot be used. (8)

We choose from those types that are mentioned in the paragraph above, a radial-reaction turbine.

1.4.5 Materials

Nozzles:

- 1- Steel Mn(C 0, 35; Mn 2%).
- 2- Steel Cr Mo Cr (C 0, 2; Cr13; M 0, 1).
- 3- Steel Cr Ni W (Cr 18 stainless).
- 4- Steel 14% de Cr. (9)

Solid rotors:

Experienced manufacturers often use St 460 TS and 461 TS steels for solid rotors. The mechanical properties of these steels and their behavior over longer periods have been under research for many years. Basic mechanical fracture investigations have given information on crack strength, crack propagation velocity, as well as the whole field of brittle fracture behavior. The chemical analysis and mechanical properties of rotor materials St 460 TS and St 461 TS are given in Table I-1.

Table I-1: Composition and Properties of Steels for Solid Rotors

	St 460 TS	St 461 TS
Chemical composition in %		
C	0.17	0.25
Mn	0.30	0.50
Cr	1.20	1.50
Ni	Max.0.60/0.50	0.80
Mo	0.70	1.20
V	0.25	0.35
Mechanical properties, kgf/mm²		
Fracture point at 20°C	σ_B	70+85
Yield point at 20°C	σ_s	Min. 60
Resistance to fatigue		Min.
$\sigma_B/10^5 h$	350°C	44
	400°C	32
	450°C	22
	500°C	13
	550°C	7

These materials correspond to DIN specification 21 CrMoV 511.

SOURCE: Asea Brown-Boveri, Baden, Switzerland. (10)

After knowing primary details about steam turbines, how it is working, and its materials, we choose our type which is a radial, reaction steam turbine. In conclusion stator and rotor are the heart of a steam turbine. For design it we need values, so in the next chapter, it is shown some mathematic methods. Before starts our calculation, we suppose values for some parameters and that is for completing the calculation of others.

Chapter II: Methods and calculation

Generally, in making designs, designer needs to calculate some parameters for any shape. So to find those parameters, some mathematics equations are utilized. All those details to obtain the required values are illustrate in next paragraphs.

There is same rhythm of calculation for two designs, which are different in inlet values of some parameter.

II.1 Element and steps of calculation

This calculation divide into many steps in case to obtain the blade's length, which are calculate of entropies, then all terms of the velocities diagram for each stage, then find the specific volume of intermediary points, then calculate the total section formed by the blades, finally put all characterized parameters founded into the second equation.

II.2 Calculate of thermal efficiency

We worked on a steam turbine which has a power of 1.5 MW.

This turbine is composed of two parts, and each part has two stages.

The electric efficiency is equal to 0.96; mechanic efficiency is equal to 0.95; interne efficiency of turbine is equal to 0.9.

Entry pressure for the turbine is 14 bars, and the entry temperature is 250°C. The pressure in the condenser is 12 kPa.

Figure II-1 shows the installation of an incineration power plant with the steam turbine in that case.

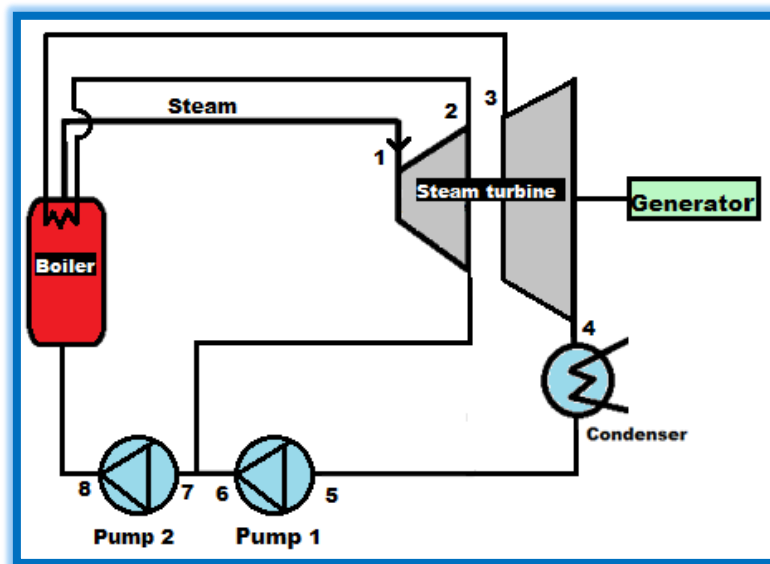


Figure II-1: installation of power plant

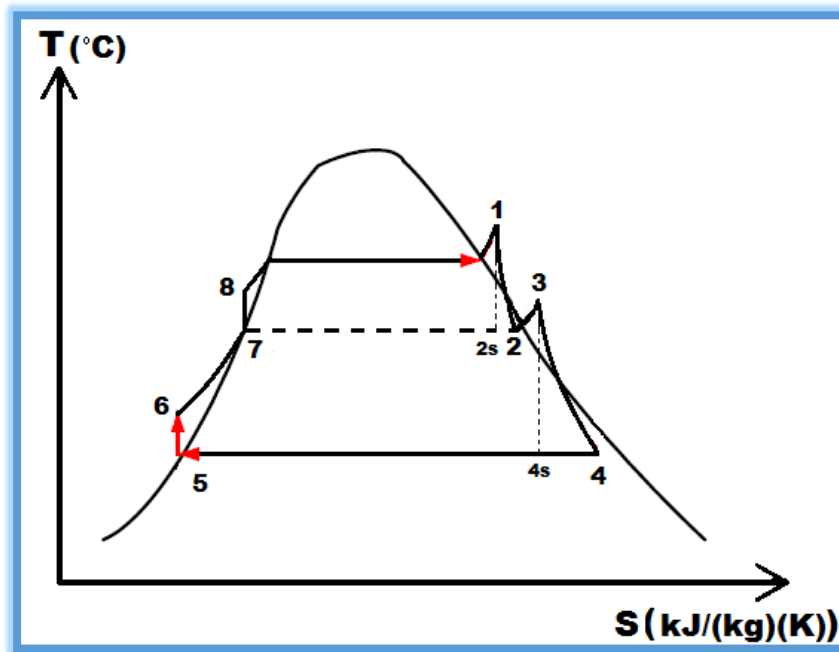


Figure II-2: Rankine cycle

Calculate enthalpy, entropy, etc... for each point.

In point 1 (superheated point)

Numbers are taken from a table named properties of superheated steam (SI units).

Pressure, bar (saturated temperature, °C)		Temperature, °C							
		100	150	200	250	300	400	500	600
6 (158.58)	v			0.3520	0.3938	0.4344	0.5137	0.5920	0.6697
	h			2850.1	2957.2	3061.6	3270.3	3482.8	3700.9
	s			6.9665	7.1816	7.3724	7.7079	8.0021	8.2674
10 (179.91)	v			0.2060	0.2327	0.2579	0.3066	0.3541	0.4011
	h			2827.9	2942.6	3051.2	3263.9	3478.5	3697.9
	s			6.6940	6.9247	7.1229	7.4651	7.7622	8.0290
20 (212.42)	v				0.1114	0.1255	0.1512	0.1757	0.1996
	h				2902.5	3023.5	3247.6	3467.6	3690.1
	s				6.5453	6.7664	7.1271	7.4317	7.7024

Figure II-3: properties of superheated steam (SI units)

Interpolation between 10 and 20 bar:

$$\left\{ \begin{array}{l} T_1 = 250 \text{ }^\circ\text{C}; \\ h_1 = 2926.56 \text{ kJ/kg}; \\ S_1 = 6.77294 \text{ kJ/kg.K}; \\ V_1 = 0.14778 \text{ m}^3/\text{kg}. \end{array} \right.$$

In point 2 (pressure's chute/first part of steam turbine)

$$\frac{P_2}{P_1} = 0.2 \text{ bar} \implies P_2 = 0.2 \times P_1 = 0.2 \times 14 = 2.8 \text{ bar} \sim 3 \text{ bar.}$$

For $P_2 = 3 \text{ bar}$:

Numbers of entropies are taken from saturated table (SI units).

Temperature, °C	Pressure,		Specific volume, m ³ /kg		Specific enthalpy, kJ/kg			Specific entropy, kJ/(kg)(K)		
	bar	psia	v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
120.2	2.0	29.008	0.001061	.8857	504.7	2201.5	2706.2	1.5300	5.5963	7.1263
127.4	2.5	36.259	0.001067	.7187	535.2	2181.3	2716.5	1.6068	5.4451	7.0519
133.6	3.0	43.511	0.001073	.6058	561.2	2163.7	2724.9	1.6710	5.3201	6.9911
143.6	4.0	58.015	0.001084	.4625	604.3	2133.8	2738.1	1.7755	5.1196	6.8951
151.9	5.0	72.519	0.001093	.3749	639.8	2108.4	2748.2	1.8594	4.9611	6.8205
158.9	6.0	87.023	0.001101	.3157	670.1	2086.3	2756.4	1.9299	4.8293	6.7592
170.4	8.0	116.03	0.001115	.2404	720.7	2048.0	2768.7	2.0451	4.6169	6.6620
179.9	10.	145.04	0.001127	.1944	762.5	2015.1	2777.6	2.1378	4.4479	6.5857

Figure II-4: Saturated steam pressure table

Steam percentage:

$$X = (S_1 - S_f) / S_{fg} = \frac{6.77294 - 1.6710}{5.3201}$$

$$X = 0.959.$$

-Point 2_s (adiabatic chute):

$$h_{2s} = X \times h_{fg} + h_f = 0.959 \times 2163.7 + 561.2$$

$$h_{2s} = 2636.1883 \text{ kJ/kg.}$$

$$\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}} \implies h_2 = h_1 - \eta_t (h_1 - h_{2s})$$

$$h_2 = 2926.56 - 0.9(2926.56 - 2636.1883)$$

$$h_2 = 2665.22547 \text{ kJ/kg.}$$

$$\left\{ \begin{array}{l} T_2 = 133.6 \text{ °C;} \\ h_2 = 2665.22547 \text{ kJ/kg;} \\ P_2 = 3 \text{ bar.} \end{array} \right.$$

In point 3 (superheated)

Numbers are taken from a table named properties of superheated steam (SI units).

Pressure, bar (saturated temperature, °C)		Temperature, °C						
		100	150	200	250	300	400	500
0.1 (45.81)	v	17.196	19.51	21.825	24.136	26.445	31.063	35.679
	h	2867.5	2783.0	2879.5	2977.3	3076.5	3279.6	3489.1
	s	8.4479	8.6882	8.9038	9.1002	9.2813	9.6077	9.8978
0.5 (81.33)	v	3.418	3.889	4.356	4.820	5.284	6.209	7.134
	h	2682.5	2780.1	2877.7	2976.0	3075.5	3278.9	3488.7
	s	7.6947	7.9401	8.1580	8.3556	8.5373	8.8642	9.1546
1.0 (99.63)	v	1.6958	1.9364	2.172	2.406	2.639	3.103	3.565
	h	2676.2	2776.4	2875.3	2974.3	3074.3	3278.2	3488.1
	s	7.3614	7.6134	7.8343	8.0333	8.2158	8.5435	8.8342
2.0 (120.23)	v		0.9596	1.0803	1.1988	1.3162	1.5493	1.7814
	h		2768.8	2870.5	2971.0	3071.8	3276.6	3487.1
	s		7.2795	7.5066	7.7086	7.8926	8.2218	8.5133
3.0 (135.55)	v		0.6339	0.7163	0.7964	0.8753	1.0315	1.1867
	h		2761.0	2865.6	2967.6	3069.3	3275.0	3486.0
	s		7.0778	7.3115	7.5166	7.7022	8.0330	8.3251

Figure II-5: properties of superheated steam (SI units)

$T_3=200\text{ }^\circ\text{C};$
 $h_3=2865.6\text{ kJ/kg};$
 $S_3=7.3115\text{ kJ/kg.K};$
 $P_3=3\text{ bar};$
 $V_3=0.7163\text{ m}^3/\text{kg}.$

In point 4 (pressure's chute/second part of steam turbine)

For $P_4=0.12\text{ bar};$

Numbers of entropies are taken from saturated table (SI units).

Pressure, bar	Pressure, psia	Temperature, °C	Specific volume, m ³ /kg		Specific enthalpy, kJ/kg			Specific entropy, kJ/(kg)(K)		
			v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
0.010	0.1450	6.98	0.0010001	129.20	29.30	2484.9	2514.2	0.1034	8.8714	8.9748
0.015	0.2176	13.04	0.0010007	87.98	54.71	2470.6	2525.3	0.1958	8.6312	8.8270
0.020	0.2901	17.51	0.0010014	67.00	73.48	2460.0	2533.5	0.2569	8.4659	8.7228
0.025	0.3626	21.08	0.0010021	54.24	88.49	2451.6	2540.0	0.3083	8.3340	8.6423
0.030	0.4351	24.10	0.0010028	45.66	101.05	2444.5	2545.5	0.3510	8.2258	8.5768
0.040	0.5802	28.98	0.0010041	34.81	121.46	2432.9	2554.4	0.4197	8.0541	8.4738
0.050	0.7252	32.90	0.0010053	28.19	137.82	2423.7	2561.5	0.4740	7.9203	8.3943
0.060	0.8702	36.16	0.0010064	23.74	151.50	2415.0	2566.9	0.5191	7.8105	8.3296
0.070	1.0153	39.03	0.0010075	20.53	163.43	2409	2572.4	0.5591	7.7149	8.2740
0.080	1.1603	41.54	0.0010085	18.10	173.9	2402.6	2576.5	0.5915	7.6364	8.2279
0.090	1.3053	43.79	0.0010094	16.20	183.3	2396.7	2580.0	0.6225	7.5635	8.1860
0.10	1.4504	45.84	0.0010103	14.68	191.9	2392.3	2584.2	0.6488	7.5006	8.1494
0.11	1.5954	47.72	0.0010111	13.40	199.7	2388.3	2588.0	0.6740	7.4420	8.1160
0.12	1.7405	49.45	0.0010112	12.36	207.1	2383.5	2590.6	0.6964	7.3891	8.0855
0.14	2.0305	52.58	0.0010113	10.69	220.3	2375.8	2596.1	0.7371	7.2964	8.0317
0.16	2.3206	55.34	0.0010115	9.433	231.9	2369.1	2601.0	0.7728	7.2124	7.9852

Figure II-6: Saturated steam pressure table

Chapter II: Methods and calculation

Steam percentage:

$$X = \frac{(s^3 - sf) \cdot 7.3115 - 0.6964}{sf \cdot 7.3891}$$

$$X = 0.895.$$

-Point 4s (adiabatic chute):

$$h_{4s} = X \times h_{fg} + h_f = 0.895 \times 2383.5 + 207.1$$

$$h_{4s} = 2340.3325 \text{ kJ/kg.}$$

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}} \implies h_4 = h_3 - \eta_t(h_3 - h_{4s})$$

$$h_4 = 2865.6 - 0.9(2865.6 - 2340.3325)$$

$$h_4 = 2392.85925 \text{ kJ/kg.}$$

$$\left\{ \begin{array}{l} T_4 = 49.45 \text{ }^\circ\text{C}; \\ h_4 = 2392.85925 \text{ kJ/kg}; \\ P_4 = 0.12 \text{ bar.} \end{array} \right.$$

In point 5 (after condensing)

$h_5 = h_f$; for $P = 0.12$ bar ("h" is taken from Figure II-6)

$$\left\{ \begin{array}{l} T_5 = 49.45 \text{ }^\circ\text{C}; \\ h_5 = 207.1 \text{ kJ/kg}; \\ S_5 = 0.6964 \text{ kJ/kg.K}; \\ P_5 = P_4 = 0.12 \text{ bar.} \end{array} \right.$$

In point 6 (water pressed)

$$h_6 = V(\Delta P) + h_5 = 0.001 \times (3 - 0.12) \times 10^2 + 207.1 = 207.388 \text{ kJ/kg.}$$

In point 7 (extraction)

$h_7 = h_f$; for $P = 3$ bar ("h" is taken from Figure II-4)

$$\left\{ \begin{array}{l} T_7 = 133.6 \text{ }^\circ\text{C}; \\ h_7 = 561.2 \text{ kJ/kg}; \\ S_7 = 1.6710 \text{ kJ/kg.K}; \\ P_7 = P_2 = 3 \text{ bar.} \end{array} \right.$$

In point 8 (water pressed)

$$\begin{aligned} h_8 &= V(\Delta P) + h_7 \\ &= 0.001 \times (14 - 3) \times 10^2 + 561.2 \\ &= 562.3 \text{ kJ/kg.} \end{aligned}$$

-Calculate of \dot{m}_1 :

$$\dot{m}_1 (h_2 - h_7) = (1 - \dot{m}_1) (h_7 - h_6) \implies \dot{m}_1 = \frac{h_7 - h_6}{h_2 - h_6};$$

$$\dot{m}_1 = \frac{561.2 - 207.388}{2665.22547 - 207.388}$$

$$= 0.14395 = 14\%.$$

Thermal efficiency

$$\eta = \frac{\text{Net work}}{\text{Quantity of steam additional}} = \frac{\Delta W_{net}}{Q}$$

$$\eta = \frac{(h_1 - h_2) + (1 - \dot{m}_1)(h_3 - h_4) - [(1 - \dot{m}_1)(h_6 - h_5) + (h_8 - h_7)]}{(h_1 - h_8) + (1 - \dot{m}_1)(h_3 - h_2)}$$

$$\eta = \frac{(2926.56 - 2665.22547) + (1 - 0.14395)(2865.6 - 2392.85925) - [(1 - 0.14395)(207.388 - 207.1) + 562.3 - 561.2]}{(2926.56 - 562.2) + (1 - 0.14395)(2865.6 - 2665.22547)}$$

$$\implies \eta = 26.26\%.$$

II.3 Calculate of mass flow

$$P_{el} = 1.5 \text{ MW} \implies P_m = \frac{P_{el}}{\eta_{el}} = \frac{1.5 \times 10^6}{0.96} = 1.562 \text{ MW};$$

$$P_{th} = \frac{P_m}{\eta_m} = \frac{1.562}{0.95} = 1.6 \text{ MW};$$

$$P_{th} = \Delta W_{net} \times Q_m \implies Q_m = \frac{P_{th}}{\Delta W_{net}};$$

$$Q_m = \frac{1.6 \times 10^6}{666.02424904 \times 10^3}$$

$$= 2.402 \text{ kg/s}.$$

II.4 Specific volume

In first part of turbine

$$\Delta H (\text{stage}) = \frac{\Delta h}{2} = \frac{h_1 - h_2}{2} = \frac{2926.56 - 2665.22547}{2} = 130.667265 \text{ kJ/kg}.$$

$$\Delta h_f (\text{fixed blades}) = 0.1 \times \Delta H (\text{stage}) = 0.05 \times 130.667265 = 13.067 \text{ kJ/kg}.$$

$$\Delta h_m (\text{moving blades}) = 0.9 \times \Delta H (\text{stage}) = 0.95 \times 130.667265 = 117.6 \text{ kJ/kg}.$$

$$h (\text{first moving blades}) = h_1 - \Delta h_f - \Delta h_m$$

$$= 2926.56 - 13.067 - 117.6$$

$$= 2795.89 \text{ kJ/kg}.$$

$$h (\text{second moving blades}) = 2795.89 - 13.067 - 117.6$$

$$= 2665.22547 \text{ kJ/kg}.$$

◆ For $h = 2795.89 \text{ kJ/kg}$; And $S = 6.79 \text{ kJ/kg.K}$. (S chosen between S_1 & S_2)
V, P and T are taken from Mollier Chart's diagram.

$$\text{So: } \begin{cases} V = 0.2875 \text{ m}^3/\text{kg}; \\ P = 7 \text{ bar}; \\ T = 177.5^\circ\text{C}. \end{cases}$$

◆ For $P = 3 \text{ bar}$; And $h = 2665.22547 \text{ kJ/kg}$:

Chapter II: Methods and calculation

“S” is taken from Mollier Chart’s diagram

So $S_2=6.82$;

And also from Mollier Chart’s diagram; specific volume V is taken:

$V=0.59 \text{ m}^3/\text{kg}$.

$X=0.968$;

$$\begin{aligned}V_X &= X \times V + (1-X) \times V(\text{liquid}) \\ &= 0.968 \times 0.59 + (1 - 0.968) \times 10^{-3} \\ &= 0.571152 \text{ m}^3/\text{Kg}\end{aligned}$$

In second part of turbine

$$\Delta H (\text{stage}) = \frac{\Delta h}{2} = \frac{h_3 - h_4}{2} = \frac{2865.6 - 2392.58925}{2} = 236.37 \text{ kJ/kg.}$$

$$\Delta h_f (\text{fixed blades}) = 0.05 \times \Delta H (\text{stage}) = 0.05 \times 236.37 = 11.8185 \text{ kJ/kg.}$$

$$\Delta h_m (\text{moving blades}) = 0.95 \times \Delta H (\text{stage}) = 0.95 \times 236.37 = 224.5519 \text{ kJ/kg.}$$

$$\begin{aligned}h (\text{first moving blades}) &= h_3 - \Delta h_f - \Delta h_m \\ &= 2868.6 - 11.8185 - 224.5518563 \\ &= 2629.26 \text{ kJ/kg.}\end{aligned}$$

$$\begin{aligned}h (\text{second moving blades}) &= 2629.26 - 11.8185 - 224.5518563 \\ &= 2392.85925 \text{ kJ/kg.}\end{aligned}$$

◆ For $h=2629 \text{ kJ/kg}$; And $S=7.39 \text{ kJ/kg.K}$. (S chosen between S_3 & S_4)

V , P and T are taken from Mollier Chart’s diagram.

$V=2.26 \text{ m}^3/\text{kg}$.

$X=0.985$;

$$\begin{aligned}V_X &= X \times V + (1-X) \times V(\text{liquid}) \\ &= 0.958 \times 2.26 + (1 - 0.985) \times 10^{-3} \\ &= 2.226115 \text{ m}^3/\text{Kg}\end{aligned}$$

So:

$$\left\{ \begin{array}{l} V_X = 2.26115 \text{ m}^3/\text{kg}; \\ P = 0.6 \text{ bar}; \end{array} \right.$$

◆ For $P=0.12 \text{ bar}$; And $h=2392.85925 \text{ kJ/kg}$:

“S” is taken from Mollier Chart’s diagram.

So $S_4=7.475$;

And also from Mollier Chart’s diagram; specific volume V is taken:

$V=11.5 \text{ m}^3/\text{kg}$.

