ANALYSIS OF DIFFERENT TYPES OF REGULATION AND ITS EFFICIENCY IN STEAM POWER CYCLES

MASTER THESIS

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ABSTRACT

The main subject of this work is the analysis of the both main ways of regulation of the rankine cycle used in thermal power plants based in steam cycle. The work is divided in three main parts. The first one is theoretical, and in it, is introduced the rankine cycle, the ideal and real, and different improves of it. The second is theoretical too, and it is about the different types of regulation and how they work. And finally the third part is practical one, in it the start point is a rankine cycle used in a plant working at nominal load. Then the both, nozzle and throttle governing are calculated with two different mass flow than nominal flow. In this part is also calculated the efficiency of the control stage of the turbine used in the nozzle regulation.

SYMBOL TABLE

Symbol	Description	Unit
ŋ	Performance	%
W	Power of work	kW
Q	Power of heat	kW
T	Temperature	ºC or K
h	Enthalpy	kJ/kg
s	Entropy	KJ/kg.K
p	Pressure	bar
m	Mass flow	kg/s
ρ	Density	kg/m^3
C	Stodola constant	
c	Absolute speed in velocity triangles	m/s
w	Relative speed in velocity triangles	m/s
u	Peripheral speed of the blades of the rotor	m/s
α	Angle between c and u in the velocity triangles	º

β	Angle between w and u in the velocity triangles	º
D	Diameter of the control stage	m
n	Revolutions of the turbine	rpm
ω	Angular speed of the turbine	rad/s
L	Length of the blades	m
ξ_s	Percentage of losses in the stator blades	%
ξ_R	Percentage of losses in the rotor blades	%
Δ_s	Losses in the stator blades	kJ/kg
Δ_R	Losses in the rotor blades	kJ/kg

In annex I the symbol $\alpha_{13\theta}$ is referred to the angle between c and u at the entrance of the rotor, so $\alpha_{13\theta} = \alpha_1$.

In annex J the symbol $\beta_{23\theta}$ is referred to the angle between w and u at the entrance of the rotor, so $\beta_{23\theta} = \beta_2$.

RANKINE CYCLE

IDEAL RANKINE CYCLE

The Rankine cycle is a thermodynamic cycle which aim is to obtain work from heat, it is a power cycle, and like other power cycle its efficiency is limited by Carnot cycle.

The Rankine cycle is the base of the working steam power plants and of the nuclear power plants. Its work fluid usually is water. A pump impulses liquid water to the boiler, there burning fuel, the liquid water becomes in steam. In a turbine the steam expands to obtain mechanical work that with a generator it becomes in electricity. Finally the fluid becomes again liquid water in the condenser. The stages of the cycle are:

- Process 1-2: Isentropic expansion of the steam in a turbine in order to get mechanical work.
- Process 2-3: Heat transmission in a condenser from the work fluid to the refrigerant to get saturated liquid. This process is at constant pressure.
- Process 3-4: Isentropic compression of the fluid in a pump, this process is made with liquid fluid in order to contribute less work.
- Process 4-1: Heat transmission to the liquid at constant pressure in order to get overheated steam.

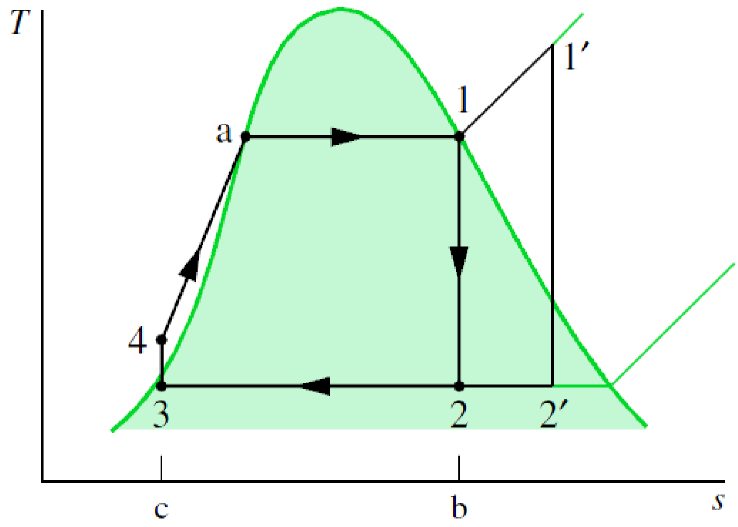


Fig 1. Ideal Rankine cycle. "Fundamentals of engineering thermodynamics", Moran, J., 5th edition, 2006.

In this cycle there is one contribution of work in the pump, one obtaining of work in the turbine and two heat transfers in the boiler and in the condenser. The efficiency of the rankine cycle is $\eta = \frac{W_{12} - W_{34}}{Q_{41}}$.

REAL RANKINE CYCLE

The ideal cycle ignores heat transfer losses in the condenser, pressure losses in the condenser and in the boiler and the fact that the compression and the expansion are not isentropic processes. So the diagram of the real Rankine cycle is:

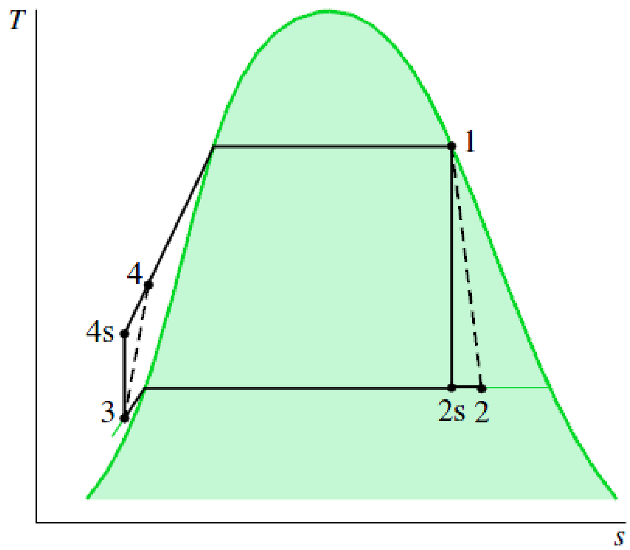


Fig 2. Effect of the irreversibilities in the pump and in the turbine. "Fundamentals of engineering thermodynamics", Moran, J. ,5th edition, 2006.

These irreversibilities make a loss in the efficiency of the cycle so the cycles used in power plants have some improvements to up the efficiency.

Pressure changes

If the pressure in the condenser gets down, the turbine offers more work but the problem is that if the pressure is too low, the title of the steam ups and it becomes a problem for the turbine blades.

If the pressure of the boiler gets up without change the temperature the efficiency of the turbine is higher and therefore the efficiency of the cycle is higher too. The problem is with higher pressures there is, like in the other case, more humidity at the exit of the turbine

SUPERHEATING

The superheating process is to heat more the steam before its entrance on the turbine to get superheated steam. This type of cycle is also called Hirn cycle. With this method the work obtained is higher and the title of the steam when it goes out of the turbine is higher too even it could be steam superheated, that is good to not to damage the blades. The common practice is to keep the steam title upper than 88%. The combination of boiler and the superheater is also called steam generator. The temperature of the superheated steam is limited by the blades of the turbine in the entrance.