$X=0.913$;

$$\begin{aligned}V_X &= X \times V + (1-X) \times V(\text{liquid}) \\ &= 0.913 \times 0.59 + (1 - 0.913) \times 10^{-3} \\ &= 10.499588 \text{ m}^3/\text{Kg.}\end{aligned}$$

II.5.1 Calculate of velocities and angles

Equations utilized for this calculation:

$$v_{s1} = \sqrt{2 \times \Delta hf}. \quad (\text{Equation 1})$$

$$v_B = \pi D N = \pi(D_1 + l) \times N. \quad (\text{Equation 2})$$

$$\tan \phi_1 = \frac{v_{s1} \times \sin \theta_1}{v_{s1} \times \cos \theta_1 - v_B}. \quad (\text{Equation 3})$$

$$v_{r1} = \frac{v_{s1} \times \sin \theta_1}{\sin \phi_1}. \quad (\text{Equation 4})$$

$$v_{r2} = \sqrt{2 \times \Delta hm + v_{r1}^2}. \quad (\text{Equation 5})$$

$$\cos \gamma_1 = \frac{v_B}{v_{r2}}. \quad (\text{Equation 6})$$

$$v_{s2} = v_{r2} \times \cos\left(\frac{\pi}{2} - \gamma_1\right) \quad (\text{Equation 7})$$

And those equations are same for the second stage.

Take as hypothesis $\theta = 30^\circ, \delta = 90^\circ$, diameter of the shaft 30 cm, then put a supposable length, so calculate all other parameters, if the result value of length is the same then supposable one, we will take its number. If not, take another value until obtain the supposable one equal to calculate value. (Iterative calculation)

- ◆ The results of calculation for the “first stage, first part” are in Table 2:

Table 2: Calculation of velocity and angles (first stage/first part)

Parameter	Value
debit (Kg/s)	2.402
delta H of first part (KJ/Kg)	130.667265
delta H of second part	236.370375
percentage of fixed blades	0.1
percentage of moving blades (KJ/Kg)	0.9
delta h fixe of first part (KJ/Kg)	13.0667265
delta h mobile of first part (KJ/Kg)	117.6005385
Vs1 (absolute velocity)(m/s)	161.6584455
diameter (m)	0.3
N (round per minute)	3000
VB (velocity of blade) (m/s)	50.57964172
teta1 (degree)	30
tan phi1	0.903920925
phi1 (rad)	0.73497714
phi1 (degree)	42.11108818
Vr1 (relative velocity)(m/s)	120.5380481
delta1 (degree)	90
Vr2	121.5097614
cos(Y1)	0.416259905
Y1(rad)	1.141468303
Y1(degree)	65.40131623
Vs2	110.4822247

Chapter II: Methods and calculation

- ◆ The results of calculation for the “second stage, first part” are in Table 3:

Table 3: Calculation of velocity and angles (second stage/first part)

Parameter	Value
debit (kg/s)	2.402
delta H of first part (kJ/kg)	130.667265
delta H of second part (kJ/kg)	236.370375
percentage of fixed blades	0.1
percentage of moving blades	0.9
delta h fixe of first part (kJ/kg)	13.0667265
delta h mobile of first part (kJ/kg)	117.6005385
Vs3 (absolute velocity)(m/s)	161.6584455
diameter (m)	0.3
N (round per minute)	3000
VB (velocity of blade) (m/s)	52.93583621
teta 2 (degree)	30
tan ϕ_2	0.928383409
ϕ_2 (rad)	0.748277074
ϕ_2 (degree)	42.87311825
Vr3 (relative velocity)(m/s)	118.8006215
δ_2 (degree)	90
Vr4	119.7864298
cos(γ_2)	0.441918474
γ_2 (rad)	1.113060142
γ_2 (degree)	63.77364849
Vs4	107.4550418

- ◆ The results of calculation for the “first stage, second part” are in Table 4:

Table 4: Calculation of velocity and angles (first stage/second part)

Parameter	Value
debit (Kg/s)	2.402
delta H of first part (KJ/Kg)	130.667265
delta H of second part	236.370375
percentage of fixed blades	0.05
percentage of moving blades (KJ/Kg)	0.95
delta h fixe of second part (KJ/Kg)	11.81851875
delta h mobile of second part (KJ/Kg)	224.5518563
Vs1 (absolute velocity)(m/s)	153.7434145
diameter (m)	0.3
N (round per minute)	3000
VB (velocity of blade) (m/s)	62.36061417
teta 1 (degree)	30
tan ϕ_1	1.085987302
ϕ_1 (rad)	0.826596232
ϕ_1 (degree)	47.36047547
Vr1 (relative velocity)(m/s)	104.49779
δ_1 (degree)	90
Vr2	106.6250057
cos(Υ_1)	0.584859187
Υ_1 (rad)	0.946089879
Υ_1 (degree)	54.20695713
Vs2	86.48725706

Chapter II: Methods and calculation

- ◆ The results of calculation for the “second stage, second part” are in Table 5

Table 5: Calculation of velocity and angles (second stage/second part)

Parameter	Value
debit (kg/s)	2.402
delta H of first part (kJ/kg)	130.667265
delta H of second part (kJ/kg)	236.370375
percentage of fixed blades	0.05
percentage of moving blades	0.95
delta h fixe of second part (kJ/kg)	11.81851875
delta h mobile of second part (kJ/kg)	224.5518563
Vs3 (absolute velocity)(m/s)	153.7434145
diameter (m)	0.3
N (round per minute)	3000
VB (velocity of blade) (m/s)	87.4933554
teta 2 (degree)	30
tan ϕ_2	1.683850052
ϕ_2 (rad)	1.034891117
ϕ_2 (degree)	59.29489323
Vr3 (relative velocity)(m/s)	89.40579501
δ_2 (degree)	90
Vr4	91.8830773
cos(Υ_2)	0.952224914
Υ_2 (rad)	0.310355982
Υ_2 (degree)	17.78208789
Vs4	28.06087408

So after all calculations, the velocity diagram for the blades that I obtained is in Figure II-9 and Figure II-10 (designed in free CAD-diagram and blades)

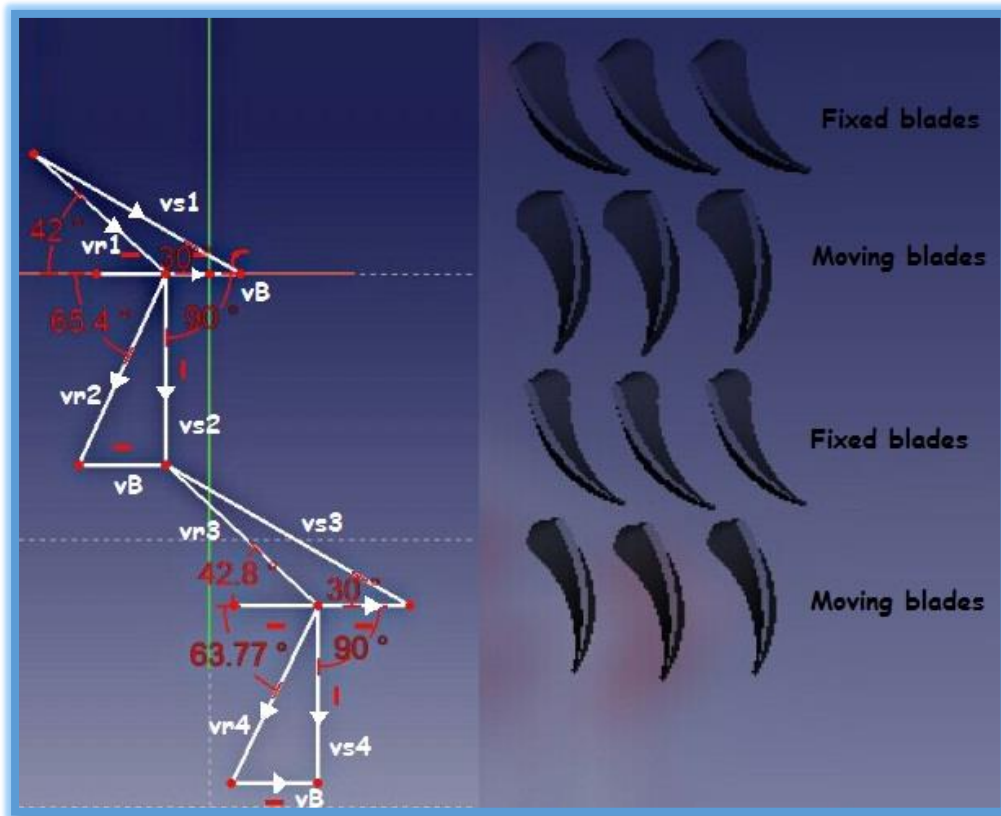


Figure II-9: Velocity diagram for first part

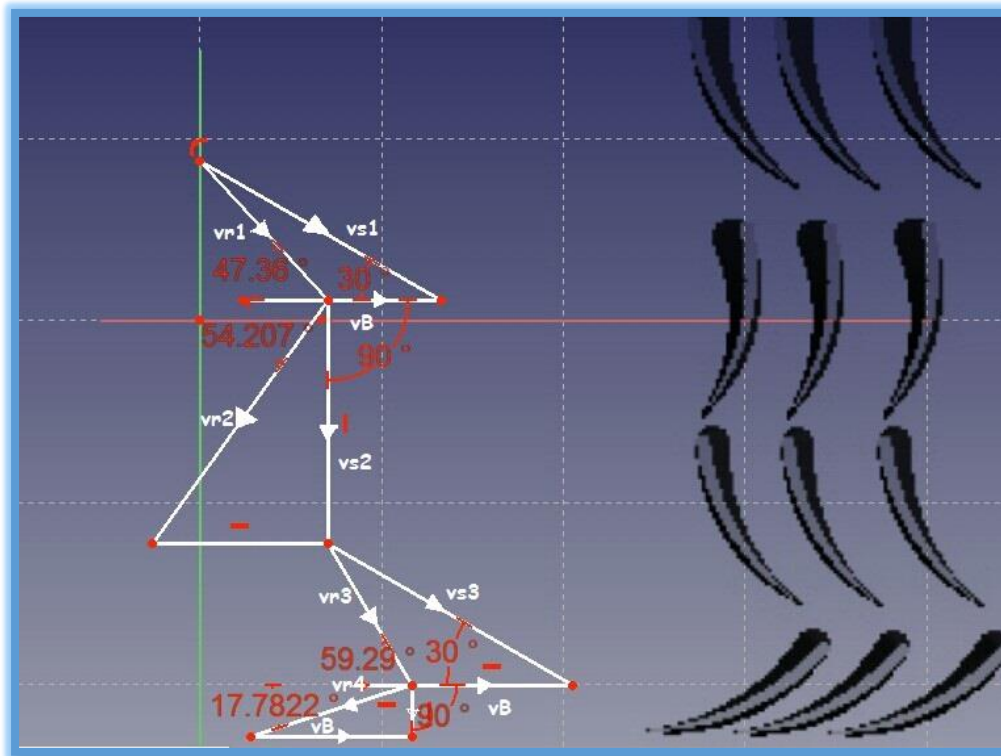


Figure II-10: Velocity diagram for second part

II.6 Calculate the height of blades

For this step of calculation, we need an equation which include one variable. This equation is a second degree equation, so its solution is two different number, known by calculate of delta (Δ). Length is a positive value, then we choose the positive solution.

II.6.1 Second degree's equation

- ◆ The perimeter of shaft is related to its diameter:
 $P = \pi \times D_1$
- ◆ The diameter limited by blades is related to length and diameter of shaft:
 $D_2 = \pi \times (D_1 + 2 \times L)$
- ◆ The number of blades is known by dividing the perimeter of shaft to the sum of the width and distance shown in the drawing:
 $n = \frac{P}{(c+d)} = \frac{\pi D_1}{(c+d)}$;

Example: $P=94.25$ cm, for $D_1=30$ cm;
And $c=0.5$ cm, $d=1.5$ cm; } \implies $n=47$ blades.

- ◆ The total section is the sum of each individual section formed between entries which has the shape of a trapezoid. So it is multiplied by n:

$$S_t = n \times \left[\left(\frac{c+b}{2} \right) \times L \right] \quad \text{(Equation 8)}$$

Or: $D_2 = n \times (b + d)$.

So:

- ◆ $b = \frac{\pi(D_1+2L) - \frac{\pi D_1}{(c+d)} \times d}{\frac{\pi D_1}{(c+d)}} \quad \text{(Equation 9)}$

Put “b” from Equation 9 in Equation 8:

$$S_t = \frac{\pi D_1}{(c+d)} \left[\frac{c + \frac{\pi(D_1+2L) - \frac{\pi D_1}{(c+d)} \times d}{\frac{\pi D_1}{(c+d)}}}{2} \right] \times L$$

$$\implies S_t = \frac{c}{2} \frac{\pi D_1}{(c+d)} \times L + \frac{\pi}{2} (D_1+2L) \times L - \frac{\pi D_1}{(c+d)} \times \frac{d}{2} \times L;$$

So the second degree's equation (L is the unknown number) is:

(Equation 10)
$$\pi L^2 + \frac{\pi D_1 c}{c+d} L - S_t = 0$$

Put in example: $D_1=30$ cm, $c=0.5$ and $d=1.5$ in Equation 10.

→
$$\pi L^2 + 7.5\pi - S_t = 0$$

The Value of S_t is known, that is explained in paragraph II.5.2.

II.6.2 Volume flow

It is necessary to know the value of volume flow, in result to know the total section. We notice that volume flow is equal to mass flow time specific volume.

$Q_v = Q_m \times V$.

V changes from stage to other (it is founded in paragraph II.4).

- ◆ Q_v for “first stage -moving blades”: $Q_v = v_{r2} \times S_t \rightarrow St = \frac{Q_v}{v_{r2}}$.
- ◆ Q_v for “second stage-moving blades”: $Q_v = v_{r4} \times S_t \rightarrow St = \frac{Q_v}{v_{r4}}$.

II.6.3 Tables of results

Before taken the width of blade shown in the face of cylinder, where blades are putted, and also the distance between them, we draw the array among the cylinder, so we put many individual blades the number that we can, then we take the distance and put it in the table to calculate the length according to those two numbers taken, “width and distance”.

This evaluation is done for the two parts of turbine.

- ◆ The results of calculation of “L” for the “first stage, first part” are in Table 6: Calculation of height (first stage-first part).

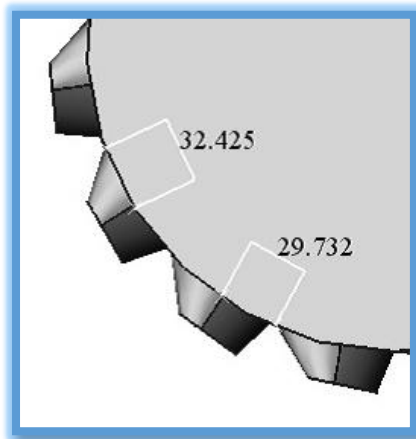


Figure II-11: distance and width for “first stage-first part”

Table 6: Calculation of height (first stage-first part)

Parameter	Value
v(specific volume)(m ³ /Kg)	0.2875
Qv (volume flow rate) (m ³ /s)	0.690575
St(total section)(cm ²)	56.83288254
distance per blades (cm)	2.9
width (cm)	3.2
Diameter (cm)	30
Δ	2721.789507
height of blade (cm)	1.172088793
height (m)	0.011720888

- ◆ The results of calculation of “L” for the “second stage, first part” are in Table 7:

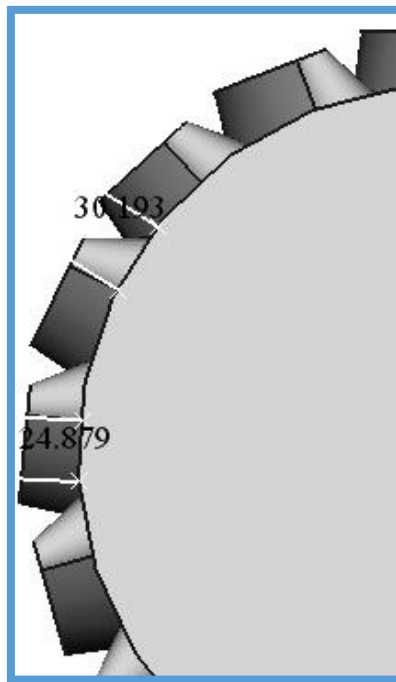


Figure II-12: distance and width for “second stage-first part”

Table 7: Calculation of height (second stage-first part)

Parameter	Value
v(specific volume)(m ³ /Kg)	0.571152
Qv (volume flow rate) (m ³ /s)	1.371907104
St(total section)(cm ²)	114.529426
distance per blades (cm)	2.4879
width (cm)	3.0193
Diameter (cm)	30
Δ	3252.004625
height of blade (cm)	2.299714521
height in (m)	0.022997145

- ◆ The results of calculation of “L” for the “first stage, second part” are in Table 8:

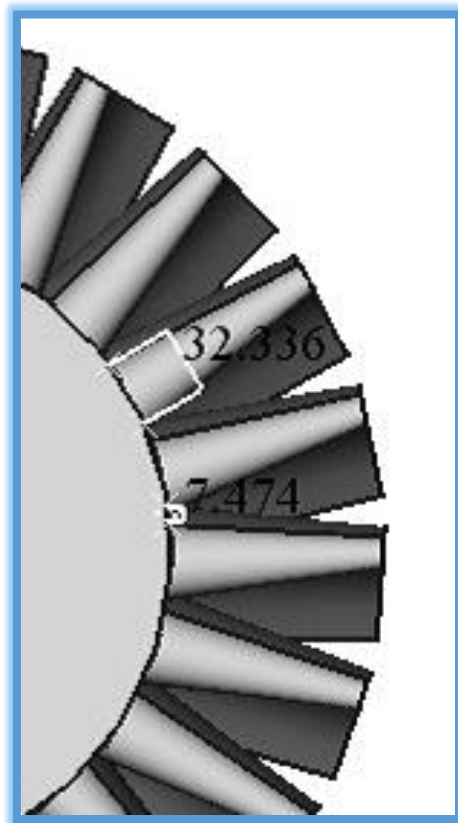


Figure II- 13: distance and width for “first stage-second part”

Table 8: Calculation of height (first stage-second part)

Parameter	Value
v(specific volume)(m ³ /Kg)	2.226115
Qv (volume flow rate) (m ³ /s)	5.34712823
St(total section)(cm ²)	501.4891391
distance per blades (cm)	0.7474
width (cm)	3.2336
Diameter (cm)	30
Δ	6614.984697
height of blade (cm)	10.12835175
height (m)	0.101283518

- ◆ The results of calculation of “L” for the “second stage, second part” are in Table 9:

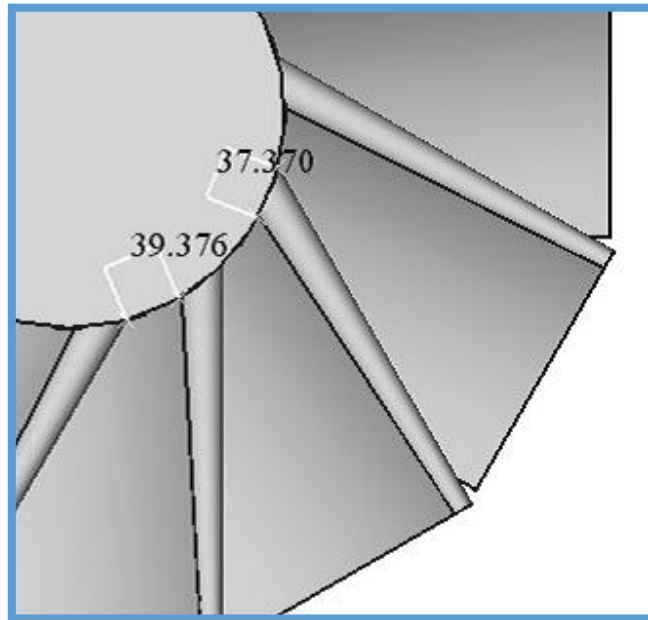


Figure II-14: distance and width for “second stage-second part”

Table 9: Calculation of height (second stage-second part)

Parameter	Value
v(specific volume)(m ³ /Kg)	10.499588
Qv (volume flow rate) (m ³ /s)	25.22001038
St(total section)(cm ²)	2744.793831
distance per blades (cm)	3.9376
width (cm)	3.737
Diameter (cm)	30
Δ	36830.36273
height of blade (cm)	22.84776811
height (m)	0.228477681

II.7 Force of a reaction stage

The work of a reaction stage can be obtained from momentum-impulse or first-law principles. The change in the momentum on the blade in the direction of motion +x is due to the change in the components of the relative velocities v_{r1} and v_{r2} in that direction. For one reaction stage in general:

$$F = \frac{Q_m}{g_c}(v_{r1} \cos \phi + v_{r2} \cos \gamma) \quad (\text{Equation 11})$$

But as $v_{r1} \cos \phi = v_{s1} \cos \theta - v_B$;

$$F = \frac{Q_m}{g_c}(v_{s1} \cos \theta - v_B + v_{r2} \cos \gamma)$$

The rate of work, or power, $\dot{W} = Fv_B$

$$\dot{W} = Q_m \frac{v_B}{g_c}(v_{s1} \cos \theta - v_B + v_{r2} \cos \gamma) \quad (\text{Equation 12})$$

From first-law principles, applies:

$$\dot{W} = \frac{Q_m}{2g_c}[(v_{s1}^2 - v_{s2}^2) - (v_{r1}^2 - v_{r2}^2)] \quad (\text{Equation 13})$$

Table 10: Force and work

Part		F(N)	\dot{W} (J/s)
Part 1	First stage	336.2808	17008.96
	Second Stage	336.2808	17801.3
Part 2	First stage	319.816	19943.92
	Second stage	319.816	27981.77

II.8 Efficiency (fixed blades, moving blades and entire stage)

The efficiency of a reaction stage is dependent upon the efficiency of the fixed blades (nozzles) and the efficiency of the moving blades. These are explained with the help of Figure II-7 and Figure II-15, which is a Mollier chart representation of Figure II-

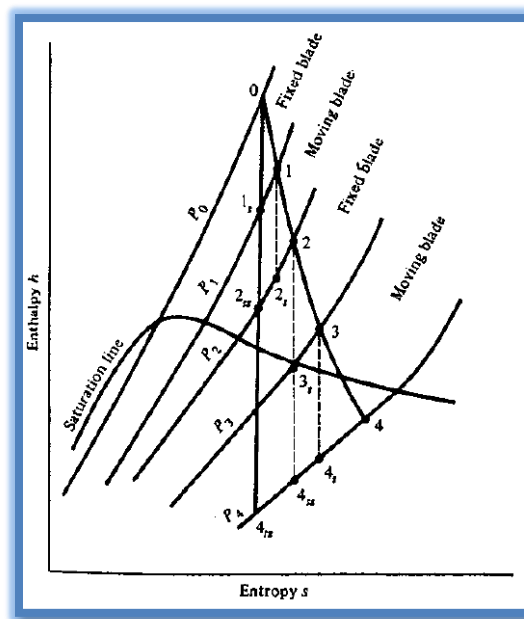


Figure II-15: Condition curve for a two-stage reaction turbine, drawn on the Mollier (enthalpy-entropy) chart.