REHEATING

With this method the steam do not expands in one only stage until the condenser pressure. The steam expands in two or more stages at intermediate pressures. Between each turbine the steam is reheated again. With this way the production of work is higher and the title of the steam in the exit of the turbine is higher too.

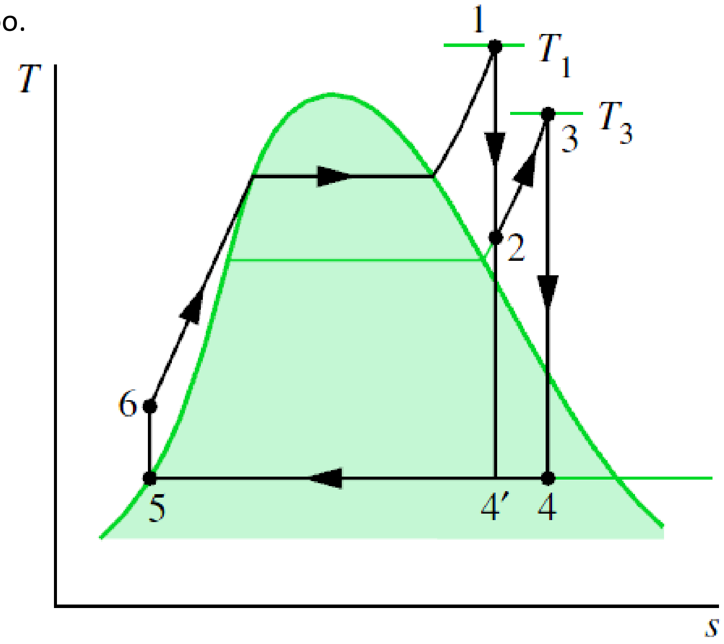


Fig 3. Rankine cycle with superheating and reheating ."Fundamentals of engineering thermodynamics" , Moran, J. ,5th edition, 2006.

REGENERATION

Another manner to improve the thermal performance of the cycle is to preheat the liquid water before its entrance in the boiler. This can be done doing steam extractions from the turbines. For example, the steam expands in the high pressure turbine and then a fraction "y" of steam goes out and the (1-y) fraction of steam expands in another turbine. The y fraction is used to heat the water before getting in the boiler. With this method the fuel consumption is lower but there is a lower production of work too. So the fractions to be extracted have to be calculated in a accurate way to get the higher thermal efficiency.

The y fraction of steam heat the water in heat exchangers. There are two types of exchangers. In closed changers there is no mix between steam and water so the work pressures can be different:

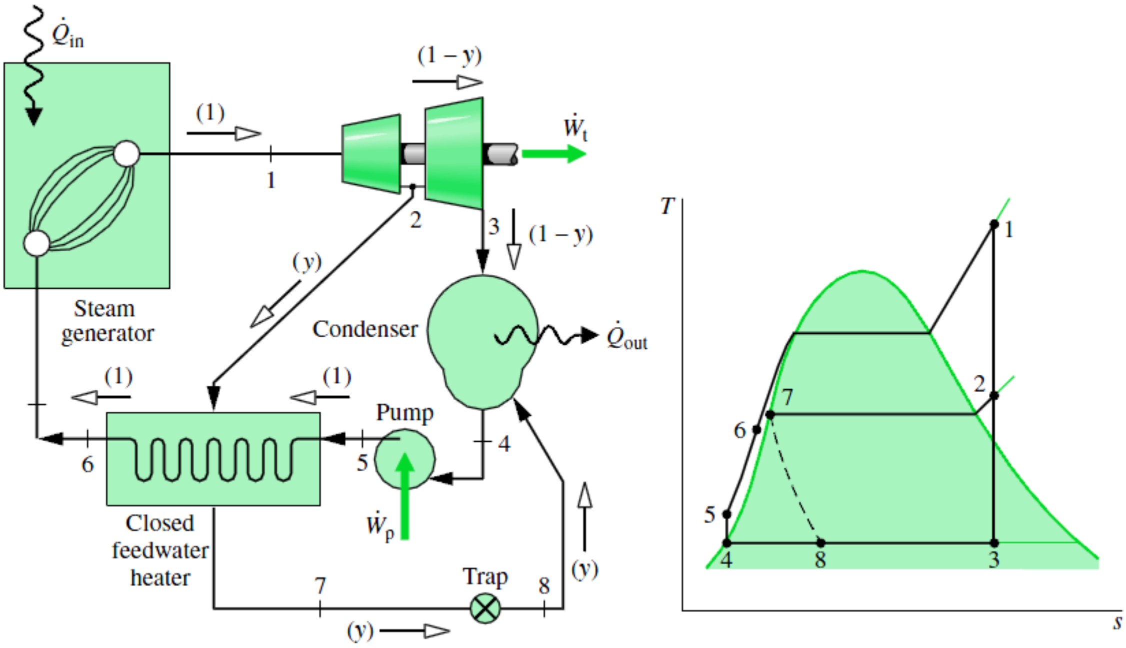


Fig 4. Rankine cycle with one steam extraction and a close exchanger. "Fundamentals of engineering thermodynamics", Moran, J. ,5th edition, 2006.

In opened exchangers the steam and the liquid get mixed:

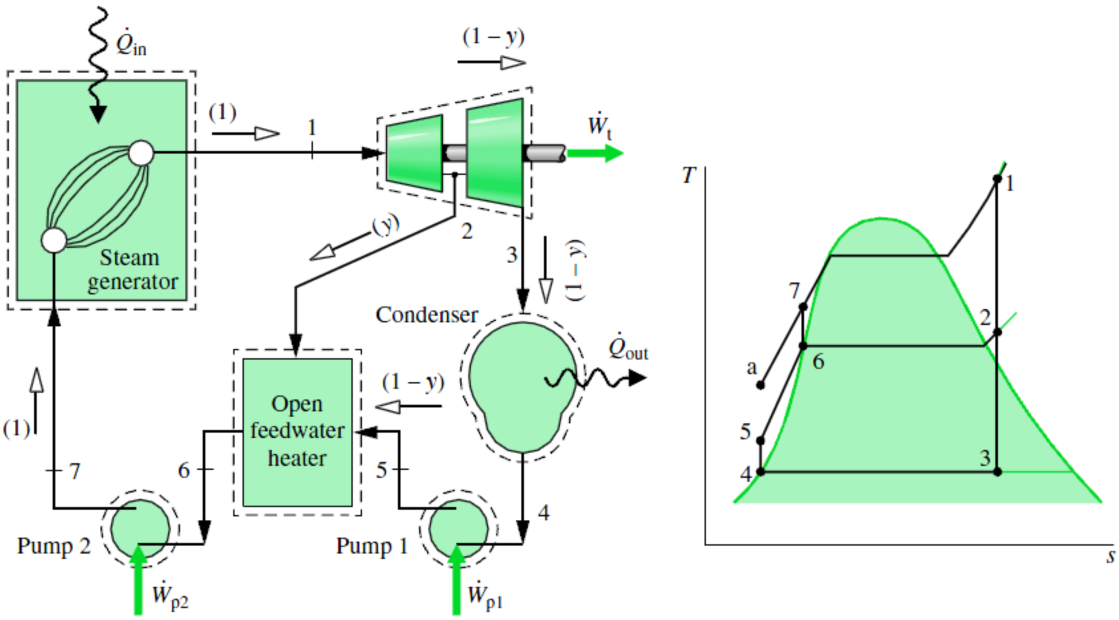


Fig 5. Rankine cycle with one steam extraction and a open exchanger. "Fundamentals of engineering thermodynamics", Moran, J., 5th edition, 2006.