Lines P_0 , P_0 , etc., represent constant pressures lines, which diverge to the right on the Mollier chart. The actual expansion 1-2-3-4 represents the actual condition of the steam in the two stages and is called a condition curve.

The fixed-blade, or nozzles, efficiency η_N is the ratio of the kinetic energy change to the adiabatic reversible (isentropic) energy change across the fixed blade. For the first fixed blade:

$$\eta_N = \frac{(1/2 g_c)(v_{s1}^2 - v_{s0}^2)}{\Delta h_{f.1}} = \frac{h_0 - h_1}{h_0 - h_{1s}} \quad (\text{Equation 14})$$

The moving-blade efficiency η_B is the work of the blade, divided by the total energy available to that blade, which consists of the kinetic energy of the incoming steam at v_{s1} plus adiabatic reversible (isentropic) enthalpy drop across it. Note that the latter is greater than Δh_m because of the friction (irreversibility) in the blade, which causes an increase in entropy. Thus

$$\eta_B = \frac{\dot{W}}{Q_m(v_{s1}^2/2 g_c) + \Delta h_{ms}} = \frac{\dot{W}}{Q_m[(v_{s1}^2/2 g_c) + (h_1 - h_{2s})]} \quad (\text{Equation 15})$$

The stage efficiency η_{stage} of a reaction stage is the work of the moving blade in the stage divided by the adiabatic reversible (isentropic) enthalpy drop for the entire stage, including fixed and moving blades. Thus

$$\eta_{stage} = \frac{\dot{W}}{Q_m \Delta h_s} = \frac{\dot{W}}{Q_m(h_0 - h_{2ss})} \quad (\text{Equation 16})$$

Where, as shown in Equation 15 to Equation 17

Δh_{fs} = isentropic enthalpy drop across fixed blade

$$= h_0 - h_{1s} \text{ (first stage), } = h_2 - h_{3s} \text{ (second stage), etc.}$$

Δh_{ms} = isentropic enthalpy drop across moving blade

$$= h_1 - h_{2s} \text{ (first stage), } = h_3 - h_{4s} \text{ (second stage), etc.}$$

Δh_s = isentropic enthalpy drop across entire stage

$$= h_0 - h_{2ss} \text{ (first stage), } = h_2 - h_{4ss} \text{ (second stage), etc.}$$

Note that because of the divergence of the constant pressure lines, the isentropic enthalpy drop charged to a row or stage is greater than that for a succession of rows or stages or for an entire turbine. In other words

$$h_1 - h_{2s} > h_{1s} - h_{2ss}$$

$$(h_0 - h_{2ss}) + (h_2 - h_{4ss}) > h_0 - h_{4ss}$$

The reaction turbine is an efficient machine that is suited for large capacities. For a given blade speed, limited by material centrifugal stresses, the steam velocity in a reaction turbine is about half that in a pressure-compounded impulse turbine, resulting in low-friction losses. On the other hand, its work, for the same " v_B ", is about half that of an impulse stage.

Contrary to an impulse stage, a reaction stage has a pressure drop across the moving blades. This makes it less suitable for work in the high pressure stages where ΔP per unit enthalpy drop is high, which results in steam leakage around the tips of the blades, which in turn leads to throttling and a loss of availability. It follows then that impulse staging is preferable in the entrance stages of a turbine, when the pressures are high, steam specific volumes are low, and the blade height is small so that steam velocities would be correspondingly low. In the low pressure stages, reaction stages are preferred because the ΔP across the moving blades is less; the blades become progressively longer so that the tip clearance becomes smaller relative to the blade height, i.e., relative to the steam volume. With large reaction blading, v_B is larger negating the disadvantage of lower power per stage than an impulse stage of the same " v_B ".

II.9 Axial thrust

Turbines rotors are subjected to an axial thrust as a result of pressure drops across the moving blades and changes in axial momentum of the steam between entrance and exit. This axial thrust must be counteracted to keep the rotor in place. In impulse turbines, there is no pressure drop across the moving blades if the turbine is ideal and little pressure drop caused by friction in a real turbine. In addition there is an axial force on the row because of the change in the axial component of momentum of the steam from entrance to exit. This is given by (see Figure II-16)

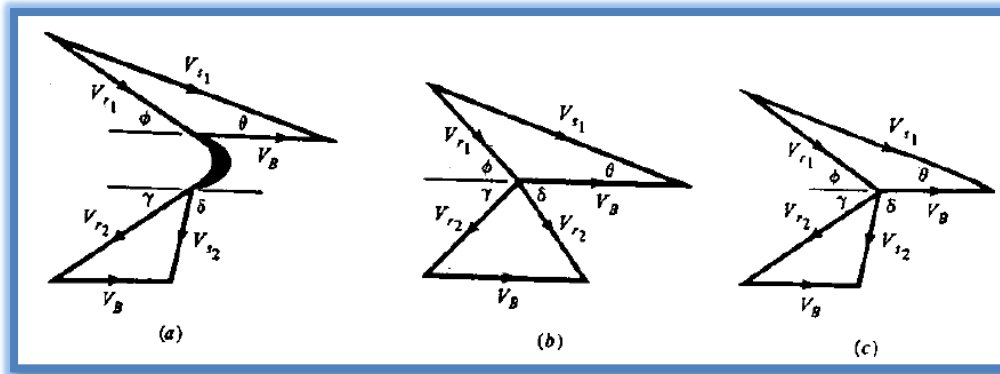


Figure II-16: Velocity diagrams on a single-stage impulse blade.

$$F_{axial} = \frac{Qm}{g_c} (v_{r1} \sin \phi - v_{r2} \sin \gamma)$$

This axial thrust results is no work. In the case of pure symmetrical impulse blades, $v_{r1} = v_{r2}$, $\phi = \gamma$ and that thrust is zero. The total axial thrust on an impulse turbine rotor is, in any case small and poses no severe problems.

The case of the reaction turbine is different. The axial components of the steam entering and leaving a reaction turbine are nearly equal. But in my results, they are not because of not symmetrical blades ($\phi \neq \gamma$).

We notice that the height of blade increases when the pressure decreases and also there is variation of velocity for each stage. Force and work change for each part of steam turbine. Usually the second step after calculation, is obtaining the results designs which illustrate how to manufacture each item. Those design are made in a 3D software.

Chapter III: Detailed design

This chapter includes the design of parts in steam turbine (producing 1.5 MW) which I could reach.

First of all, my responsible give me permission to use only “Free CAD” program for a 3D design, so for that I did some training on it. And by the time, I knew how to use it much more.

Before understanding well about details, design of blades and calculation of turbine parameters, I did two preliminary designs to be more experienced on the program. That is shown in Figure III-4 and Figure III-5.

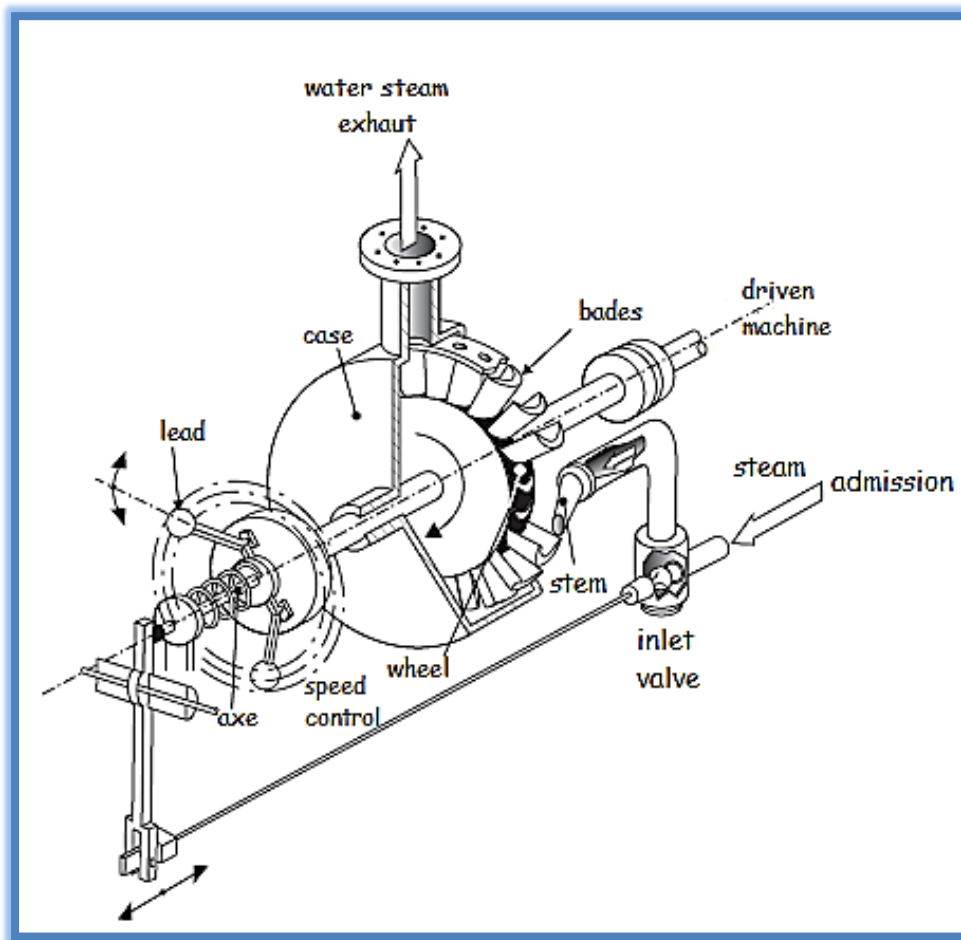


Figure III-1: Parts for one stage steam turbine (12)

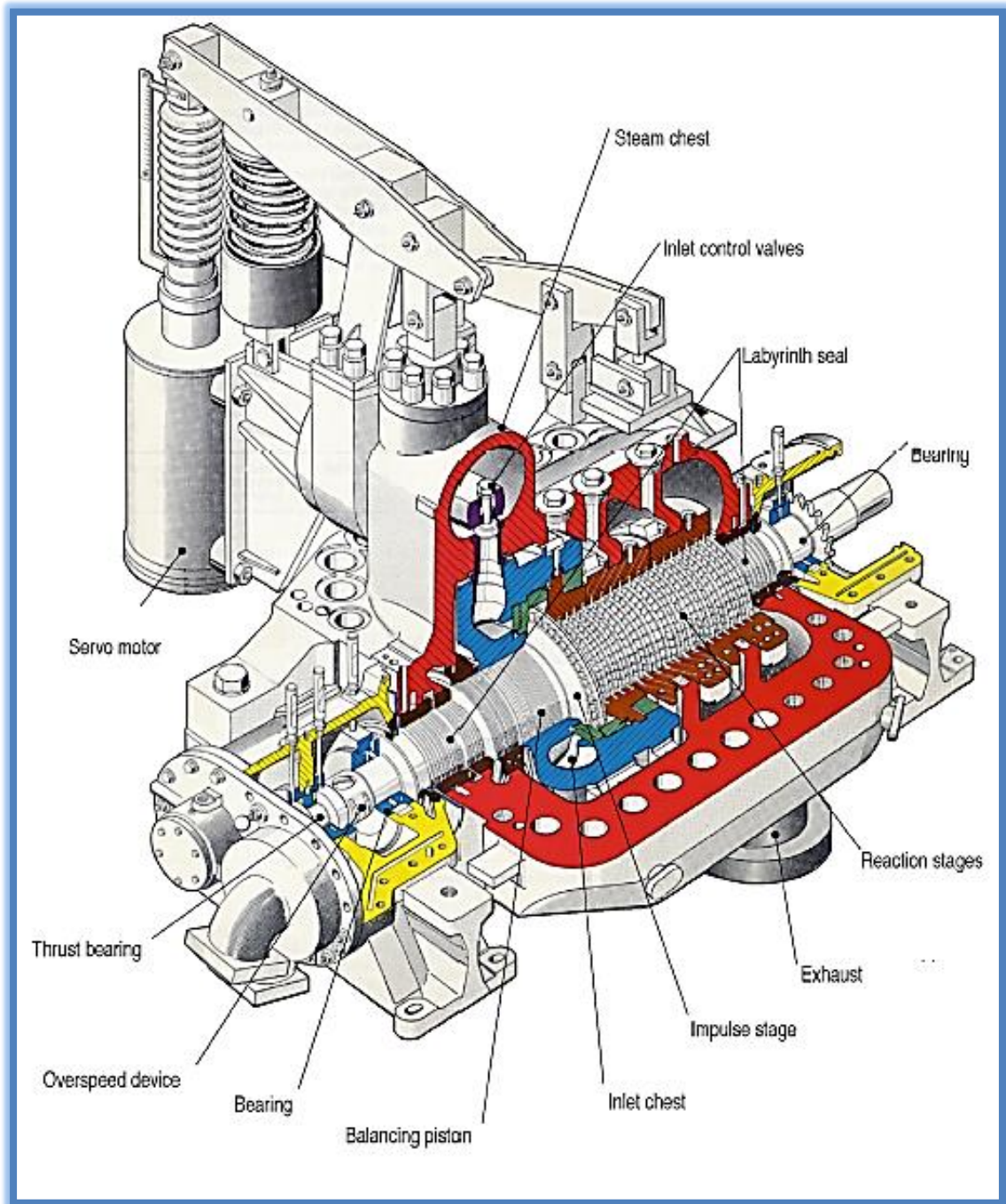


Figure III-2: parts of a multi-stage steam turbine (12)



Figure III-3: stator design (example of real one) (13)

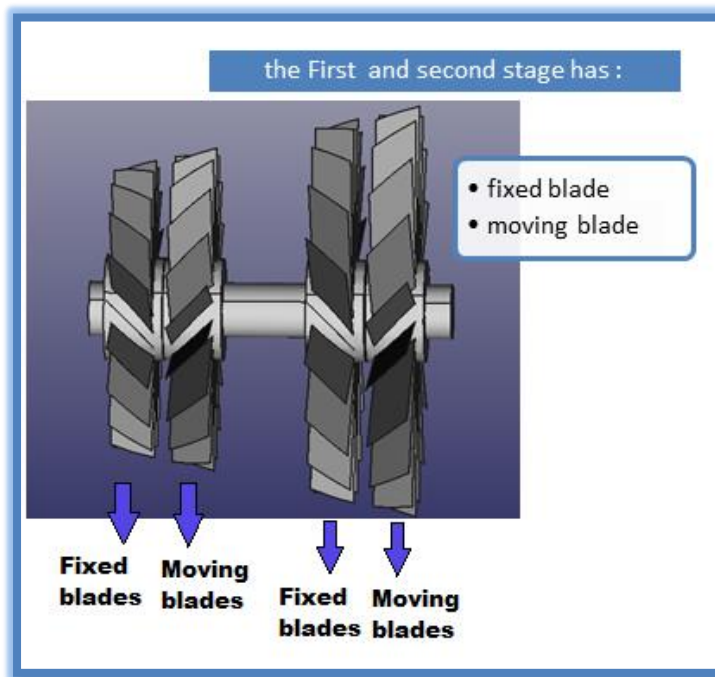


Figure III-4: design with annotation

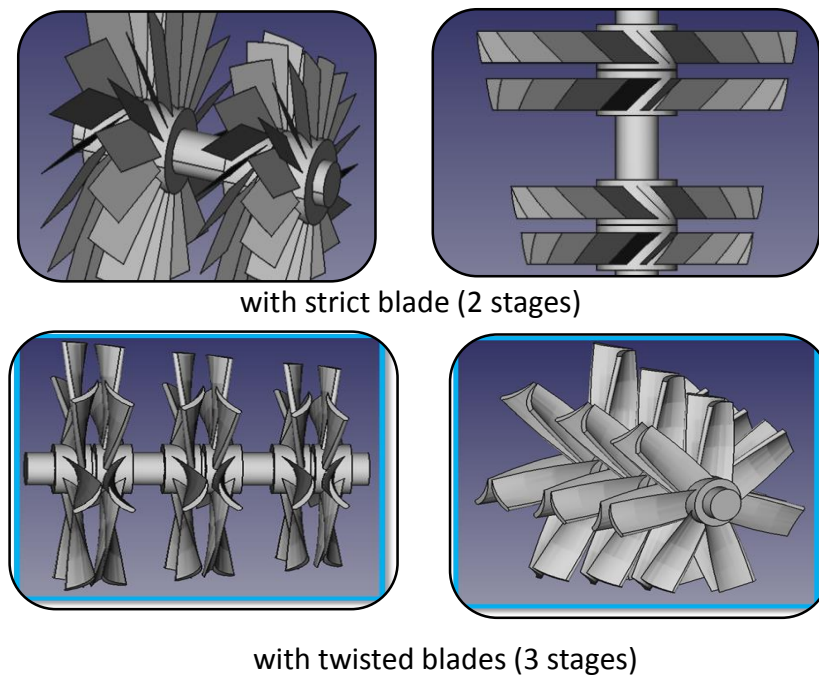


Figure III-5: preliminary designs for stator and rotor

III.1 Blade design

In Free CAD, after doing two sketch up and down then joining them by a line, the blade form is designed.

The blade terminology are shown in Figure III-6.

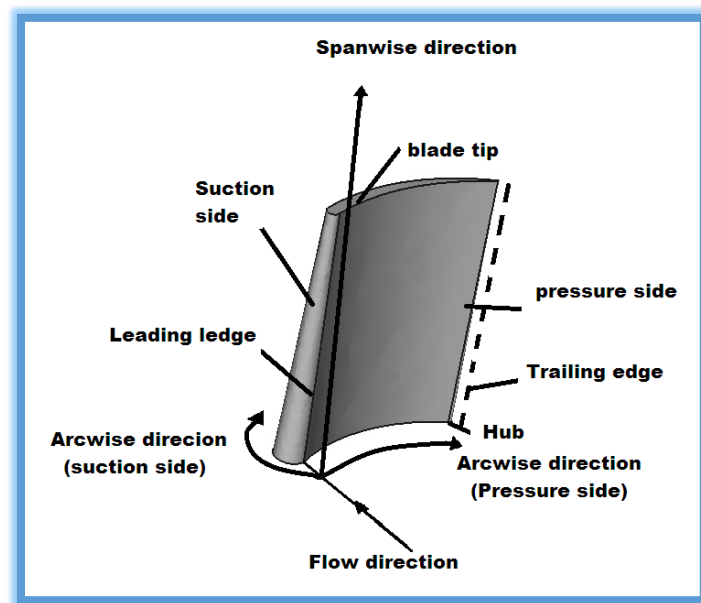


Figure III-6: Blade terminology

The sketch used for the blade design is as shown in figure III-7:

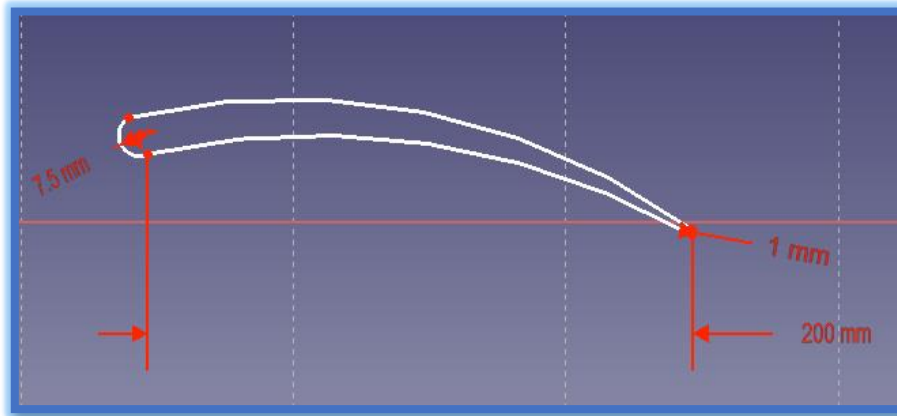


Figure III-7: blade sketch

The height of blades for each stage is in Figure III-8.

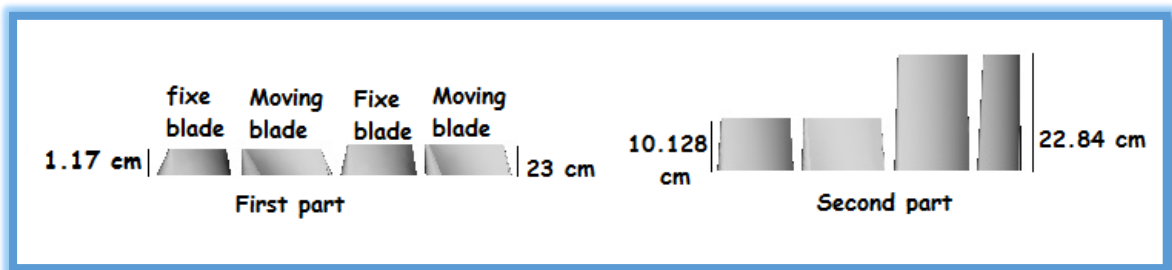


Figure III-8: Height of blades

Blade design in 3D view for each stage after knowing it height are in Figure III-9.

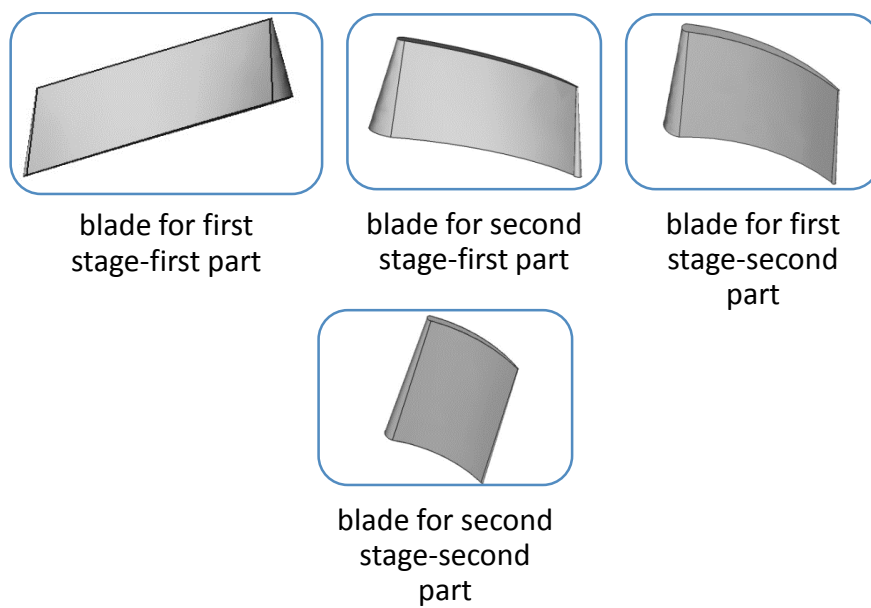


Figure III-9: blades design

III.2 Shaft design

The design of shaft where the blades are putted is in Figure III-10.
The shape is cutting as this way for putting blades and make a distance between stages.

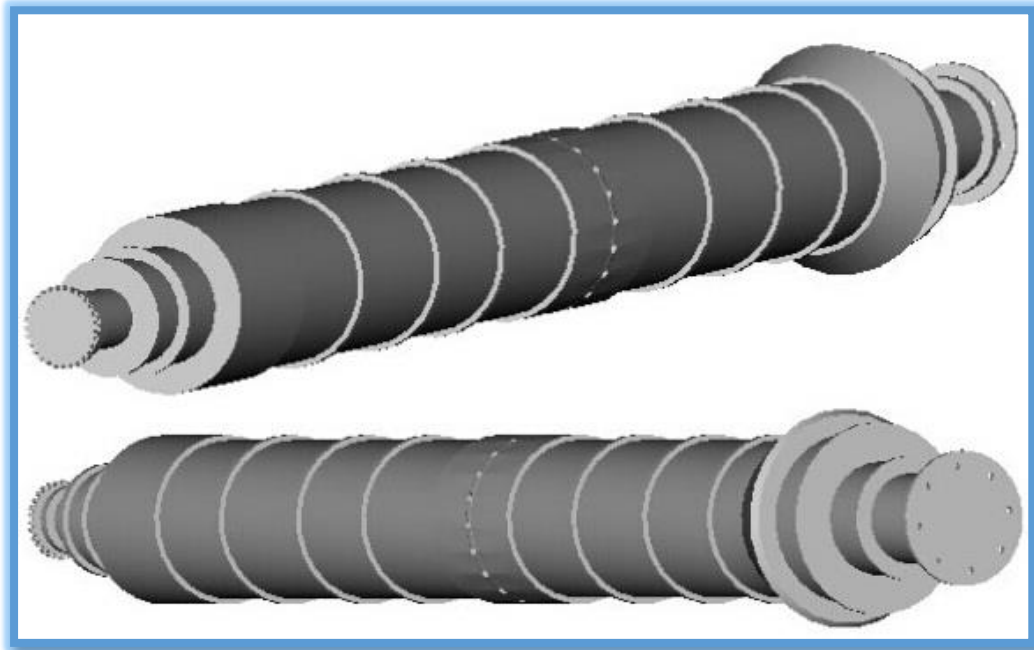


Figure III-9: shaft

The height of this shaft after designed it is about 3 m.

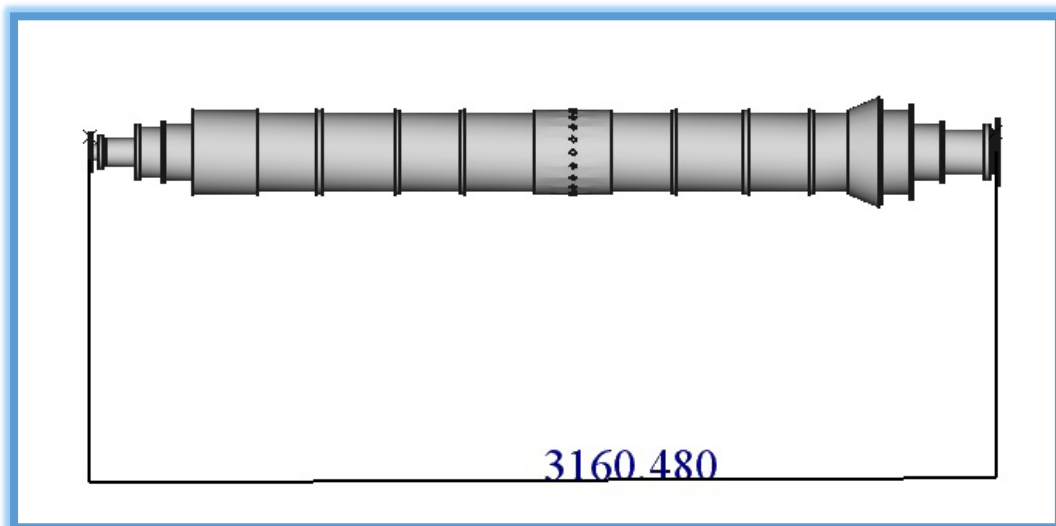


Figure III-10: shaft height (in mm)

III.3 Stator and rotor design

After knowing the number of blades for each stage ($=n$), make an array around the cylinder designed for fixed blades which is putted on the shaft.

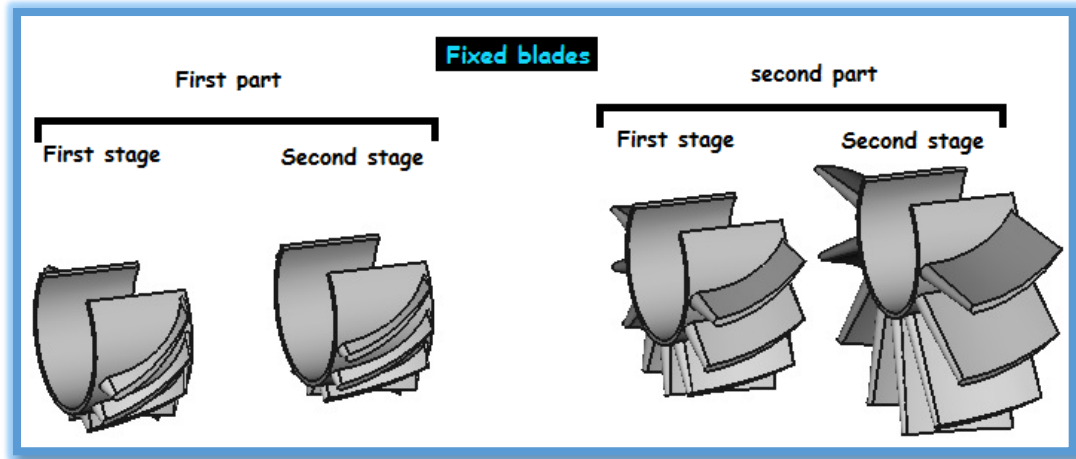


Figure III-11: Fixed blades

There is a cover for fixed blade because they not rotate. And usually it is cutting into two parts which are joined with long screws. First parts are putting down then the shaft including moving parts. Finally the second parts of fixed stages are putting up. This two parts, after joining, make a shape of 360°.

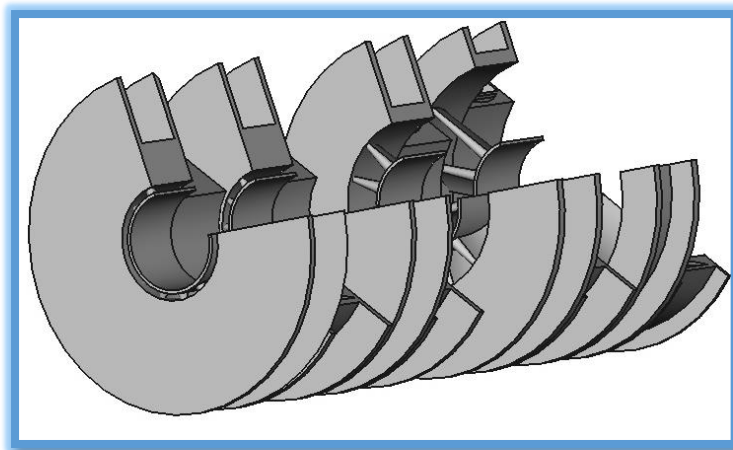


Figure III-12: fixed blades with cover

Also, after knowing the number of moving blades, make an array for each stage. In this case, the shaft is where they are putted.

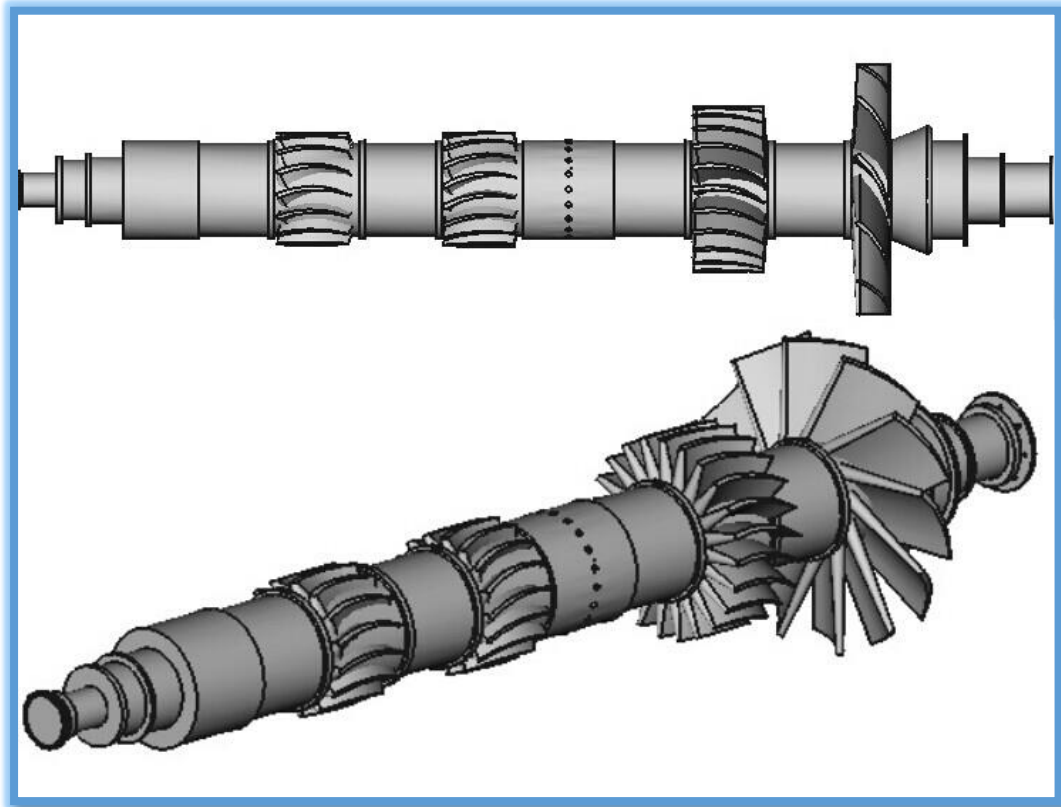


Figure III-13: moving blades

In Figure III-14 cover for moving blades have a distance between the array of blades, (to be able to move) which are putting in the case in last.

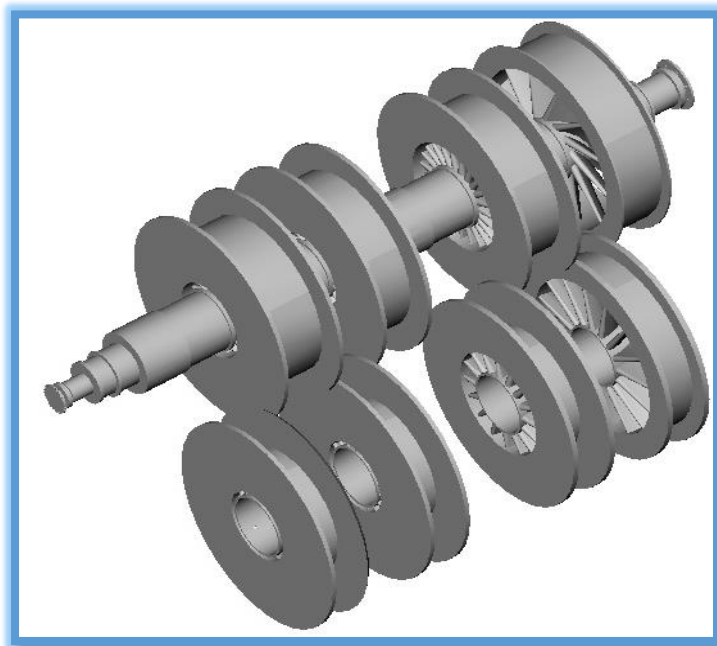


Figure III-14: stator and rotor

Then fixed and moving stages are putting along the shaft.

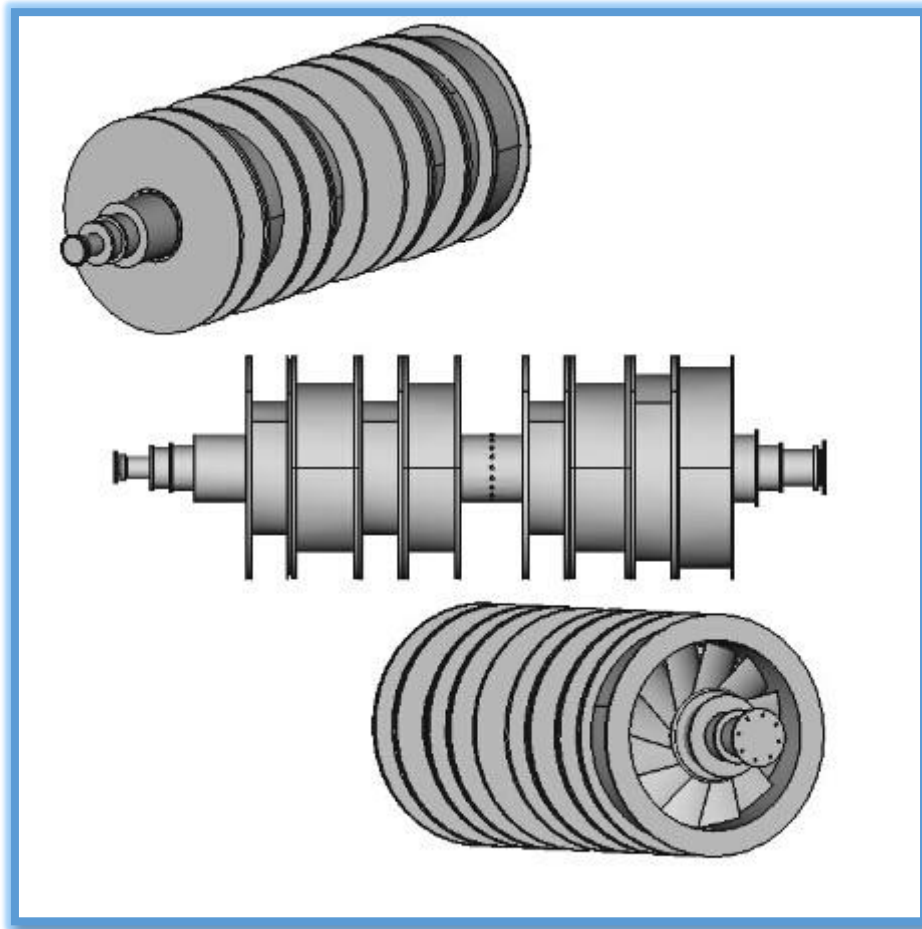


Figure III-15: the entire design

The following paragraph, contains design for another applications of a steam turbine, which are produce the same power but hypotheses are different then the preview design. And the rhythm of calculation are the same.

III.4 Other applications

For simplify the manufacturing, I did other application based on other utilization of data.

This turbine is composed of two stages.

The electric efficiency is equal to 0.96; mechanic efficiency is equal to 0.95; interne efficiency of turbine is equal to 0.9.

Entry pressure for the turbine is 14 bars, and the entry temperature is 195°C. The pressure in the condenser is 1 bar.

So the same steps are followed to obtain the desired design

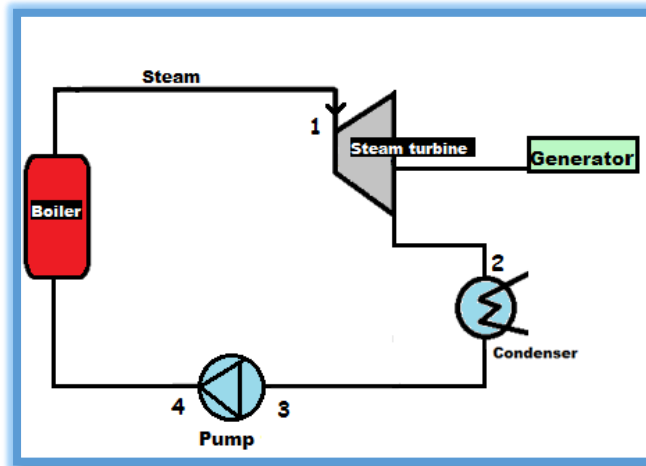


Figure III-16: Installation (one part of a steam turbine)

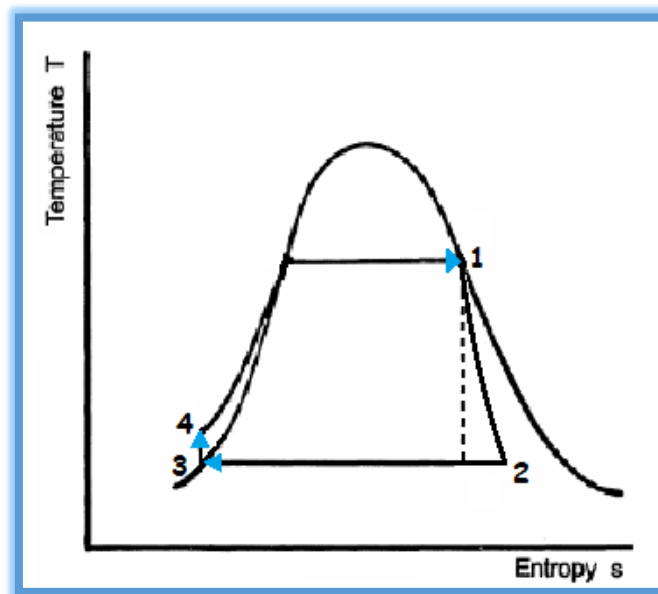


Figure III=17: Rankin cycle (saturated)

Chapter III: Detailed design

Values of h, S, P, and T, and V after calculations, for points 1, 2, 3, and 4 are in Table 11.

Table 11: Values of h, S, P, T, and V for each point

Parameters Points	h (kJ/kg)	S (kJ/kg.K)	P (bar)	T (°C)	V (m ³ /kg)
1 (saturated steam)	2789.6	6.4685	14	195.1	0.1408
2 (chute)	2387.8589	6.58	1	99.64	1.45
3 (condensing)	418	1.3038	1	99.64	0.001
4 (after pump)	419.3	1.3038	-	-	0.001

§ Thermal efficiency: $\eta_{th}=16.89\%$

§ Mass flow: $Q_m=3.996\text{ kg/s}$.

- ◆ Put all specific points on the Mollier chart. (Figure III-18);
- ◆ Take: $h_0 - h_{1s} = h_{1s} - h_{2ss} = h_{2ss} - h_{3sss} = h_{3sss} - h_{4(4s)} = \frac{h_0 - h_{4(4s)}}{4}$.
- ◆ h_0 and h_4 in Figure III-18 are same as h_1 and h_2 which are utilized before.
- ◆ h and V are known from Figure III-18.
- ◆ Results are putted in Table 12:

Table 12: values of different parameters for points in Mollier chart

parameter point	h (kJ/kg)	P (bar)	X (percentage)	V(m ³ /kg)
0	2789.6	14	(saturated)	0.1408
1 ^s	2678.45	8		
1	2689	8	0.96	0.23044
2 ^s	2577.5	4.3		
2 ^{ss}	2567.3	4.3		
2	2590	4.3	0.928	0.379
3 ^{sss}	2455.15	2		
3	2483.3	2	0.898	0.629
4 ^(4s)	2345	1		
4	2387.858 9	1	0.87	1.262

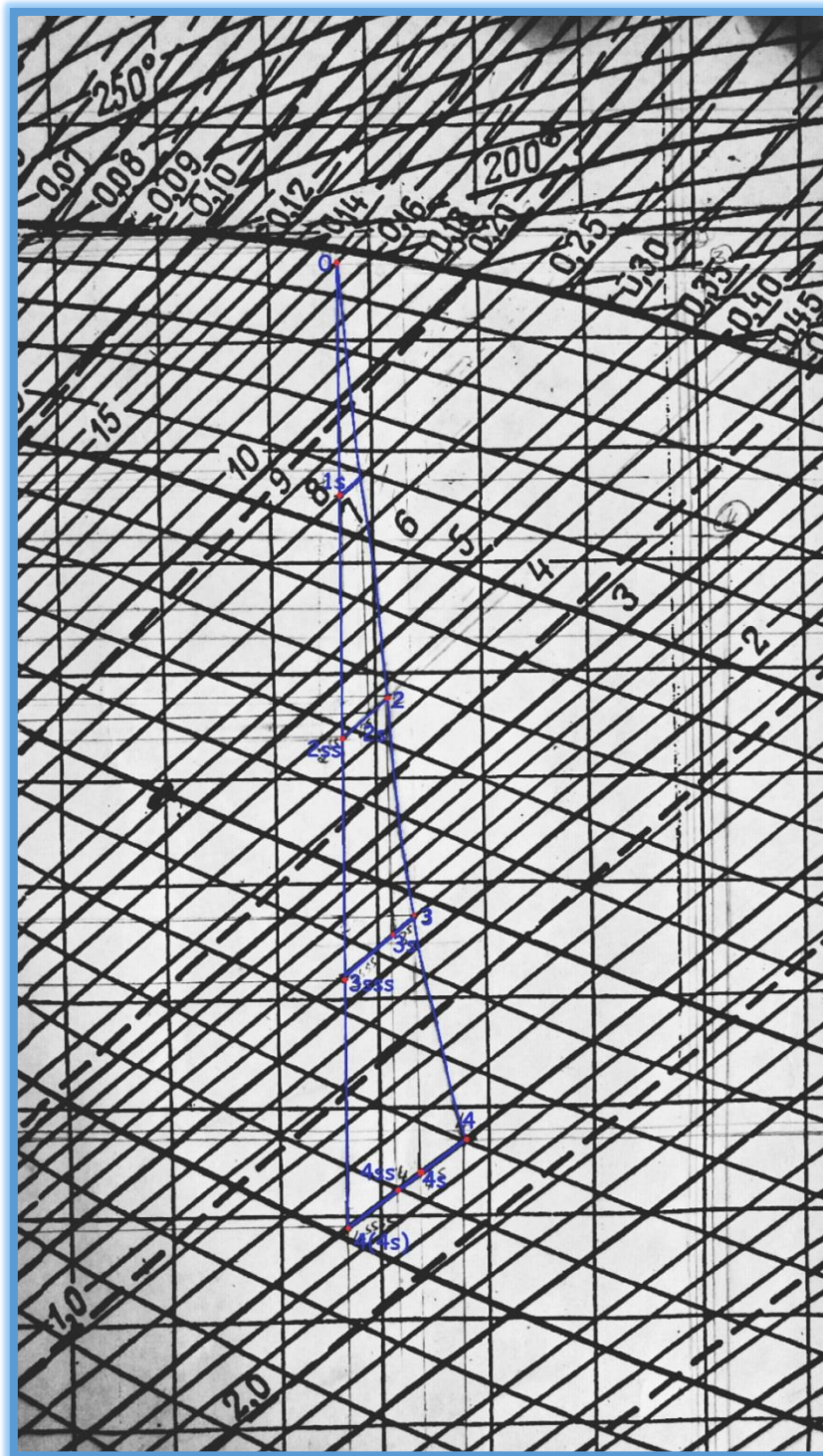


Figure III-18: Condition curve for a two-stage reaction turbine, drawn on the Mollier (enthalpy-entropy) chart for this application

- ◆ Calculation of velocities and angles is illustrated in Table 13 and Table 14.