BINARY CYCLE

In a binary cycle there are two working fluids. One works in high temperature and the other at low temperature. One example is the Hg and water. The Hg works at high temperature and the heat that gives in the condenser is profited by water to vaporize. But the water needs extra heating to get to the optimal temperature in the entrance of the turbine. This type of cycle has a good thermal efficiency, generates big power and saves fuel but the construction is much more expensive. But the main problem of this type of cycle is that they are dangerous because heavy metals are very poisonous.

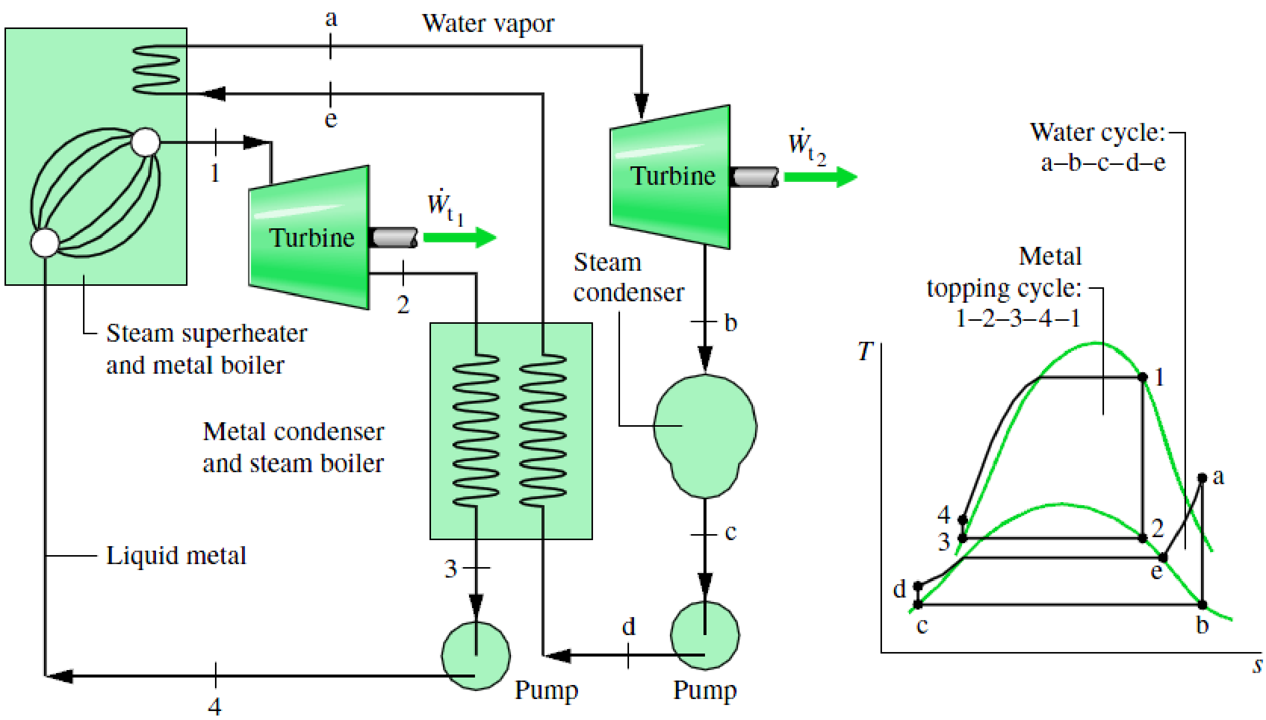


Fig 6 . Water – Metal binary steam cycle. "Fundamentals of engineering thermodynamics", Moran, J., 5th edition, 2006.

REGULATION OF STEAM TURBINES

The regulation of the steam turbines is used to adapt the power of the turbine in a constant velocity to the requirements of the charge, the alternator.

There are two main types of regulation:

- The qualitative regulation or throttle governing, which consists in the control of the pressure of the steam in the entrance of the turbine
- The quantitative regulation or nozzles governing, where the control is over the number of nozzles or stages of the turbine that receive the steam.

THROTTLE REGULATION

This governing is made by getting down the pressure of the steam, this is made by closing a valve in the entrance of the turbine, the process is called lamination. In this process there is no interchange of heat or work with the exterior, so the process is at constant enthalpy. The entropy gets up and the pressure lows. The aim is to reduce the mass flow rate to get reduced the available energy and hence the output lows.

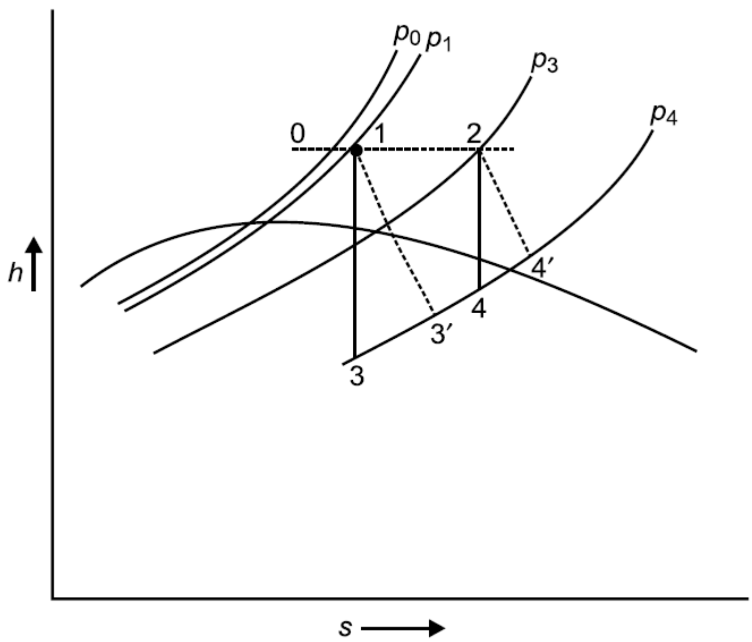


Fig 7. Qualitative regulation in h-s diagram. Expansion lines. "Applied thermodynamics", Singh, Onkar , 3th edition, 2009.

As the valve is closed, the constant enthalpy process occurs through the valve with an increase in entropy and a decrease in availability of energy per unit of mass flow rate. It may be noted that even when the valve is fully open, there is a drop of pressures. So with this type of regulation there is a loss of pressure at all loads of the turbine.

NOZZLE REGULATION

In an ideal situation with this regulation the steam instead of enter in all the nozzles of the crown of the turbine the steam would enter only in some nozzles in all the stages. But this is not reasonable so usually is done only in some stages. The regulation can be done controlling the power in the first stage doing a segmentation of the disc of nozzles like in the next image:

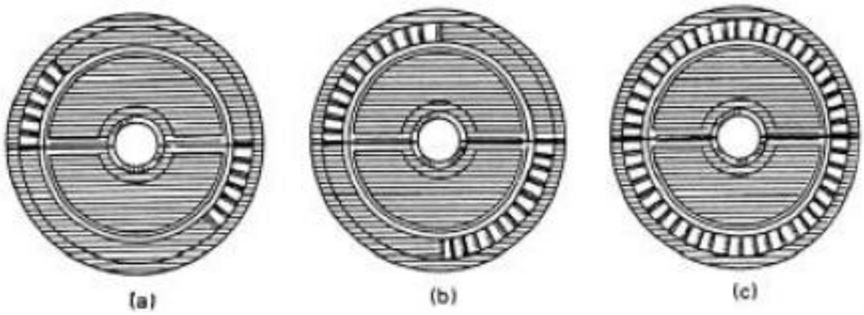


Fig 8. Diagrams of steam turbines with different degree of admission: (a) and (b): partial admission; (c): total admission. “Control and governing of steam turbines”, University of Buenos Aires, Own edition, 1998.

The segmentation can be done dividing the distributing conduct in circular sectors and feeding each sector with closing and opening valves, like in the next image:

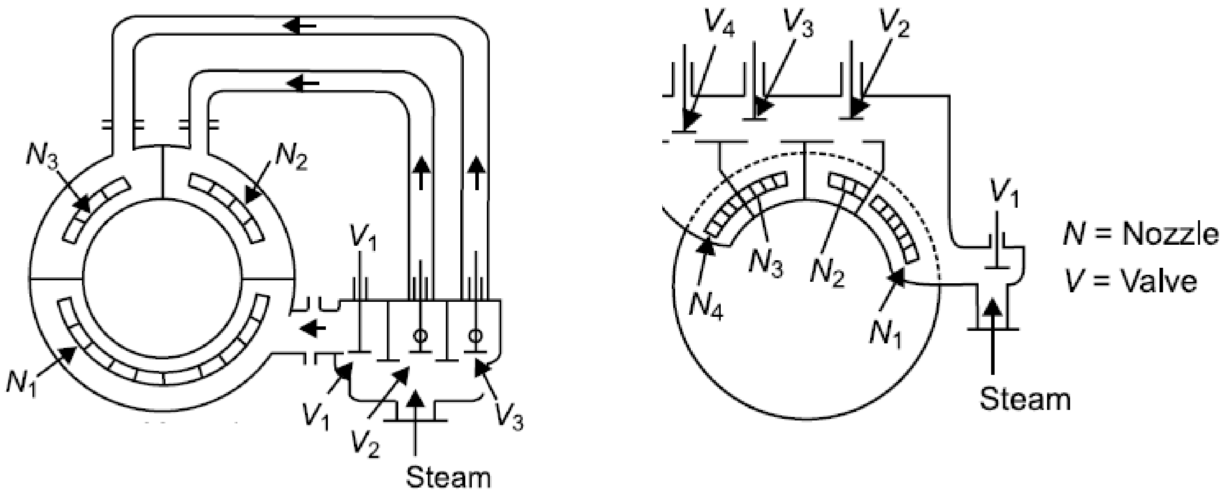


Fig 9. Schematic of nozzle governing. “Applied thermodynamics”, Singh, Onkar , 3th edition, 2009.

The process of closing part of the crown of nozzles in the first stage is represented in the next image:

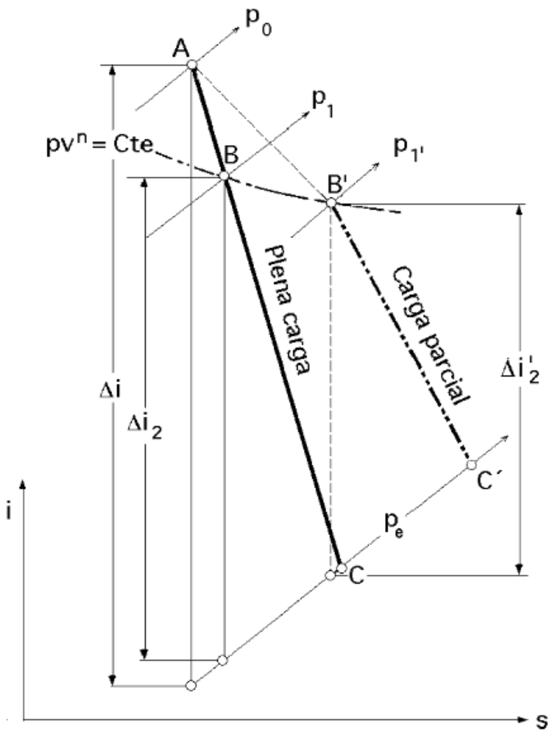


Fig 10. Quantitative regulation in h-s diagram. Expansion lines. “Regulation and control devices of steam turbines”, University of Cantabria, Own edition, 2001.

To close some part of the nozzles of the first stage, the steam expands until a lower pressure B´, with a weak charge and with worse efficiency. From here the rest of the turbine works like as a lamination regulation, with the exception that the admission point doesn´t move over a isenthalpic line, it moves in a polytrophic line with the form $pv^n = constant$.

OVERCHARGE

In some situations it is required to get a 10% or 20% extra over the nominal power of the turbine in situations of peak demand of energy. For that reason some turbines are able to work in overcharge, but the efficiency gets down compared with nominal work.

DIFFERENCES BETWEEN THROTTLE GOVERNING AND NOZZLE GOVERNING

THROTTLE GOVERNING	NOZZLE GOVERNING
Because of throttling of the steam at the inlet of the turbine there are important throttling losses.	The degree of losses in the nozzle governing valves is negligible.
It has smaller losses due to partial admission of steam.	It has large partial admission losses.
This type is used in both impulse turbines and reaction turbines.	This regulation is used in impulse turbines and only in reaction turbines that have initial impulse stages.

SYSTEM OF CONTROL AND REGULATION

The system of regulation and control of the steams turbines is very linked with the work of the whole plant where the turbines are. So there isn´t a only system, but the most common and basic elements of control are:

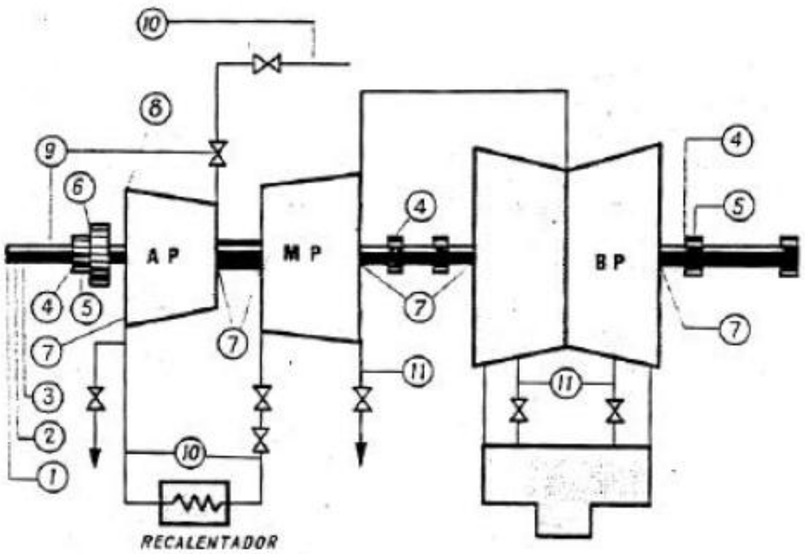


Fig 11. Instrumentation of a steam turbine. "Control and governing of steam turbines", University of Buenos Aires, Own edition, 1998.