Table 13: Calculation of velocities and angles for first stage

Parameter	Value
debit (Kg/s)	3.996
delta h fixe (KJ/Kg)	100.6
delta h mobile (KJ/Kg)	99
Vs1 (absolute velocity)(m/s)	448.5532298
diameter (m)	0.15
N (round per minute)	3000
VB (velocity of blade) (m/s)	28.11725425
teta1 (degree)	30
tan ϕ 1	0.622400634
ϕ 1 (rad)	0.556727924
ϕ 1 (degree)	31.89816041
Vr1 (relative velocity)(m/s)	424.4358699
δ 1 (degree)	90
Vr2	424.6690566
cos(Υ 1)	0.066209802
Υ 1(rad)	1.504538054
Υ 1(degree)	86.20368064
Vs2	423.7372153

Table 14: Calculation of velocities and angles for second stage

Parameter	Value
debit (kg/s)	3.996
delta h fixe (kJ/kg)	106.7
delta h mobile (kJ/kg)	95.4411
Vs3 (absolute velocity)(m/s)	461.9523785
diameter (m)	0.15
N (round per minute)	3000
VB (velocity of blade) (m/s)	30.94468764
teta2 (degree)	30
tan ϕ 2	0.625751954
ϕ 2 (rad)	0.559139864
ϕ 2 (degree)	32.03635434
Vr3 (relative velocity)(m/s)	435.4284738
δ 2 (degree)	90
Vr4	435.6476076
cos(Υ 2)	0.071031465
Υ 2(rad)	1.499704995
Υ 2(degree)	85.92676673
Vs4	434.5471946

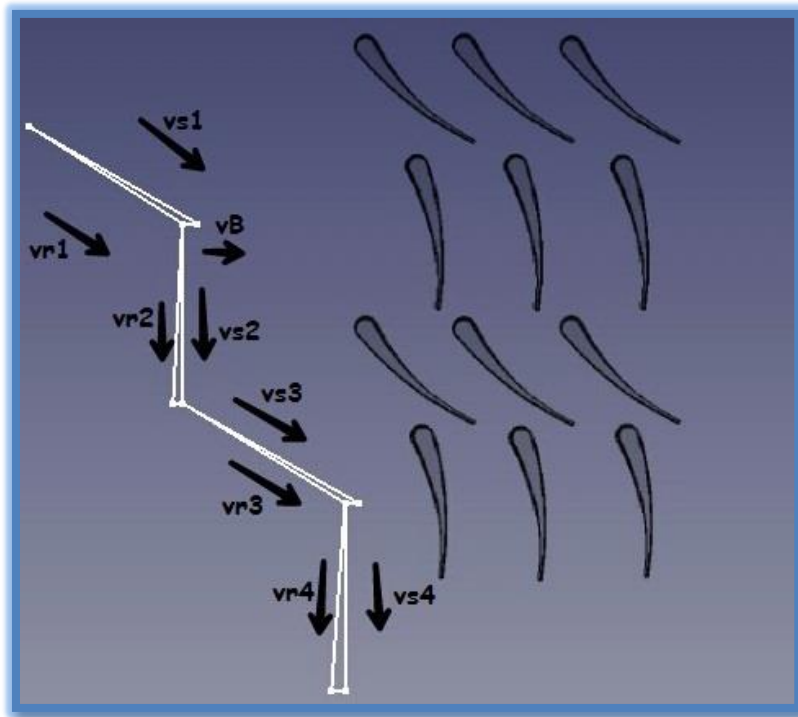


Figure III-19: Velocity diagram for first and second stage

Utilize the same equations in paragraph II.5

- ◆ Calculation of blade's height for first and second stage is illustrated in Table 15 and Table 16.

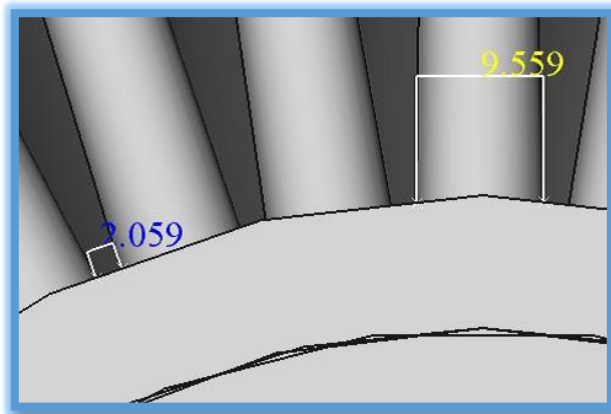


Figure III-20: distance and width for "first stage"

Table 15: calculation of height for first stage

Parameter	Value
v(specific volume)(m ³ /Kg)	0.379
Qv (volume flow rate) (m ³ /s)	1.514484
St(total section)(cm ²)	35.66268784
distance per blades (cm)	0.2059
width (cm)	0.9559
Diameter (cm)	15
Δ	517.8986253
height of blade (cm)	2.292763033
height in m	0.02292763

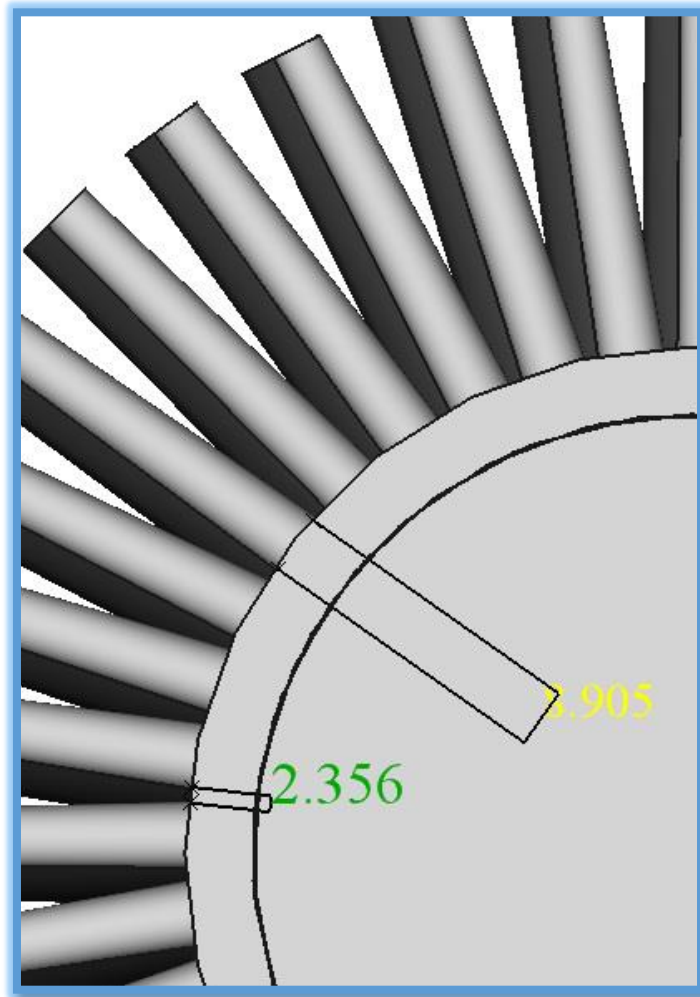


Figure III-21: distance and width for second stage

Chapter III: Detailed design

Table 16: calculation of height for second stage

Parameter	Value
v (specific volume)(m^3/Kg)	1.262
Q_v (volume flow rate) (m^3/s)	5.042952
St (total section)(cm^2)	115.7575966
distance per blades (cm)	0.2356
width (cm)	0.8905
Diameter (cm)	15
Δ	1551.855692
height of blade (cm)	4.700553704
height in m	0.047005537

Figure 22 contains the sketch of blade up and down.

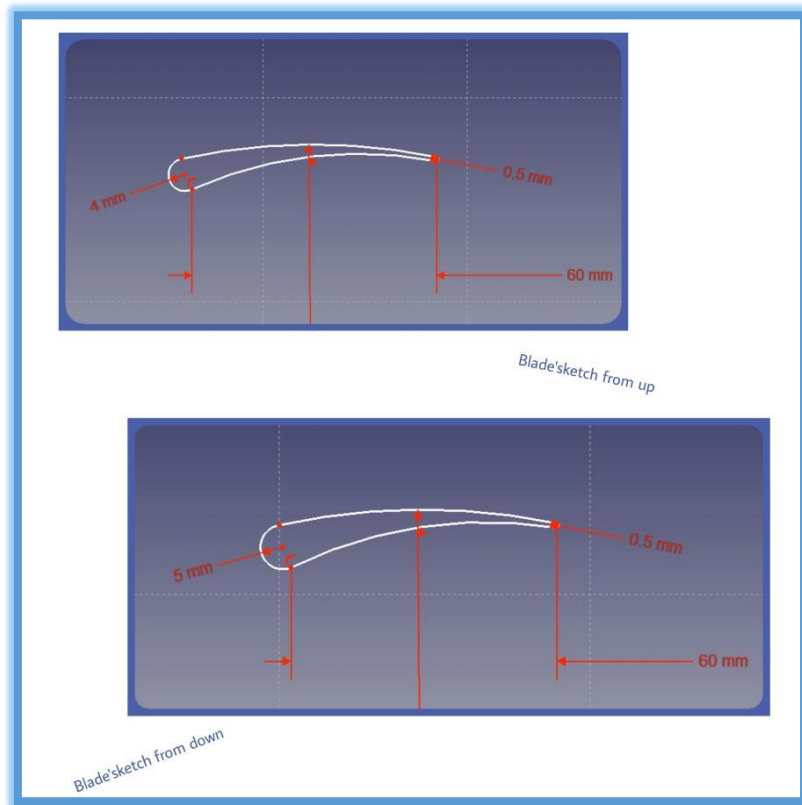


Figure III-22: blade 'sketch

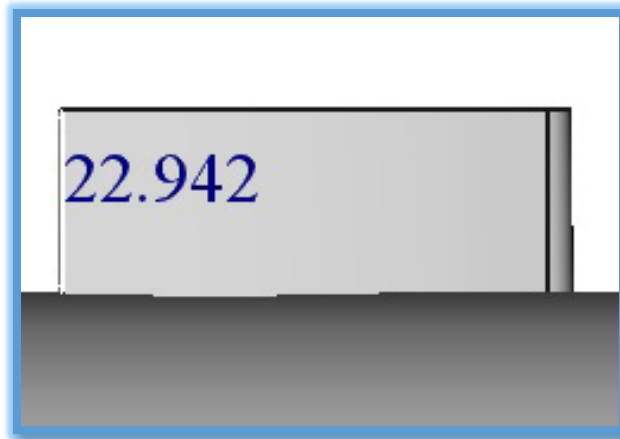


Figure III-23: Height of blade from first stage (in mm)

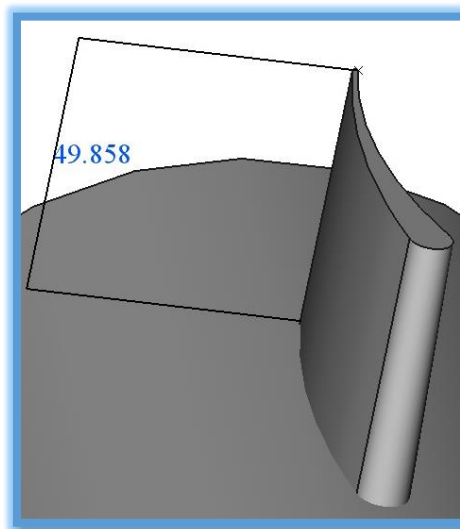


Figure III-24: height of blade for second stage (in mm)

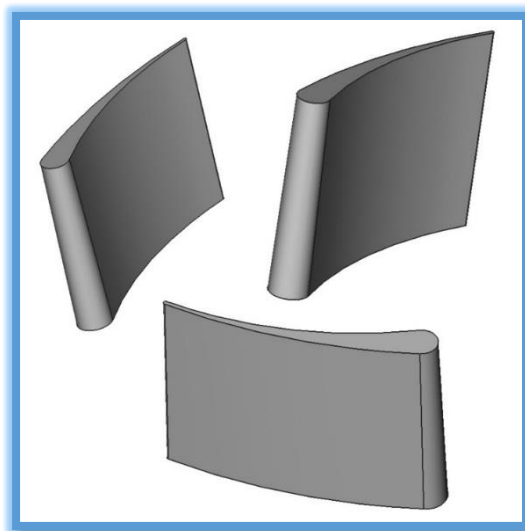


Figure III-25: blades design

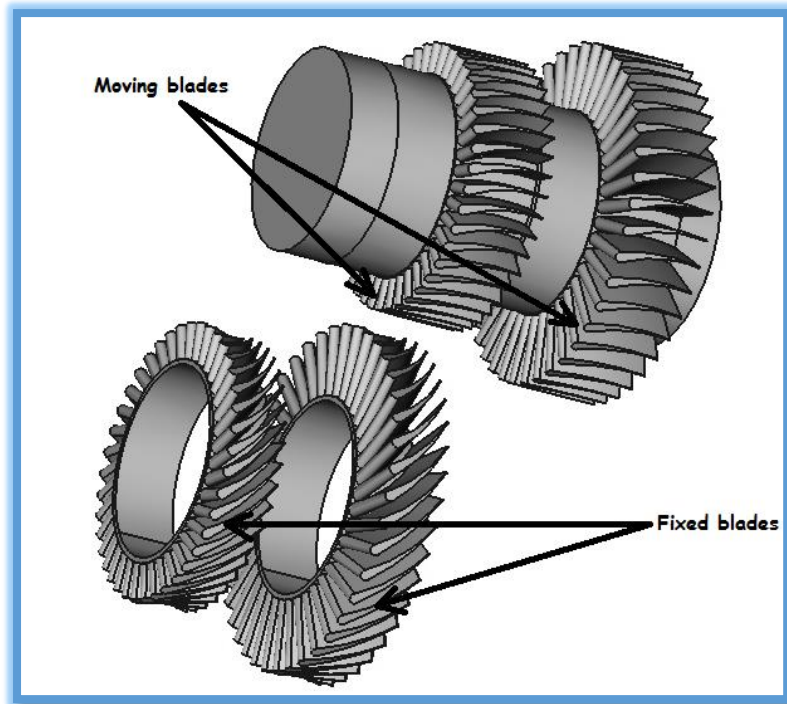


Figure III-26: fixed and moving blades

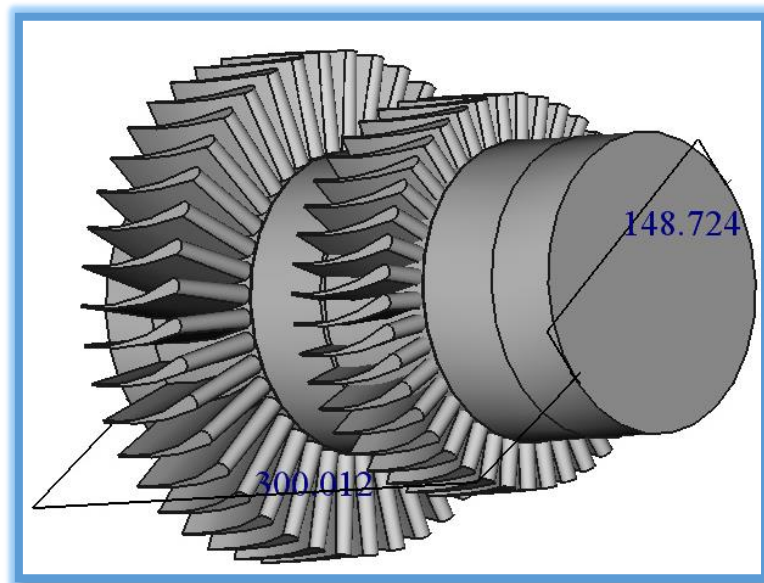


Figure III-27: diameter and height for the shaft (in mm)

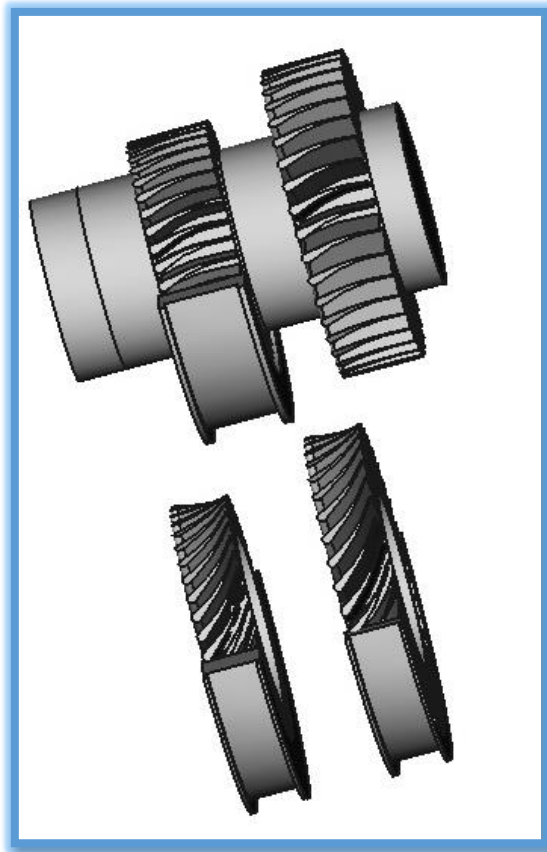


Figure III-28: blades with their cover

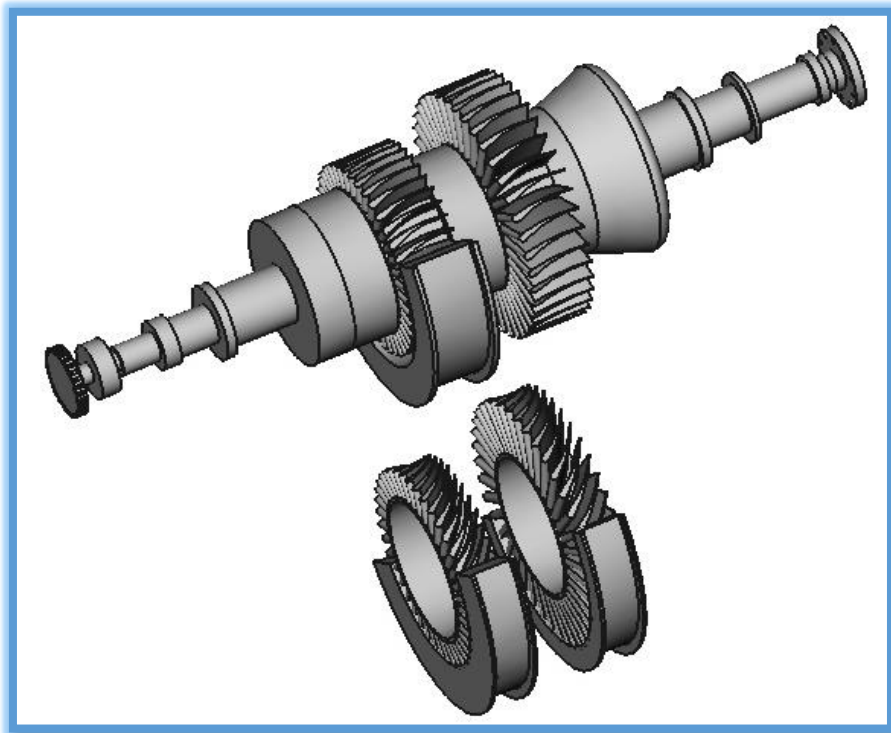


Figure III-29: blades with shaft and their cover

Chapter III: Detailed design

After detailed sketch and many steps, the case design that I reached is in Figure III-30.