1. Revolution counter **2.** Measurer of the eccentricity of the rotor **3.** Measurer of rotor position **4.** Vibrations measurer **5.** Measurer of temperature and pressure of the bearing oil **6.** Measurer of the deferential expansion **7.** Measurer of temperature and pressure of the labyrinthine closing **8.**Measurer of the pressure in the entrances of the nozzles **9.** Measurer of the position of the admission valve **10.** Measurer of flux, pressure and temperature **11.** Measurer of the temperature and pressure of the steam conducts.

CALCULATION PART

NOMINAL CYCLE DESCRIPTION

The base of this work is rankine cycle used in a thermal power plant. It is a regenerative rankine cycle with extractions in the three parts of the steam turbine, the high pressure, the intermediate pressure and the low pressure part. It also has reheater and superheater. In annex C there is graphic of the cycle with the different states numbered, and in annex D there is the same graphic but with the states of nominal cycle.

The cycle starts at the exit of the superheater of the boiler (Point 1), with a temperature of 540ºC and a pressure of 168 bar. Then the steam flows to the high pressure turbine but first there, is the regulation valve, it is an isenthalpic valve; in this case the valve is wide open because it is the nominal cycle, but even so, there is a loss of pressure of 2% (point 2). The high pressure turbine expands steam until 46 bar (point 3), but it has two stages of extractions, the first extraction (point 20) it is at 70 bar and the second extraction (point 21) it is at the same pressure than the exit, 46 bar.

Then the flow of steam enters again in the boiler, in the reheater part. In this process there is a loss of pressure of 4.6 bar. So at the exit of the reheater (point 4) the pressure it is of 41.4 bar and the temperature is again 540ºC. Then before entering in the intermediate pressure turbine there is another regulation valve in where the pressure of the steam flow drops again 2% (point 5). In the intermediate pressure turbine the main flow expands until 4.184 bar, and it has three stages of extractions; the first at 22 bar, the second at 11 bar and the third at 4.184 again.

From the exit of the second part of the turbine (point 6), the flow of steam gets inside the low pressure turbine, where is expanded until 0.04 bar (point 7). This last part has 4 stages, and the extractions are at 1.529 bar, 0.39 bar and at 0.175 bar. Then the main flow enters in the condenser. Inside the condenser also gets in the liquid which comes from the extractions of the low pressure turbine and the last extraction of intermediate pressure turbine, leaving before from their respective exchangers.

Condensate liquid at 0.04 bar exits the condenser (point 8), and enters in the first pump where its pressure rises until 25 bar (point 9). Then the liquid flow is heated in 4 closed exchangers, the heat in each exchanger is provided for the extractions from the turbine. So the first, second and third exchangers receive the third, second and first extraction respectively from the low pressure turbine, and the fourth exchanger receives the last extraction of the intermediate pressure turbine. In the closed exchangers at the exit of the side of extraction there is always saturated liquid at the pressure of the extraction.

Then the flow goes through a valve to low his pressure until 10.67 bar, which is the pressure of the open feedwater and the pressure of the second extraction of the intermediate pressure. This open exchanger is also fed by the saturated liquid that comes from the extractions of high pressure turbine and the first extraction of intermediate pressure turbine, leaving before from their respective exchangers. In the exit of the feedwater (point 15) there is saturated liquid at the pressure of the feedwater 10.67 bar.

The flow goes through a pump which rises it pressure until 210 bar, and then like in the previous process, the flow increases its enthalpy and temperature going along three closed exchangers. These exchangers are also fed by the rest extractions from the turbine, and in their exits there is saturated liquid. At the exit of the last closed exchanger (point 19) the steam flow gets inside the boiler and the superheater and it exits at 168 bar and 540ºC (point 1). So there is a pressure drop of 42 bar. And in this point the cycle starts again.

The efficiencies of the different parts of the turbine and the pumps are:

High pressure turbine:

η_{HP} = 91.42% η_{HP1S} = 92.34% η_{HP2S} = 88.16%

Intermediate pressure turbine:

η_{IP} = 93.74% η_{IP1S} = 93.16% η_{IP2S} = 92.27%

Low pressure turbine:

η_{LP} = 88.58% η_{LP1S} = 90% η_{LP2S} = 92.16% η_{LP3S} = 95.98%

Pumps:

η_{FP} = 71.82% η_{SP} = 75.54%

The total efficiency of the cycle is 47.76% the work produced by the turbines is of 183000 kW approximately and the pumps consume 4800 kW.

The mass flow of the nominal cycle is m=150 kg/s ,the next step is to calculate this cycle in off-design situations, when m=130 kg/s and m=110 kg/s using the two main types of regulation of a steam power cycle, throttling governing and nozzle governing.

BASIC ASSUMPTIONS

In order to make all the calculations of the different states of the cycles some assumptions are established:

1) The pressure at the exit of the low pressure turbine is the same than in the nominal cycle, 0.04bar.
2) Liquid saturated leaves always the condenser at 0.04 bar.
3) The pressures in the output of the pumps are the same than in the nominal cycle.
4) At the exit of the closed exchangers in the extraction side there is always saturated liquid.
5) The loss of pressure in the boiler plus superheater is the same than in the nominal cycle.
6) The loss of pressure in the reheater is the same than in the nominal cycle.

7) The efficiencies of the turbines and of its different stages, and of the pumps are the same than in the nominal cycle.
8) The second regulation valve between states 4 and 5 is always fully opened, and the pressure drops 2%.

STODOLA CONSTANTS

The first step is to calculate using the Stodola equation, the Stodola constants for each stage of each turbine, starting for the exit of the low pressure turbine and going back.

The equation is: $$C := \frac{m}{\sqrt{p_1^2 - p_2^2}} \qquad (1)$$

where p_1 is the pressure of enter at the stage and p_2 is the pressure at the exit. Once all the constants are calculated the next step is for each different value of mass flow, to calculate the different new pressures. The pressure at the end of the low pressure turbine is the same in all cases, this pressure is 0.04 bar, then the calculation starts from the last stage of this turbine and continues until arrive at the first stage of the high pressure turbine, it is easy to calculate these pressures using:

$$p_{1'} := \sqrt{\frac{m^2}{C^2} + p_2^2} \qquad (2)$$

All the calculations of the Stodola constants, and of the new pressures for both new steam flows are in the annex A.

THROTTLE GOVERNING CALCULATION

The point 2 in the cycle is completely known because the pressure of enter in the turbine is known thanks to the Stodola equation, and the enthalpy is known too because the regulation valve is isenthalpic. As state 2 is known , states 3 ideal and 20 ideal are also known, so:

$$h_{3'} := h_{2'} - n_{HP} \cdot \left(h_{2'} - h_{3id'}\right)$$
$$h_{21'} := h_3 \qquad (3)$$
$$h_{20'} := h_{2'} - n_{HP1S} \left(h_{2'} - h_{20id'}\right)$$

Then state 4 is also known because is known the temperature in the output of the reheater and the drop of pressure between 3 and 4. Therefore state 5 is also known because the regulation valve in the intermediate turbine is wide open. So following the same procedure than before there is:

$$h_{6'} := h_{5'} - n_{IP} \cdot \left(h_{5'} - h_{6id'}\right)$$
$$h_{22'} := h_{5'} - n_{IP1S} \left(h_{5'} - h_{22id'}\right) \qquad (4)$$
$$h_{23'} := h_{5'} - n_{IP2S} \left(h_{5'} - h_{23id'}\right)$$
$$h_{24'} := h_6$$

And knowing state 6 the same is for the low pressure turbine:

$$h_{7'} := h_{6'} - n_{LP} \cdot \left(h_{6'} - h_{7id'}\right)$$
$$h_{25'} := h_{6'} - n_{LP1S} \left(h_{6'} - h_{25id'}\right)$$
$$h_{26'} := h_{6'} - n_{LP2S} \left(h_{6'} - h_{26id'}\right) \qquad (5)$$
$$h_{27'} := h_{6'} - n_{LP3S} \left(h_{6'} - h_{27id'}\right)$$

The results for both different mass flows in this type of regulation are in the graphic annex E and F.

NOZZLE GOVERNING CALCULATION

In this case the point of entrance in the turbine, the state 2, is also known because in this type of regulation de regulation valve is fully open and the enthalpy and pressure of state 1 is known, there is only the 2% pressure loss. Thanks to

Stodola equations, the pressures of the extraction points are also known. As state 2 is known, states 3 ideal and 20 ideal are also known, so:

$$
\begin{aligned}
&h_{3'} := h_{2'} - n_{HP} \cdot \left(h_{2'} - h_{3id'}\right) \\
&h_{21'} := h_3 \\
&h_{20'} := h_{2'} - n_{HP1S} \cdot \left(h_{2'} - h_{20id'}\right)
\end{aligned}
\tag{6}
$$

Then state 4 is also known because is known the temperature in the output of the reheater and the drop of pressure between 3 and 4. Therefore state 5 is also known because the regulation valve in the intermediate turbine is wide open. So following the same procedure than before there is:

$$
\begin{aligned}
&h_{6'} := h_{5'} - n_{IP} \cdot \left(h_{5'} - h_{6id'}\right) \\
&h_{22'} := h_{5'} - n_{IP1S} \cdot \left(h_{5'} - h_{22id'}\right) \\
&h_{23'} := h_{5'} - n_{IP2S} \cdot \left(h_{5'} - h_{23id'}\right) \\
&h_{24'} := h_6
\end{aligned}
\tag{7}
$$

And knowing state 6 the same is for the low pressure turbine:

$$
\begin{aligned}
&h_{7'} := h_{6'} - n_{LP} \cdot \left(h_{6'} - h_{7id'}\right) \\
&h_{25'} := h_{6'} - n_{LP1S} \cdot \left(h_{6'} - h_{25id'}\right) \\
&h_{26'} := h_{6'} - n_{LP2S} \cdot \left(h_{6'} - h_{26id'}\right) \\
&h_{27'} := h_{6'} - n_{LP3S} \cdot \left(h_{6'} - h_{27id'}\right)
\end{aligned}
\tag{8}
$$

The results for both different mass flows in this type of regulation are in the graphic annex G and H.

ITERATIVE PROCESS

Once all calculations of the states of the extractions and of the states in the exit of the turbines, are done, the next step is to calculate the mass flow of each extraction and the states between the exchangers. This process is common for the two types of regulation. It is important to know that the states in the output of the closed exchangers in the extractions sides are already known, because the pressures of work of the exchangers are known and in the output there is saturated liquid. These states are: 28, 29, 30, 15, 31, 32, 33 and 34.

This procedure is an iterative process, starting from point 8 which is known, balances of mass and energy are made in the pumps, closed exchangers and in the open exchanger in order to get the values of the different states and the mass of the different streams of the cycle. The process starts first calculating the mass flows of the extractions and then the enthalpies of the states between exchangers. In the first iteration the process start from some approximated values of the mass extractions.

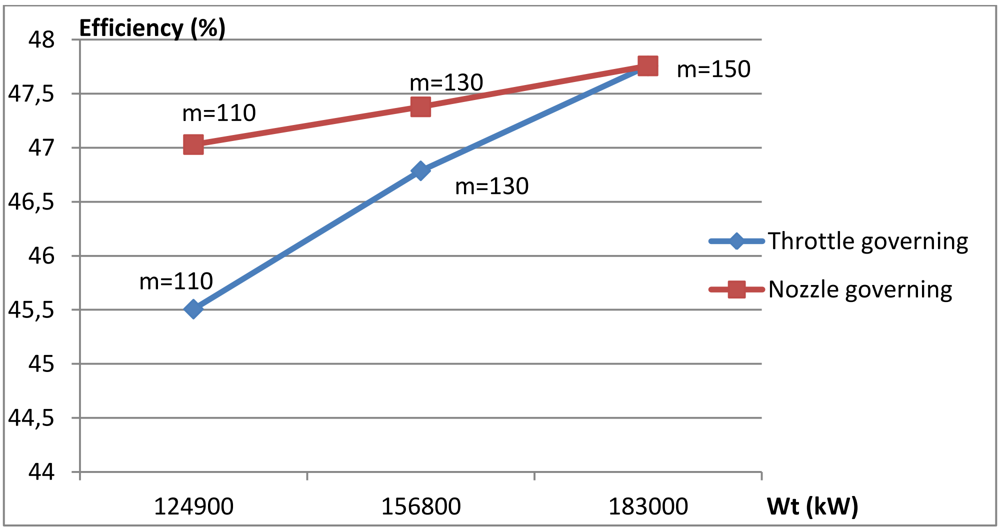
This iterative process is detailed in the annex B, and the different final results are expressed in the different graphic annexes.