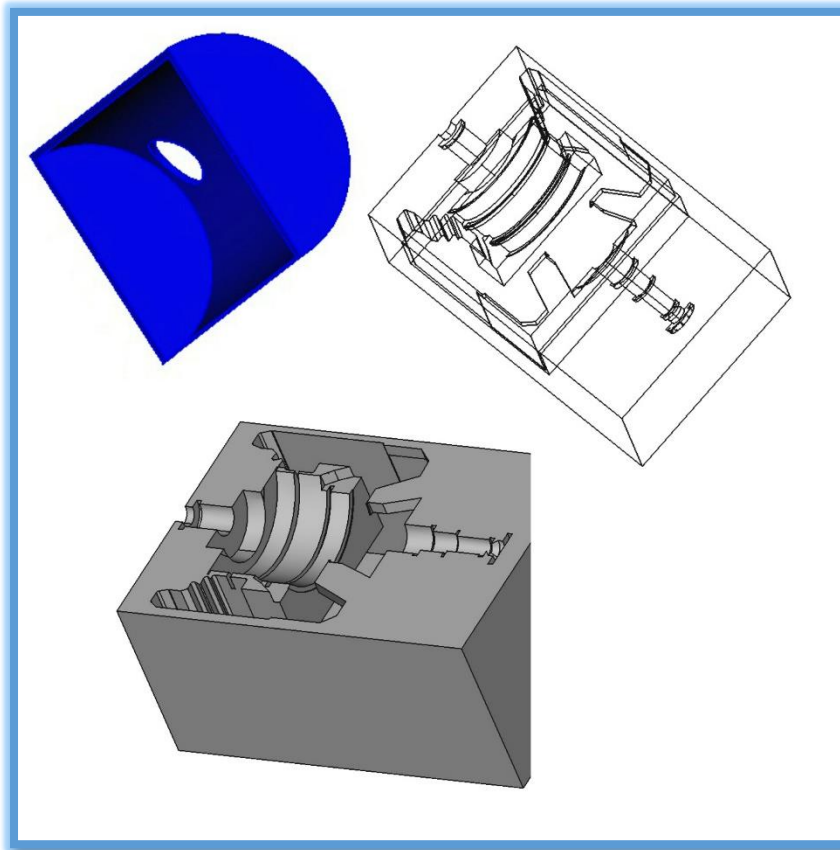


Figure III-30: case

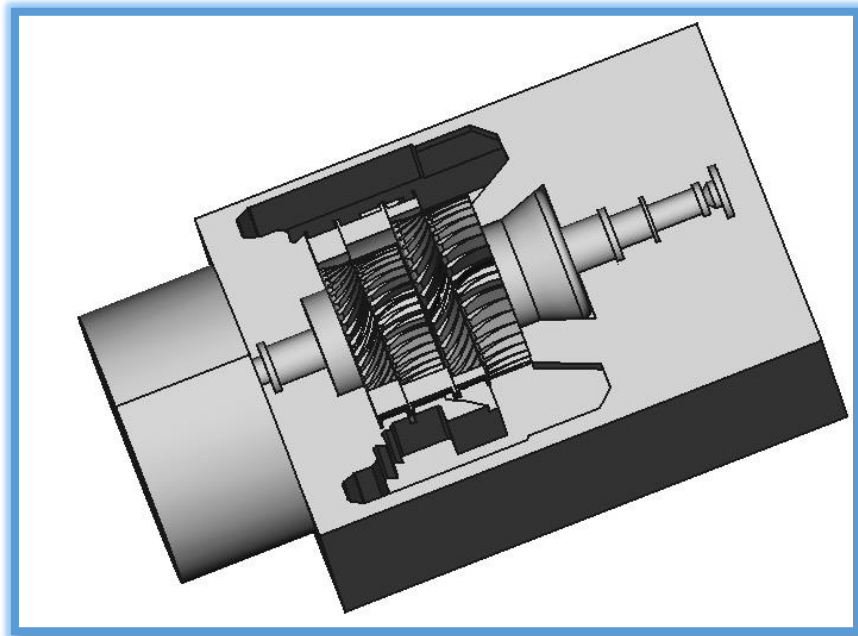


Figure III-31: blades, shaft, and casing

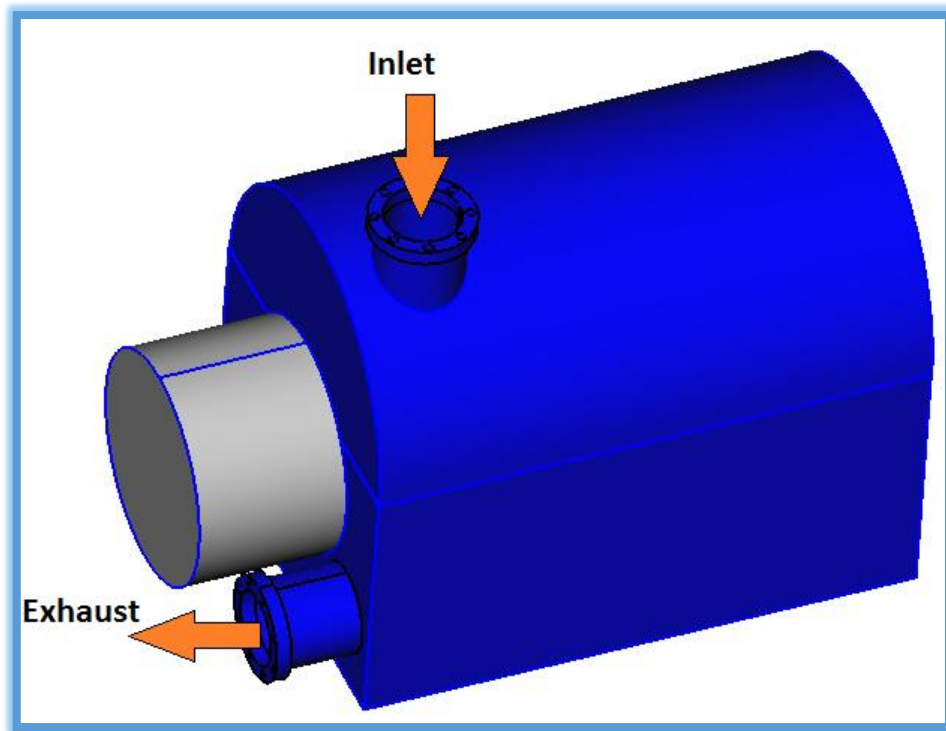


Figure III-32: the entire design of the steam turbine

The next application has a same hypotheses, but the diameter is bigger, which is equal to 40 cm. so after all calculations and main design, we reached the design in figures below.

In this application, we make a stand for blades.

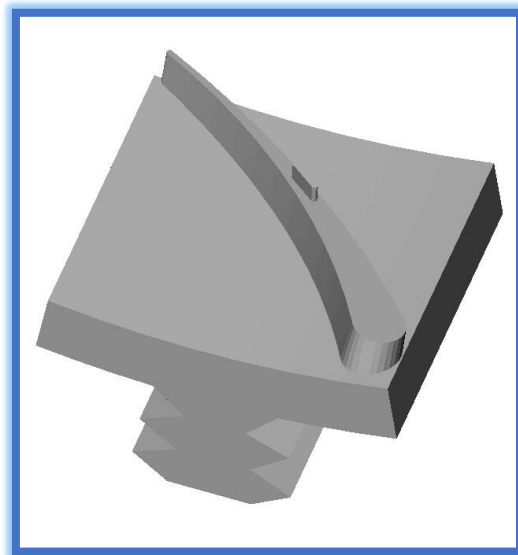


Figure III-33: design for fixed blade-first stage

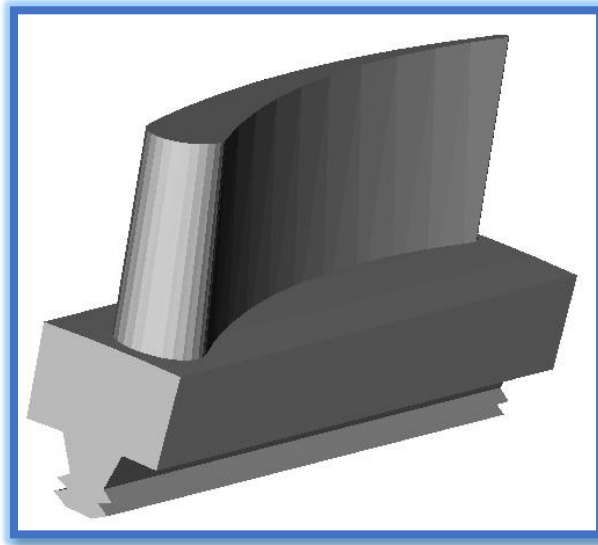


Figure III-34: design for moving blade-second stage

Those two blades have a length equal to 0.81 cm.

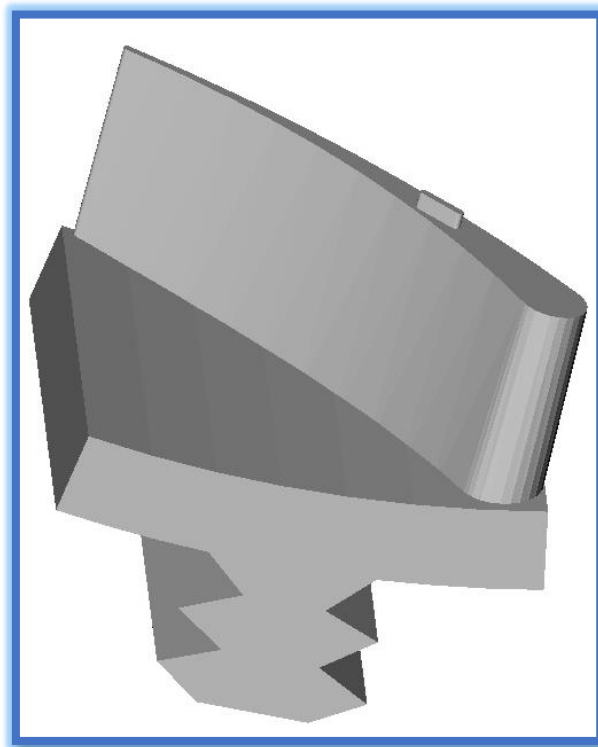


Figure III-35: design for fixed blade-second stage

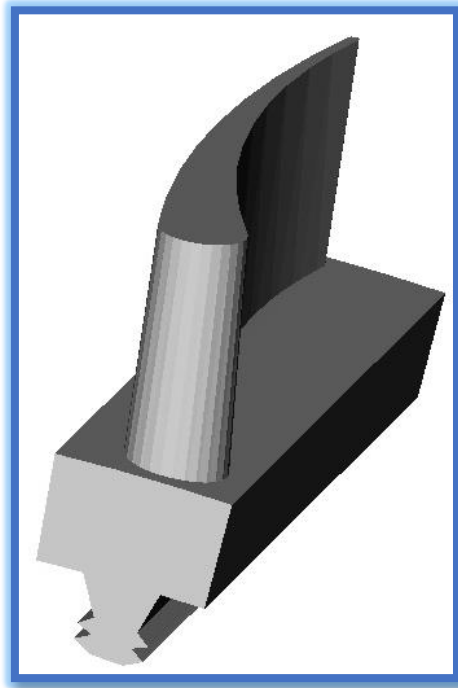


Figure III-36: Design for moving blade-second stage

Those two blades have a length equal to 1.68 cm.
All blades design are putted in the cylinder which is cutting in same shape of stand.

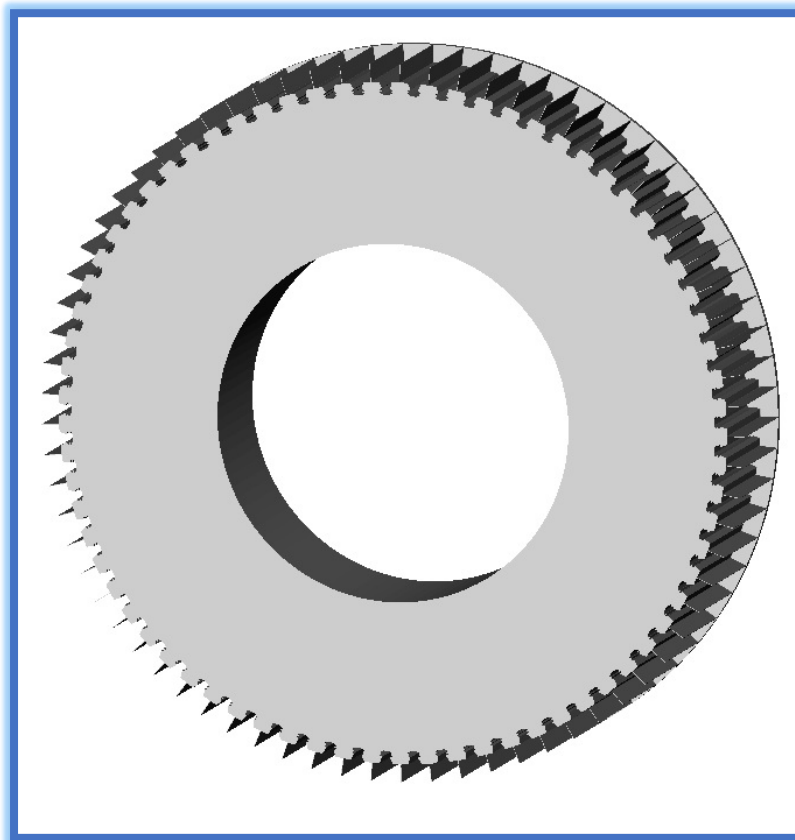


Figure III-37: cylinder for moving blade-first stage

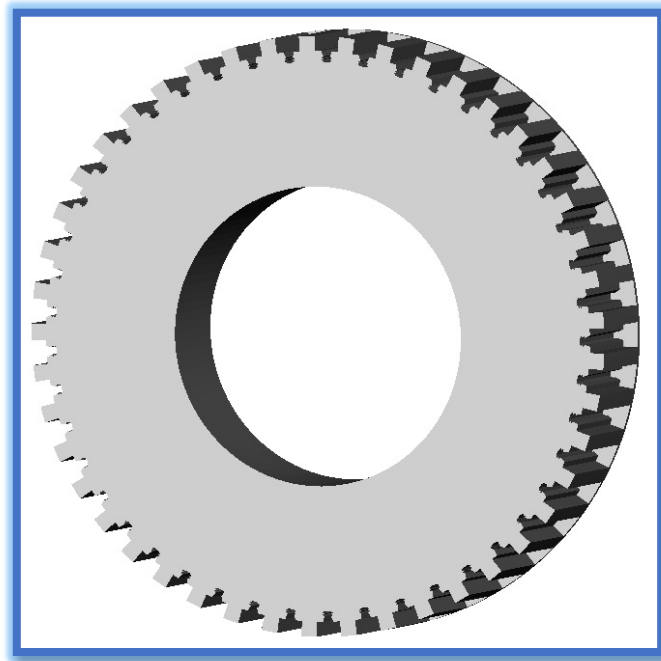


Figure III-38: cylinder for moving blade-second stage

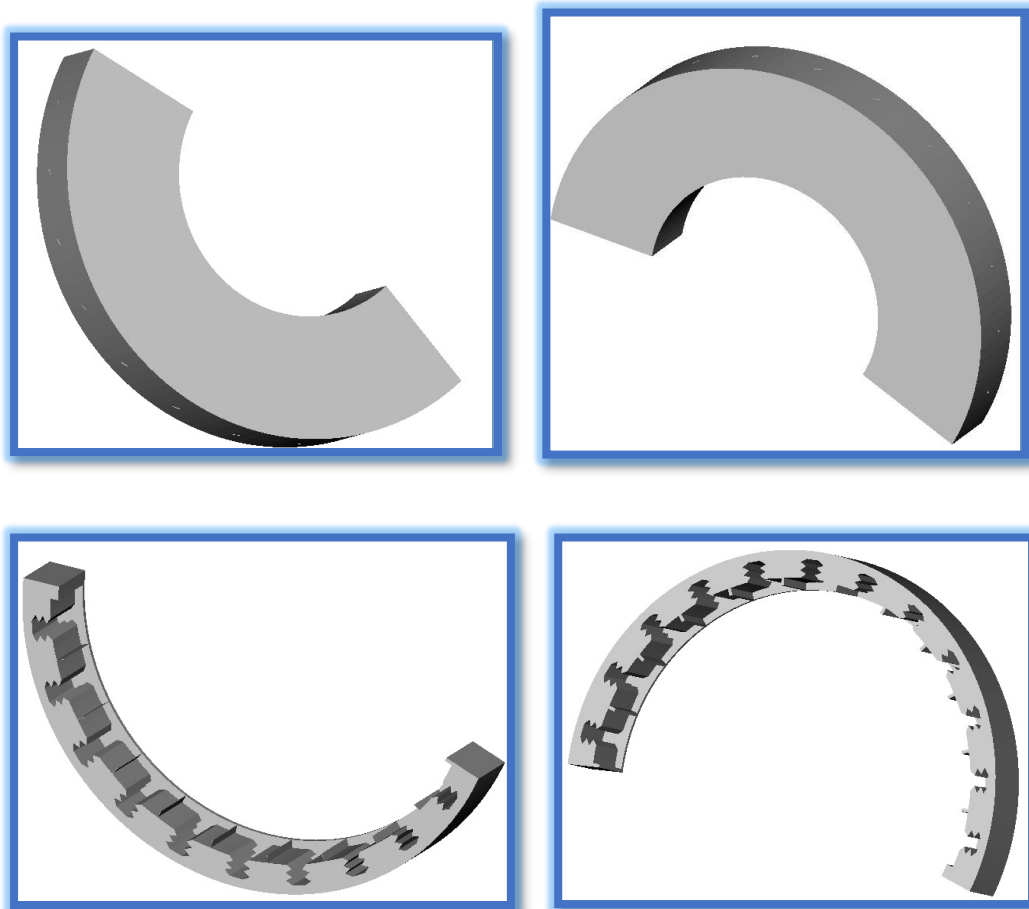


Figure III-39: covers for fixed stage up and down

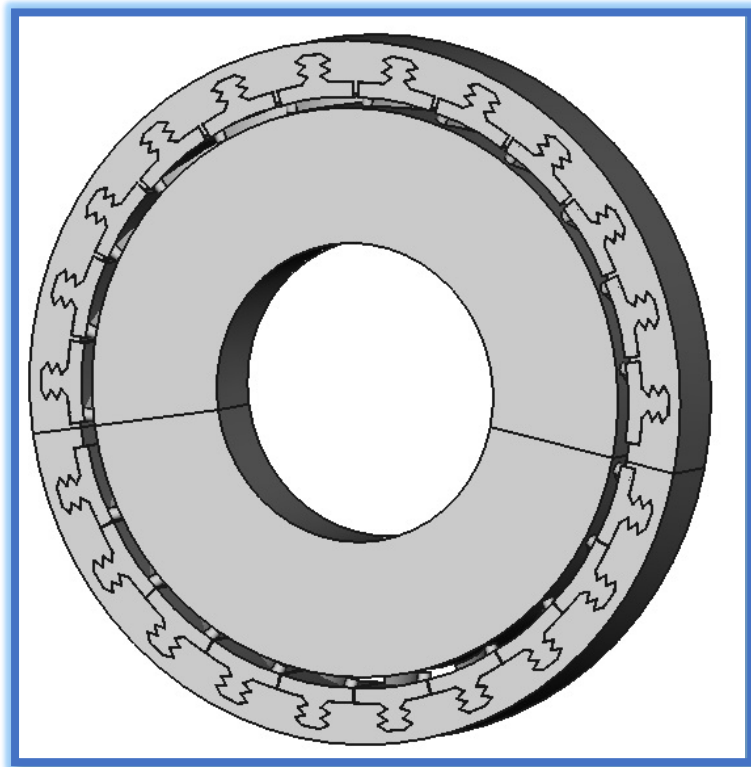


Figure III-40: fixed blade with its covering-first stage

For the second stage, it is the same one but with another length for blades. Moving blades for first and second stage are putting in the shaft.

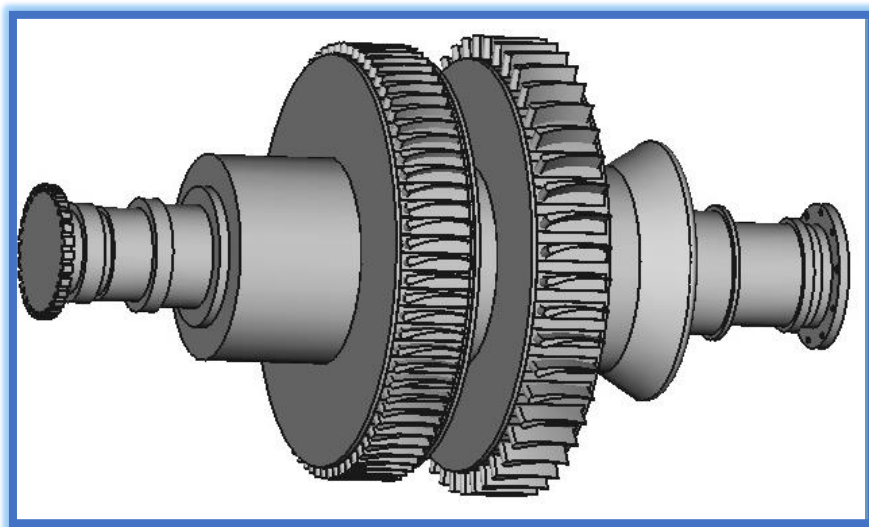


Figure III-41: moving blades on the shaft

It is clear that when we change some inlet parameter as diameter, all other values change, for example the height of blades decreases.

Almost, in mechanical product the regulation system is important. So this is illustrate in the next chapter. Where we talk about three devices for regulation and its mechanism.

Chapter IV: Control system

After manufacturing any device, we should do the system of controlling which regulates all inlet values, instead of fixed it constant.

So in our case, a steam turbine also has a system controller.

Usually, system controller can be electrical or mechanical, but for an easy maintenance, and long lifetime we use the mechanical one.

In next paragraph, we talk about three devices, governor, steam control, and steam separator.

IV.1 Governor

Steam turbine governing is using to control the flow rate of steam so as to maintain its speed of rotation as constant. If there is variation in load during the operation of a steam turbine, this problem can have a significant impact on its performance. The primary objective in the steam turbine operation is to stay on a same speed of rotation during its work irrespective of the varying load. This can be achieved by governor in a steam turbine.

In throttle governing the pressure of steam is reduced at the turbine entry thereby decreasing the availability of energy. In this method steam is passed through a restricted passage thereby reducing its pressure across the governing valve. (14) The flow rate is controlled using a partially opened steam control valve. The reduction in pressure leads to a throttling process in which the enthalpy of steam remains constant. (15)

In our application we choose 3000 round per minute.

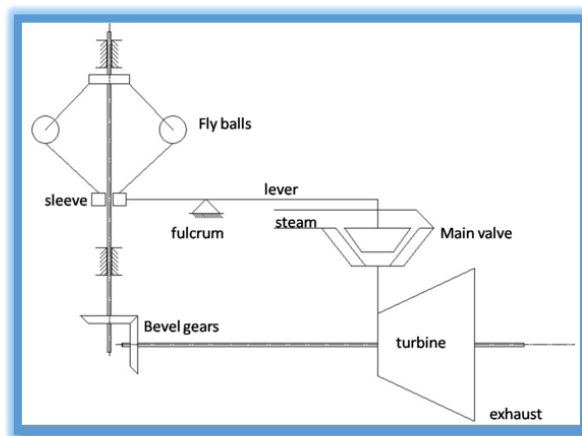


Figure IV-1: schematic of throttle governor

Low initial cost and simple mechanism makes throttle governing the most application method for small steam turbines. The mechanism is illustrated in Figure IV-1. The valve is actuated by using a centrifugal governor which consists of flying balls attached to the arm of the sleeve. A geared mechanism connects the turbine shaft to the rotating shaft on which the sleeve reciprocates axially.

Chapter IV: Control system

With a reduction in the load the turbine shaft speed increases and brings about the movement of the flying balls away from the sleeve axis. This result in an axial movement of the sleeve followed by the activation of a lever, which in turn actuates the main stop valve to a partially opened position to control the flow rate. (14)

IV.2 Steam controller



Figure IV-2: steam controller

This is an aero flow steam conditioning valve, zero leak technology, it is easy to maintenance it.