GRAPHIC RESULTS AND CONCLUSIONS

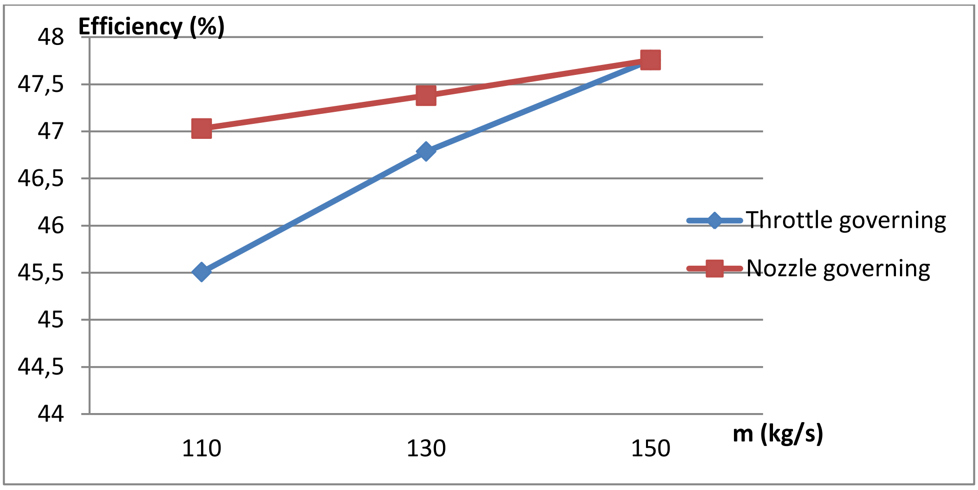
With the knowledge of all states for each type of regulation and each quantity of mass flow it is possible to make the following significant graphics:

Throttle	Efficiency (%)	Turbine Power (kW)	**Nozzle**	Efficiency (%)	Turbine Power (kW)
110	45,507	124900	**110**	47.03	131900
130	46,787	156800	**130**	47,379	160300
150	47,758	183000	**150**	47,758	183000

Graphic efficiency vs Turbine Power:



Graphic efficiency vs mass flow:



As the graphics show, when the mass flow of the cycle decreases, the efficiency lows too, and comparing both types of regulations, it is easy to see that nozzle governing is better than throttle governing because for each case of mass flow, efficiency is higher in the nozzle governing one. This is because of the throttle losses caused at regulation valve in the throttle regulation, while losses in the inlet of the turbine in nozzle governing are negligible. It is true that in nozzle regulation, the partial admission losses are larger than in throttle regulation, but this type of losses is not so important as the throttle ones.

So it is possible to say that nozzle governing is better than throttle governing when the cycle works in off-design loads.

CONTROL STAGE CALCULATION

The aim of this part is to calculate the efficiency of the control stage of the high pressure turbine in the nozzle regulation of the nominal power cycle which this work is based. This control stage is a pure action stage, in this type all the enthalpy decrease happens in the stator; there is not enthalpic jump in the rotor.

The conditions at the entrance of the turbine are defined already, the entrance pressure is 164.64 bar and the enthalpy is 3403.1 kJ/kg. In this control stage the steam expands until 90 bar.

The conditions of the speed inside the stage of a turbine are defined for the velocity triangle:

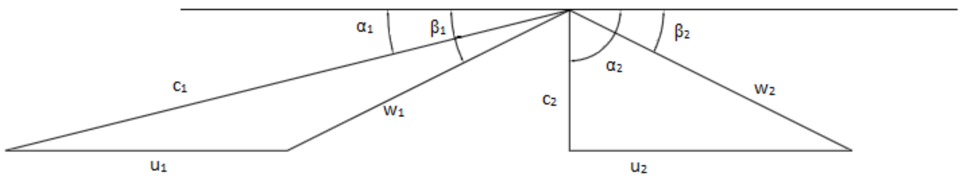


Fig 12. Example of a generic triangle of velocities. Creation of the author

The number 1 is referred at the entrance, and 2 at the exit. In this type of diagram c is the absolute velocity of the steam flow, w is the relative velocity (is the velocity if the spectator was in the blades of the rotor), and u is the peripheral speed of the blades of the rotor. Respect the angles, α is the angle between c and u, and β is the one between w and u.

Now the first step is to build the possible velocity triangles of the stage, both entrance and exit. But first is necessary to establish some previous considerations. These are:

1) The speed at the entrance of the rotor is: c_1= 300 m/s.
2) The steam flow has an angle of entrance of α_1= 14º.
3) The turbine works at 3000 rpm.
4) As is an action stage, $|w_1| = |w_2|$ and then $|\beta_1| = |\beta_2|$.
5) It is easy to see that u_1=u_2.

The equations to calculate the velocity triangles are the following:

Entrance triangle:

a) $c_1 * \sin \propto_1 = w_1 * \sin \beta_1$

b) $c_1 * \cos \propto_1 = u_1 + w_1 * \cos \beta_1$

c) $w_1 = \frac{\sin \propto_1}{\sin \beta_1} * \cos \beta_1 * c_1$

Introducing (c) in (b) and clearing u_1:

d) $u_1 = c_1 * \cos \propto_1 - c_1 * \sin \propto_1 * \cot \beta_1$

Exit triangle:

e) $c_2 * \sin \propto_2 = c_1 * \sin \propto_1$

f) $c_2 * \sin \propto_2 = w_2 * \sin \beta_2$

g) $u_2 = w_2 * \cos\beta_2$

Introducing first (f) in (g) and then (e) in the result:

h) $u_2 = c_2 * \cot\beta_2 * \sin\alpha_2 = c_1 * \sin\alpha_1 * \cot\beta_2$

To approximate to the best triangle to the stage, the first triangle will have α_2= 90º and as u_1=u_2, (d) =(h), simplifying this equality , having in count that $|\beta_1| = |\beta_2|$, finally it is possible to arrive to:

i) $\tan\beta = 2 * \tan \propto_1$

Then as α_1= 14º:

β= 26.50º

c_2= 72.50 m/s

u_1=u_2= 145.6 m/s

Knowing n=3000 rpm, then the angular speed will be $\omega = \frac{2*\pi*n}{60} = 100\pi$ rad/s and then the diameter of the stage would be $D = 2 * \frac{u}{\omega}$ =0.93 m. Having this diameter it is possible to calculate the length of the blades of the stage, so using:

$$L \cong \frac{\dot{m}}{\pi * D * \rho * c_1 * \sin\alpha_1}$$

The mass flow is $\dot{m}$= 150 kg/s and the density at the entrance conditions is ρ= 50.41 kg/m^3, so with these values the length obtained is L= 0.014 m.

Once are calculated all the parameters related with this first triangles, the next step is to build some different triangles with diameter of the same magnitude, and then analyze which of them is the best. The angle of entrance (α_1) and the speed at the entrance of the rotor (c_1) don´t change, but the other parameters they do. So using the previous equations the results are:

D (m)	U (m/s)	β (º)	W (m/s)	α_2 (º)	C_2 (m/s)	L (m)
0.93	145.6	26.5	162.65	90	72.57	0.014
0.8	125.66	23.69	180.63	61.29	82.74	0.016
0.7	109.96	21.84	195.09	45.57	101.63	0.019
0.6	94.25	20.23	209.89	35.25	125.75	0.022
0.5	78.54	18.85	224.63	28.43	152.44	0.026

The next images are the draws in scale of the 5 different triangles:

D= 0.93 m

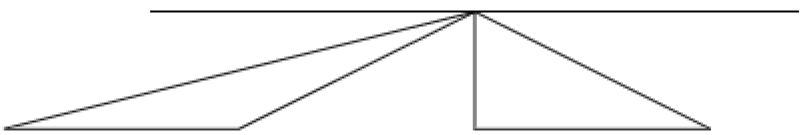


Fig 13. Triangle of velocities for D= 0.93 m. Creation of the author.