Steps of working for this steam controller:

- ◆ When the steam flow begins, pilot plug opens allowing steam flow
- ◆ Then the main plug opens
- ◆ Steam flows through characterized cage.
- ◆ When the main plug opens more, steam flows through drilled cage.
- ◆ Then, water enters Laval steam assist nozzle. Water mixes with steam flowing through center of chambers to atomize water (water enters upper chambers). Steam enters lower chambers.
- ◆ Steam creates swirl effect.
- ◆ Steam assist Laval nozzle reduces downstream piping.

The water is for cooling the steam and make it at the same entering temperature.

IV.3 Steam Separators

In some steam systems, wet steam is generated. This wet steam contains water droplets that can reduce the effectiveness of the steam system.

Chapter IV: Control system

Water droplets erode piping elbows, turbine blades and passages, and pressure reducing valves, thus reducing efficiency and life. Furthermore, liquid water can significantly reduce heat transfer rates in heat exchange components, as well as result in water hammer.

Removing water droplets before they reach end-use equipment is necessary.

Steam separators remove water droplets, generally relying on controlled centrifugal flow. This action forces the entrained moisture to the outer wall where it is removed from the separator. The means of moisture removal could be a steam trap or a drain. Some manufacturers include the trap as an integral part of the unit. Additional accessories include water gauge connections, thermometer connections, and vent connections.

Steam separators can be installed in either a horizontal or vertical line. They are capable of removing 99% of particulate entrainment 10 microns and larger over a wide range of flows. (16)

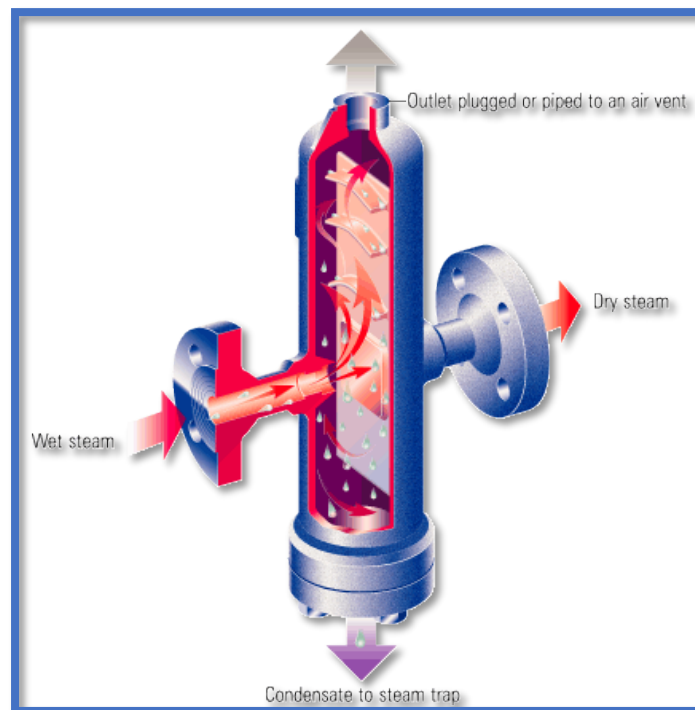


Figure IV-3: steam separator

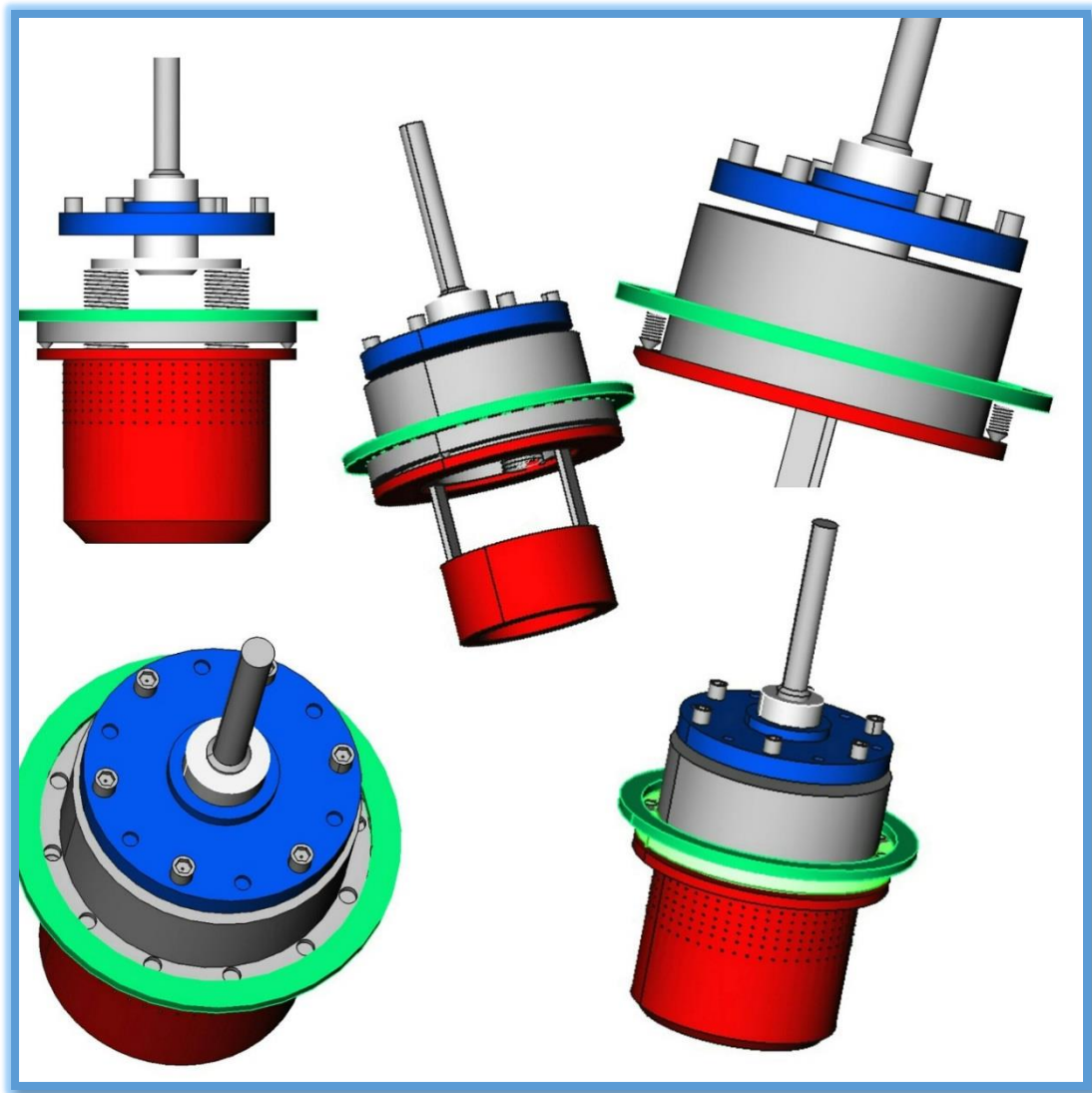


Figure IV-4: steam controller design in Free CAD

Each controller has its own design. Also, they differ in their method of working. Generally, there are many types of governor; we utilized the governor for small turbines. But for big turbines, another one is utilized.

After any result, it is obvious to discuss our results; this discussion is cleared and resumed in the next chapter.

Chapter V: Results and discussion

First of all, it must have the shape of the blades, which are similar as an airfoil, and after several calculation, we find values of length for blades. These are for the first application: 1.17 cm for first part-first stage, 2.29 cm for first part-second stage, 10.12 cm for second part-first stage, and 22.8 cm for second part-second stage. So it is clear that, for high pressure stages, the length is bigger than others.

After knowing the length, fixed and moving blades are designed and put it on the shaft, then bearings are designed.

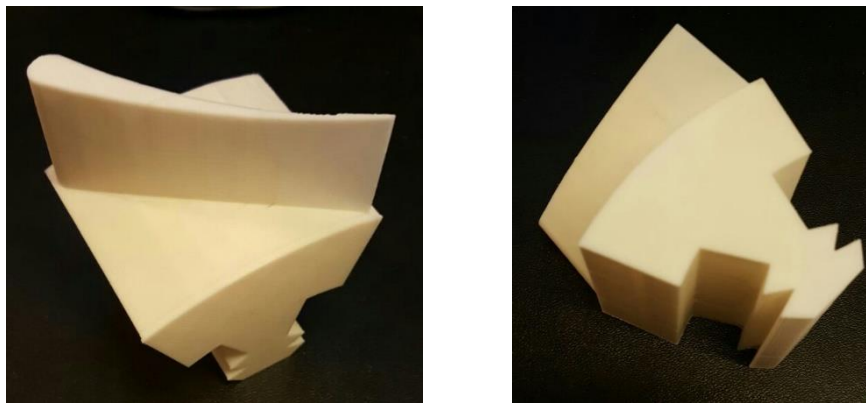
Finally, make a design of the case where the body of a steam turbine is putted. This case must don't have any vacuum between bearings and case, because steam is a sensitive fluid which has its own pressure.

If we compare the first and the second steam turbine, we find that when there is not superheated steam and also not many stage, the thermal efficiency decrease.

The height of blades changes when we change the diameter of shaft, difference of fixed and moving enthalpy, distance per blades, and width of the blade...

When the diameter is bigger, then the height of blades decreases.

For simplify the manufacturing of steam turbine, a stand for each blade must designed, for enters it easily to the shaft .So after designed it in the 3D software, it was printed in CNC machine. A prototype is shown in figure



FigureV-1: blade printed from CNC machinery

General conclusion

Land pollution or trash are distributed in diverse region in Lebanon, so we must find a fast solution for this disaster issue. Incinerate all those waste is useful for heating the water in the incineration power plant. So, those garbage replace fossil fuel which is usually expensive, but there is huge quantity of trash daily.

To complete all parts of our company, we going to manufacture the steam turbine which was designed, when we will have enough budget. And it is need more time, because, if it is not working for the first time, we must repair it to know where the fault is. So that is not easy at all.

And the material which is used should have the ability to withstand high temperatures, no flexible, and don't face a stress corrosion problem. That for maximize the life time of the steam turbine, or don't lose it. So this opens approved to another research topic.

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Table A-4b Saturated steam: pressure table (SI units)

bar	Pressure, psia	Temperature, °C	Specific volume, m ³ /kg			Specific enthalpy, kJ/kg			Specific entropy, kJ/(kg)(K)		
			v_f	v_g	$v_g - v_f$	h_f	h_g	$h_g - h_f$	s_f	s_g	$s_g - s_f$
0.010	0.1450	6.98	0.0010001	129.20	29.30	2484.9	2514.2	0.1034	8.8714	8.9748	
0.015	0.2176	13.04	0.0010007	87.98	54.71	2470.6	2525.3	0.1958	8.6312	8.8270	
0.020	0.2901	17.51	0.0010014	67.00	73.48	2460.0	2533.5	0.2569	8.4659	8.7228	
0.025	0.3626	21.08	0.0010021	54.24	88.49	2451.6	2540.0	0.3083	8.3340	8.6423	
0.030	0.4351	24.10	0.0010028	45.66	101.05	2444.5	2545.5	0.3510	8.2258	8.5768	
0.040	0.5802	28.98	0.0010041	34.81	121.46	2432.9	2554.4	0.4197	8.0541	8.4738	
0.050	0.7252	32.90	0.0010053	28.19	137.82	2423.7	2561.5	0.4740	7.9203	8.3943	
0.060	0.8702	36.16	0.0010064	23.74	151.50	2415.0	2566.9	0.5191	7.8105	8.3296	
0.070	1.0153	39.03	0.0010075	20.53	163.43	2409	2572.4	0.5591	7.7149	8.2740	
0.080	1.1603	41.54	0.0010085	18.10	173.9	2402.6	2576.5	0.5915	7.6364	8.2279	
0.090	1.3053	43.79	0.0010094	16.20	183.3	2396.7	2580.0	0.6225	7.5635	8.1860	
0.10	1.4504	45.84	0.0010103	14.68	191.9	2392.3	2584.2	0.6488	7.5006	8.1494	
0.11	1.5954	47.72	0.0010111	13.40	199.7	2388.3	2588.0	0.6740	7.4420	8.1160	
0.12	1.7405	49.45	0.001012	12.36	207.1	2383.5	2590.6	0.6964	7.3891	8.0855	
0.14	2.0305	52.58	0.001013	10.69	220.3	2375.8	2596.1	0.7371	7.2964	8.0317	
0.16	2.3206	55.34	0.001015	9.433	231.9	2369.1	2601.0	0.7728	7.2124	7.9852	
0.18	2.6107	57.82	0.001016	8.445	242.4	2362.9	2605.3	0.8045	7.1397	7.9442	
0.20	2.9008	60.09	0.001017	7.649	251.9	2357.4	2609.3	0.8332	7.0745	7.9077	
0.25	3.6259	64.99	0.001020	6.204	272.6	2345.1	2617.7	0.8947	6.9359	7.8306	
0.30	4.3511	69.12	0.001022	5.229	289.9	2334.9	2624.8	0.9458	6.8220	7.7678	
0.40	5.8015	75.88	0.001026	3.993	318.3	2318.0	2636.3	1.0279	6.6413	7.6692	
0.50	7.2519	81.35	0.001030	3.240	341.3	2304.1	2645.4	1.0930	6.5001	7.5931	
0.60	8.7023	85.95	0.001033	2.732	360.6	2292.4	2653.0	1.1471	6.3841	7.5312	
0.80	11.6030	93.52	0.001038	2.087	392.3	2273.0	2665.3	1.2344	6.1994	7.4338	
1.0	14.5038	99.64	0.001043	1.694	418.0	2257.0	2675.0	1.3038	6.0548	7.3580	
1.013	14.696	100	0.001043	1.673	419.5	2256.1	2675.6	1.3079	6.0462	7.354	
1.2	17.4045	104.81	0.001047	1.428	439.7	2243.4	2683.1	1.3617	5.9356	7.2973	
1.4	20.305	109.3	0.001051	1.237	458.6	2231.4	2690.0	1.4115	5.8341	7.2456	
1.6	23.206	113.3	0.001054	1.091	475.5	2220.5	2696.0	1.4553	5.7456	7.2009	
1.8	26.107	116.9	0.001058	.9775	490.8	2210.6	2701.4	1.4945	5.6670	7.1615	

(continued)

Figure 1: Table for saturated steam

Table A-4b Saturated steam: pressure table (SI units) (Continued)

Temperature, °C	Pressure,		Specific volume, m ³ /kg			Specific enthalpy, kJ/kg			Specific entropy, kJ/(kg)(K)		
	bar	psia	v_f	v_g	h_f	h_g	s_f	s_g	v_f	v_g	s_g
120.2	2.0	29.008	0.001061	.8857	504.7	2201.5	2706.2	1.5300	5.5963	7.1263	
127.4	2.5	36.259	0.001067	.7187	535.2	2181.3	2716.5	1.6068	5.4451	7.0519	
133.6	3.0	43.511	0.001073	.6058	561.2	2163.7	2724.9	1.6710	5.3201	6.9911	
143.6	4.0	58.015	0.001084	.4625	604.3	2133.8	2738.1	1.7755	5.1196	6.8951	
151.9	5.0	72.519	0.001093	.3749	639.8	2108.4	2748.2	1.8594	4.9611	6.8205	
158.9	6.0	87.023	0.001101	.3157	670.1	2086.3	2756.4	1.9299	4.8293	6.7592	
170.4	8.0	116.03	0.001115	.2404	720.7	2048.0	2768.7	2.0451	4.6169	6.6620	
179.9	10.	145.04	0.001127	.1944	762.5	2015.1	2777.6	2.1378	4.4479	6.5857	
188.0	12	174.05	0.001139	.1633	798.5	1985.9	2784.4	2.2160	4.3065	6.5225	
195.1	14	203.05	0.001149	.1408	830.2	1959.4	2789.6	2.2838	4.1847	6.4685	
201.4	16	232.06	0.001159	.1238	858.8	1934.8	2793.6	2.3440	4.0770	6.4210	
207.1	18	261.07	0.001168	.1104	884.9	1911.8	2796.7	2.3981	3.9805	6.3786	
212.4	20	290.08	0.001176	.0996	908.9	1890.2	2799.1	2.4474	3.8927	6.3401	
224.0	25	362.59	0.001197	.0800	962.4	1840.2	2802.6	2.5549	3.7018	6.2567	
233.9	30	435.11	0.001216	.0667	1008.7	1795.0	2803.7	2.6461	3.5400	6.1861	
250.4	40	580.15	0.001252	.0498	1087.6	1713.4	2801.0	2.7968	3.2725	6.0693	
264.0	50	725.19	0.001286	.0394	1154.5	1639.4	2793.9	2.9206	3.0520	5.9726	
275.6	60	870.23	0.001319	.0324	1213.7	1570.2	2783.9	3.0271	2.8613	5.8884	
285.9	70	1015.3	0.001352	.0274	1267.4	1504.3	2771.7	3.1216	2.6909	5.8125	
295.1	80	1160.3	0.001385	.0235	1317.0	1440.5	2757.5	3.2073	2.5351	5.7424	
303.3	90	1305.3	0.001417	.0205	1363.7	1379.3	2743.0	3.2870	2.3910	5.6780	
311.1	100	1450.4	0.001453	.0180	1407.9	1316.4	2724.3	3.3600	2.2533	5.6133	
318.2	110	1595.4	0.001489	.0160	1450.2	1255.0	2705.2	3.4296	2.1224	5.5520	
324.8	120	1740.5	0.001527	.0143	1491.2	1193.2	2684.4	3.4960	1.9956	5.4916	
330.9	130	1885.5	0.001567	.0128	1531.1	1130.7	2661.8	3.5599	1.8717	5.4316	
336.8	140	2030.5	0.001610	.0115	1570.4	1066.8	2637.2	3.6220	1.7490	5.3710	
347.4	160	2320.6	0.001710	.0093	1648.9	931.3	2508.2	3.7441	1.5007	5.2448	
357.1	180	2610.7	0.001840	.0075	1731.4	777.4	2508.8	3.8703	1.2336	5.1039	
365.8	200	2900.8	0.002041	.00584	1828.5	581.0	2409.5	4.0172	0.9093	4.9265	
374.1	220.89	3203.7	0.003155	.003155	2098.8	0	2098.8	4.4289	0	4.4289	

Figure 2: table for saturated steam (continued)

Table A-5 Properties of superheated steam (SI units)

Pressure, bar (saturated temperature, °C)	Temperature, °C										
	100	150	200	250	300	400	500	600	700	800	
0.1 (45.81)	v	17.196	19.51	21.825	24.136	26.445	31.063	35.679	40.295	44.911	49.526
	h	2867.5	2783.0	2879.5	2977.3	3076.5	3279.6	3489.1	3705.4	3928.7	4159.0
	s	8.4479	8.6882	8.9038	9.1002	9.2813	9.6077	9.8978	10.1608	10.4028	10.6281
0.5 (81.33)	v	3.418	3.889	4.356	4.820	5.284	6.209	7.134	8.057	8.981	9.904
	h	2682.5	2780.1	2877.7	2976.0	3075.5	3278.9	3488.7	3705.1	3928.5	4158.9
	s	7.6947	7.9401	8.1580	8.3556	8.5373	8.8642	9.1546	9.4178	9.6599	9.8852
1.0 (99.63)	v	1.6958	1.9364	2.172	2.406	2.639	3.103	3.565	4.028	4.490	4.952
	h	2676.2	2776.4	2875.3	2974.3	3074.3	3278.2	3488.1	3704.7	3928.2	4158.6
	s	7.3614	7.6134	7.8343	8.0333	8.2158	8.5435	8.8342	9.0976	9.3398	9.5652
2.0 (120.23)	v	0.9596	1.0803	1.1988	1.3162	1.5493	1.7814	2.013	2.244	2.475	2.706
	h	2768.8	2870.5	2971.0	3071.8	3276.6	3487.1	3704.0	3927.6	4157.8	4381.3
	s	7.2795	7.5066	7.7086	7.8926	8.2218	8.5133	8.7770	9.0194	9.2449	9.4576
3.0 (135.55)	v	0.6339	0.7163	0.7964	0.8753	1.0315	1.1867	1.3414	1.4957	1.6499	1.8041
	h	2761.0	2865.6	2967.6	3069.3	3275.0	3486.0	3703.2	3927.1	4157.8	4381.3
	s	7.0778	7.3115	7.5166	7.7022	8.0330	8.3251	8.5892	8.8319	9.0576	9.2703
4.0 (143.63)	v	0.4708	0.5342	0.5951	0.6548	0.7726	0.8893	1.0552	1.1215	1.2372	1.3529
	h	2752.8	2860.5	2964.2	3066.8	3273.4	3484.9	3702.4	3926.5	4157.3	4380.8
	s	6.9299	7.1706	7.3662	7.5662	7.8985	8.1913	8.4558	8.6987	8.9244	9.1371
5 (151.86)	v	0.4249	0.4744	0.5226	0.6173	0.7109	0.8041	0.8969	0.9869	1.0741	1.1586
	h	2855.4	2960.7	3064.2	3271.9	3483.9	3701.7	3925.9	4156.9	4381.1	4605.3
	s	7.0592	7.2709	7.4599	7.7938	8.0873	8.3522	8.5952	8.8211	9.0311	9.2261

(continued)

Figure 3: table for superheated steam

Table A-5 Properties of superheated steam (SI units) (Continued)

Pressure, bar (saturated temperature, °C)	Temperature, °C									
	100	150	200	250	300	400	500	600	700	800
6 (158.58)	v	0.3520	0.3938	0.4344	0.4744	0.5137	0.5920	0.6697	0.7472	0.8245
	h	2850.1	2957.2	3061.6	3270.3	3482.8	3700.9	3925.3	4156.5	4396.5
	s	6.9665	7.1816	7.3724	7.7079	8.0021	8.2674	8.5107	8.7367	8.9423
10 (179.91)	v	0.2060	0.2327	0.2579	0.3066	0.3541	0.4011	0.4478	0.4923	0.5357
	h	2827.9	2942.6	3051.2	3263.9	3478.5	3697.9	3923.1	4154.7	4396.5
	s	6.6940	6.9247	7.1229	7.4631	7.7622	8.0290	8.2731	8.4996	8.7067
20 (212.42)	v	0.1114	0.1255	0.1512	0.1757	0.1996	0.2232	0.2467	0.2697	0.2923
	h	2902.5	3023.5	3247.6	3467.6	3690.1	3917.4	4150.3	4388.3	4631.3
	s	6.5453	6.7664	7.1271	7.4317	7.7024	7.9487	8.1765	8.3807	8.5622
30 (233.9)	v	0.0758	0.08114	0.0994	0.1162	0.1324	0.1484	0.1641	0.1794	0.1943
	h	2855.8	2993.5	3230.9	3456.5	3682.3	3911.7	4145.9	4384.7	4628.3
	s	6.2872	6.5390	6.9212	7.2338	7.5085	7.7571	7.9862	8.1967	8.3907
40 (250.4)	v	0.0588	0.0734	0.0864	0.09885	0.11095	0.12287	0.1345	0.1461	0.1576
	h	2960.7	3213.6	3445.3	3674.4	3905.9	4141.5	4381.1	4624.7	4872.3
	s	6.3615	6.7690	7.0901	7.3688	7.6198	7.8502	8.0617	8.2557	8.4332
50 (263.99)	v	0.0453	0.0578	0.06857	0.07869	0.08849	0.0981	0.1076	0.1171	0.1266
	h	2924.5	3195.7	3433.8	3666.5	3900.1	4137.1	4377.7	4621.7	4869.1
	s	6.2084	6.6459	6.9759	7.2589	7.5122	7.7440	7.9566	8.1516	8.3301
60 (275.64)	v	0.0362	0.0474	0.0567	0.0653	0.0735	0.0816	0.0892	0.0969	0.1046
	h	2884.2	3177.2	3422.2	3658.4	3894.2	4132.7	4374.7	4619.7	4867.7
	s	6.0674	6.5408	6.8903	7.1677	7.4234	7.6566	7.8681	8.0601	8.2346
70 (285.88)	v	0.0295	0.0393	0.0481	0.0557	0.0628	0.0698	0.0766	0.0833	0.0899
	h	2838.4	3410.3	3650.3	3888.3	4128.2	4372.2	4619.2	4869.2	5123.2
	s	5.9305	6.4478	6.7975	7.0894	7.3476	7.5822	7.7957	8.0007	8.1982

Figure 4: table for superheated steam (continued)

Table A-5 Properties of superheated steam (SI units) (Continued)

Pressure, bar (saturated temperature, °C)		Temperature, °C									
		100	150	200	250	300	400	500	600	700	800
80 (295.06)	v					0.0243	0.0343	0.0418	0.0486	0.0548	0.0610
	h					2785.0	3138.3	3398.3	3642.0	3882.4	4123.8
	s					5.7906	6.3634	6.7240	7.0206	7.2812	7.5173
90 (303.40)	v					0.0299	0.0368	0.0429	0.0486	0.0541	
	h					3117.8	3386.1	3633.7	3876.5	4119.3	
	s					6.2854	6.6576	6.9589	7.2221	7.4596	
100 (311.06)	v					0.0264	0.0328	0.0384	0.0436	0.0486	
	h					3096.5	3373.7	3625.3	3870.5	4114.8	
	s					6.2120	6.5966	6.9029	7.1687	7.4077	
150 (342.24)	v					0.0157	0.0208	0.0249	0.0286	0.0321	
	h					2975.5	3308.6	3582.3	3840.1	4092.4	
	s					5.8811	6.3443	6.6776	6.9572	7.2040	
200 (365.81)	v					0.0099	0.0148	0.0182	0.0211	0.0239	
	h					2818.1	3238.2	3537.6	3809.0	4069.7	
	s					5.5540	6.1401	6.5048	6.7993	7.0544	
250	v					0.0060	0.0111	0.0141	0.0167	0.0189	
	h					2580.2	3162.4	3491.4	3777.5	4047.1	
	s					5.1418	5.9592	6.3602	6.6707	6.9345	
300	v					0.0028	0.0087	0.0115	0.0137	0.0156	
	h					2151.1	3081.1	3443.9	3745.6	4024.2	
	s					4.4728	5.7905	6.2331	6.5606	6.8332	
350	v					0.0021	0.0069	0.0095	0.0115	0.0313	
	h					1987.6	2994.4	3395.5	3713.5	4001.5	
	s					4.2126	5.6282	6.1179	6.4631	6.7450	
400	v					0.0019	0.0056	0.0081	0.0099	0.0115	
	h					1930.9	2903.3	3346.4	3681.2	3978.7	
	s					4.1135	5.4700	6.0114	6.3750	6.6662	
500	v					0.00173	0.0039	0.0061	0.0077	0.0091	
	h					1874.6	2720.1	3247.6	3616.8	3933.6	
	s					4.0031	5.1726	5.8178	6.2189	6.5290	

Figure 5: table of superheated steam continued (2)

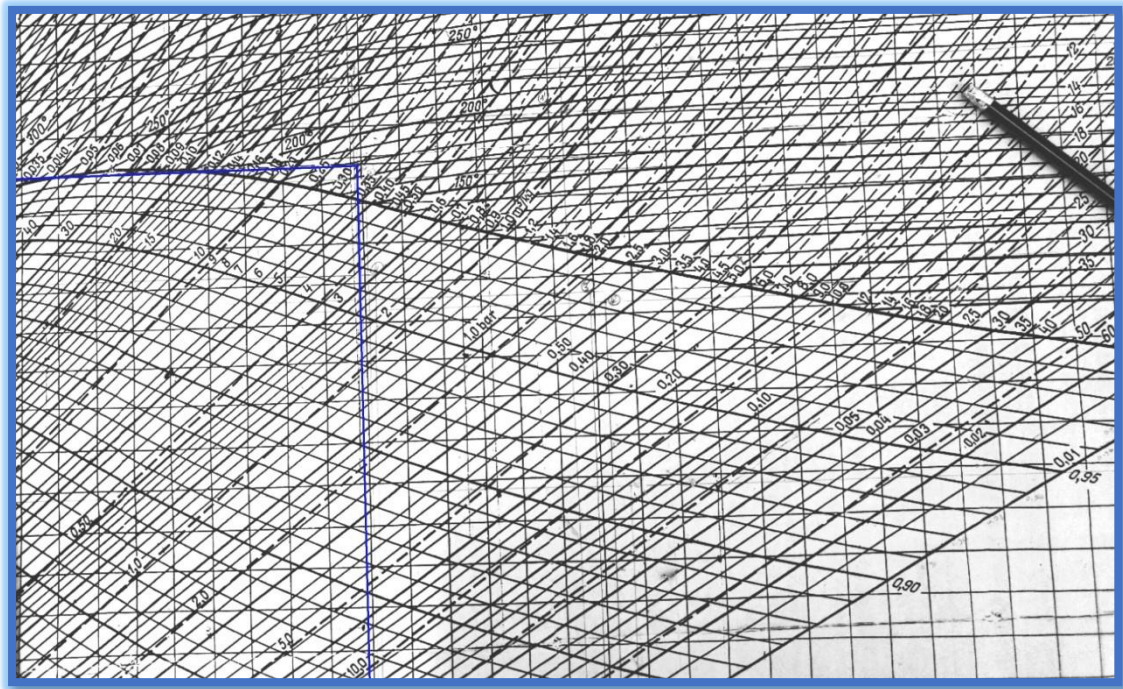


Figure 6: First point-first part

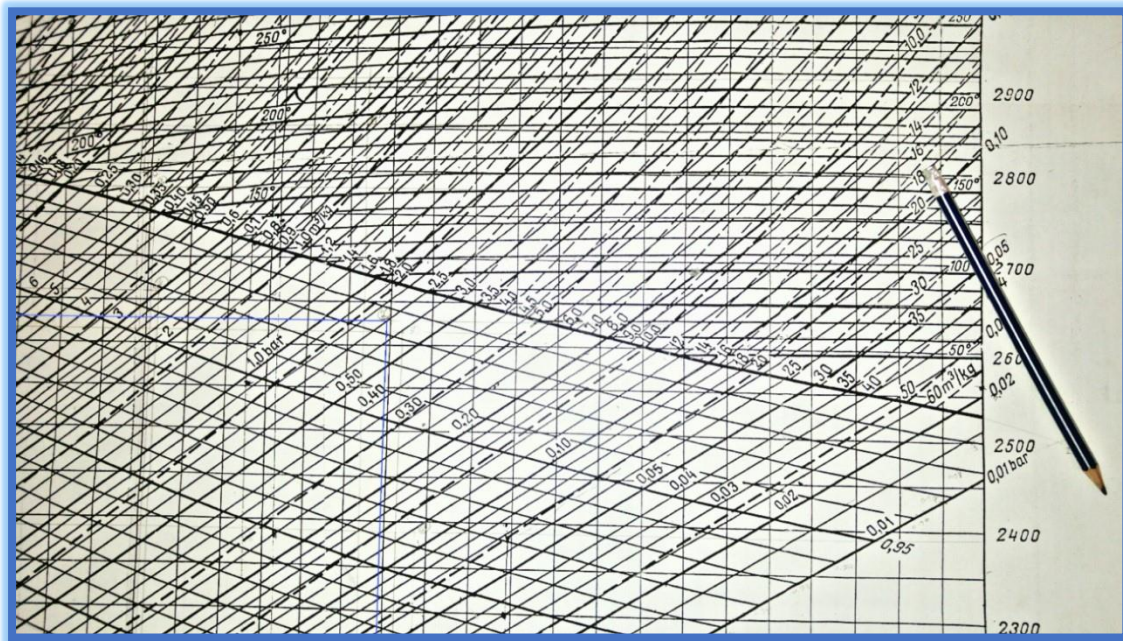


Figure 7: First point-second point

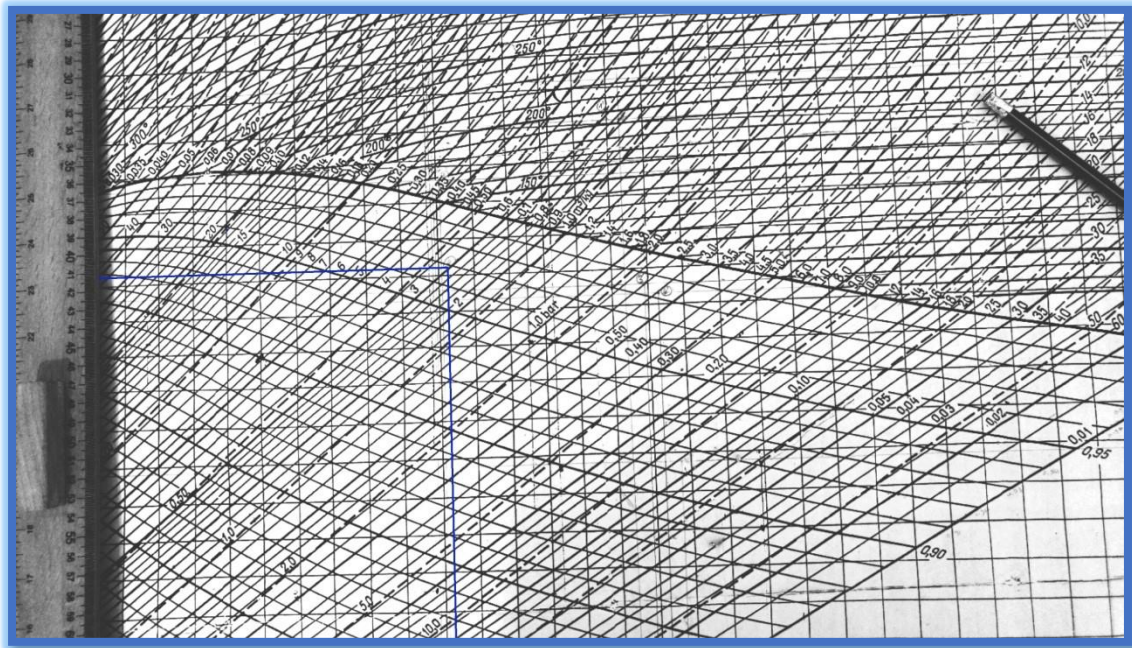


Figure 8: Second point-first part

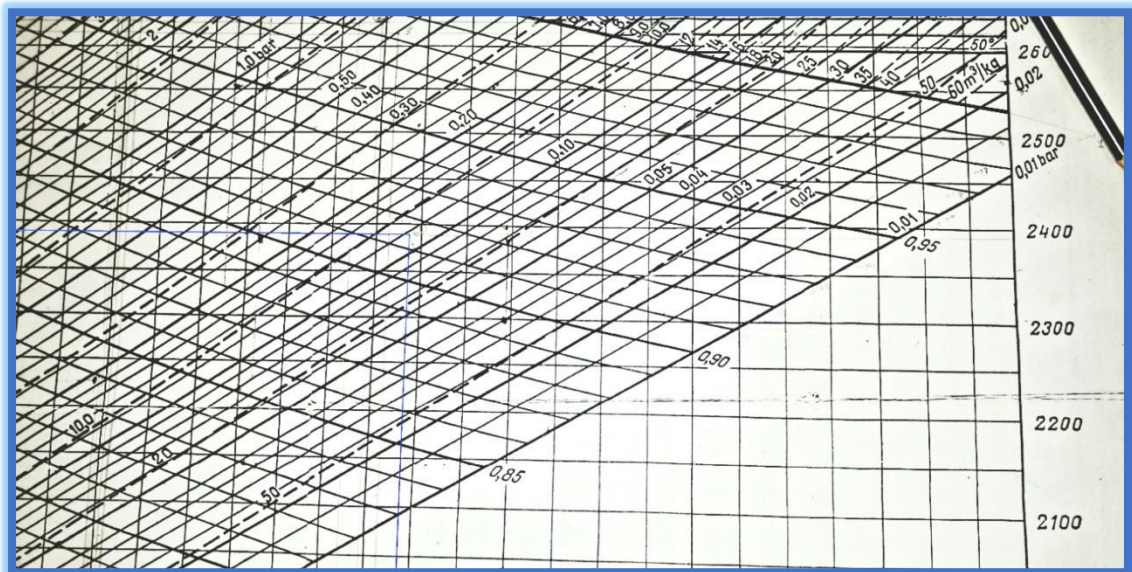
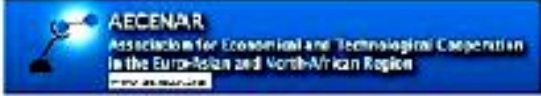




Figure 9: second point-second part

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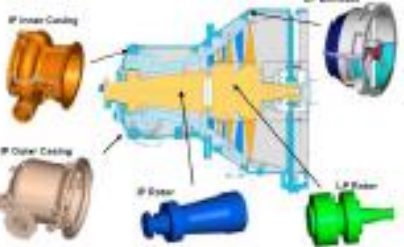
Master Thesis proposal

<p>Laboratory name: AECENAR Laboratory website: http://www.aecenar.com</p>	<p>Supervisors (contact persons) Samir Mourad, smourad69@googlemail.com, Mob. +961 76 341 526 Basim El Kardi, basimkardi@hotmail.com, Mob. +961 76 655 639</p>
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Ras Nhache, 5.3.2016

Thesis title: Design, Regulation and Test of an enlarged turbine (1,5 MW el. power) for a the waste incineration power plant TEMO-IPP

Abstract: The motive power in a steam turbine is obtained by the rate of change in momentum of a high velocity jet of steam impinging on a curved blade which is free to rotate. The steam from the boiler is expanded in a nozzle, resulting in the emission of a high Velocity jet. This jet of steam impinges on the moving vanes or blades, mounted on a shaft. Here it undergoes a change of direction of motion which gives rise to a change in momentum and therefore a force. The turbine modules are furthermore divided into sub-modules of different sizes, which may be combined as required.


Working packages:

- Reading and searching about the subject (2 Weeks) (<http://www.aecenar.com/publications>)
- Detailed design for the cylinder, rotor, and blades (3 Weeks)
- Waste incineration based regulation concept for the turbine (e.g. air supply regulation) (3 weeks)
- Support of the CNC machine based manufacturing of the turbine (4 weeks) (- 15 June 16)
- Test of the delivered turbine with the the TEMO-IPP incinerator (2 weeks) (- 30 June 16)
- Calculation of the efficiency of the turbine (1 Week)
- Documentation (4 Weeks).

(the duration of packages is only approximately)

Key Words: alternative energy, FreeCAD, steam turbine regulation, cylinders, rotor, blades, CAM (Computer Aided Manufacturing)

Figure 10: master thesis proposal



CNC LAB - RAMI NASSOUH & CO.
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 Bahsas, Tripoli, Lebanon
 VAT#: 2574250-601
 Phone # +961 6 412895
 E-mail sales@cnclablb.com
 Web Site www.cnclablb.com

Sales Receipt

Date	Sale No.
23-Sept-2016	SR16-203

Sold To					
Samir Mourad					
		Customer Price Level			
Item	Description	Qty	U/M	Rate	Amount
3D Rapid pototyping	3D Rapid prototyping(Plastic 3D Printing)	4.16	hr	8.00	33.28
	VAT Sales Tax			10.00%	0.00
Total					USD 33.28

Figure 11: sales receipt

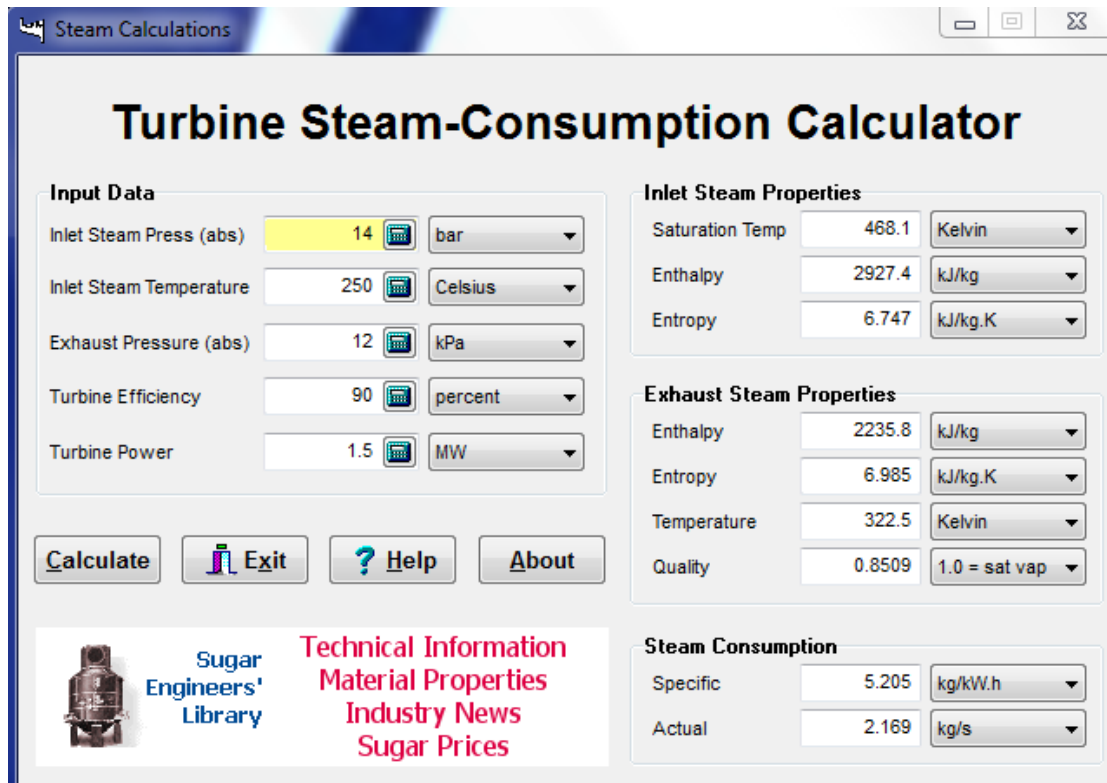


Figure 12: verification for value of steam flow for first turbine (in chapter 2 and 3)

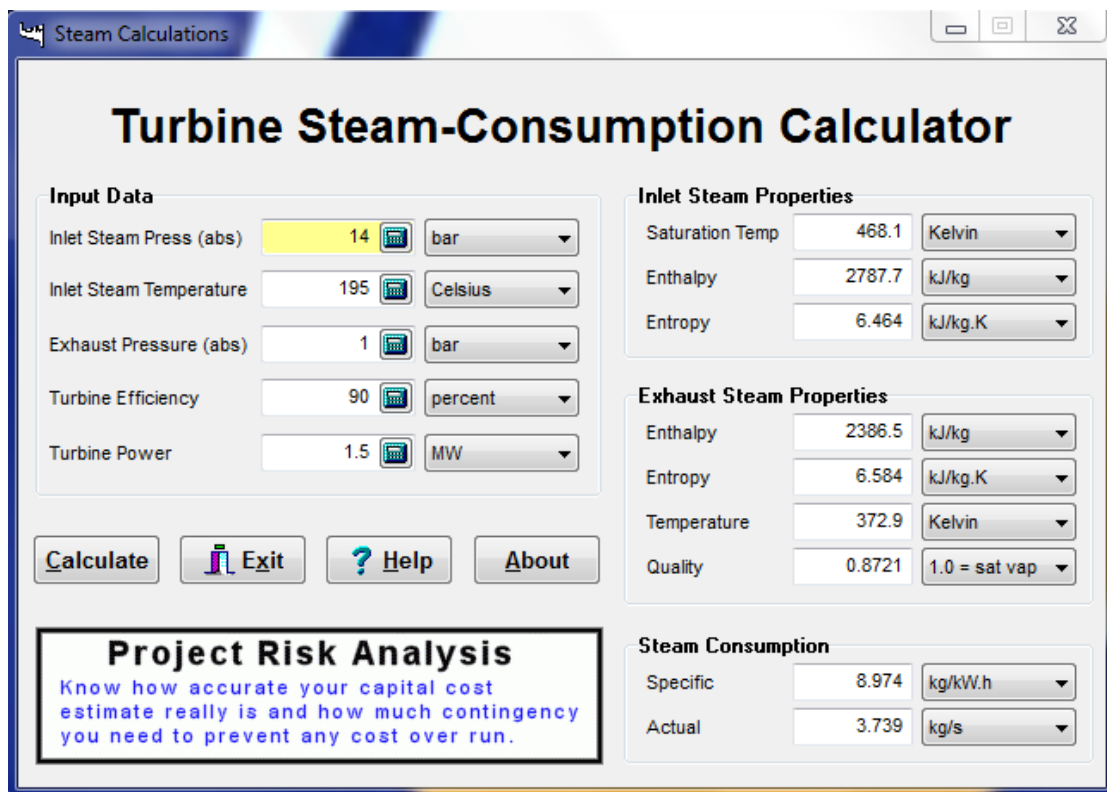


Figure 13: verification for value of steam flow for second turbine (in chapter 4)



Figure 13: Steam turbine of 40 kW in North Lebanon Power Plant Company



Figure 14: method of putting fixed blade



Figure 15: fixed blades with its cover



Figure 16: steam turbine model- inner parts