D= 0.8 m

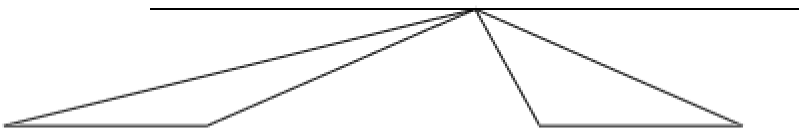


Fig 14. Triangle of velocities for D= 0.8 m. Creation of the author.

D= 0.7 m

Fig 15. Triangle of velocities for D= 0.7 m. Creation of the author.

D= 0.6 m

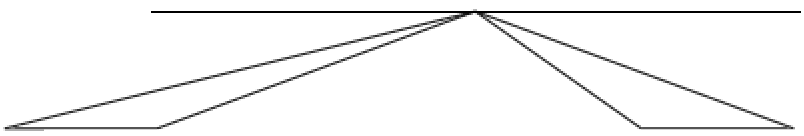


Fig 16. Triangle of velocities for D= 0.6 m. Creation of the author.

D= 0.5 m

Fig 17. Triangle of velocities for D= 0.5 m. Creation of the author.

Analyzing all the different triangles, the most suitable triangle is which has the diameter of 0.6 m and the length of the blades of 0.022 m.

So once established the triangle and the rest of parameters, the next thing to do is to calculate the losses in both the stator and rotor. For this, looking in the atlas of blades, model C-9009A was chosen for the stator, and model P-2617A for the rotor. Searching in the charts and graphics of the book it is possible to appreciate that the percentage of losses in the stator is ξ_s=3.5% and for the rotor is ξ_r= 6%. The graphics for the stator blade C-9009A is showed in annex I, and the graphics for the rotor blade P-2617A are in annex J.

Then the losses for the stator and the rotor are:

Stator: $\Delta_S = \frac{c_0^2}{2} * \xi_s$ J/kg (9)

Rotor: $\Delta_R = \frac{w_1^2}{2} * \xi_r$ J/kg (10)

To calculate c_0, which is the speed at the entrance of the stator the next equation is valid:

$$c_0 = \frac{\dot{m}}{\rho_{0 * \pi * D * L}} \quad (11)$$

And in this case c_0= 71.75 m/s. So the losses are:

Δ_s= 0.09 kJ/kg

Δ_r= 1.32 kJ/kg

At the entrance point, the pressure is 164.64 bar, the enthalpy is h_1=3403.1 kJ/kg , the entropy is s_1=6.426 kJ/kg.K and the kinetic energy is $c_0^2/2$= 2.57kJ/kg. The pressure at the exit of the stage is 90 bar, the ideal enthalpy at the exit is h_{2s}=3214.28, and adding the losses the real enthalpy is obtained, so $h_2 = h_{2s} + \Delta_s + \Delta_r$= 3215.69. The kinetic energy at this point is $c_2^2/2$= 7.91 kJ/kg.

In the next graphic it is possible to appreciate all the process that follows the steam flow in the stage:

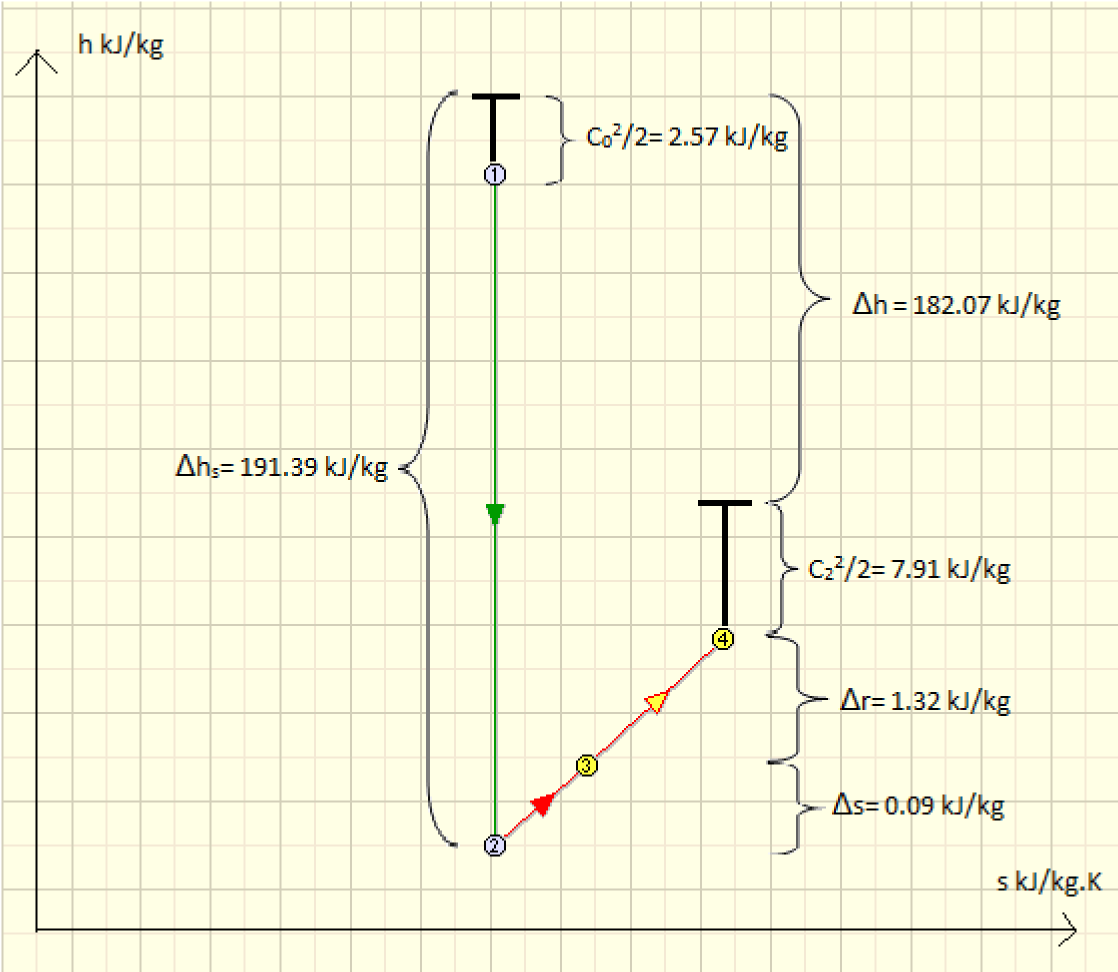


Fig 18. Process of expansion. Creation of the author

As the graphic shows:

$\Delta h_s=(h_1+ c_0^2/2) - (h_{2s})= 191.39$ kJ/kg

$\Delta h=(h_1+ c_0^2/2) - (h_2+c_2^2/2)= 182.07$ kJ/kg

Finally the efficiency of the stage is:

$$\eta = \frac{\Delta h}{\Delta h_s}*100= 95.13\%$$

BIBLIOGRAPHY

Figures:

- Figures 1, 2, 3, 4, 5 and 6: “Fundamentals of engineering thermodynamics” , Moran J.; 5th edition, 2006
- Figures 7 and 9: “Applied thermodynamics”. Onkar Singh; 3th edition, 2009.
- Figures 8 and 11: “Control and governing of steam turbines”. University of Buenos Aires; Own edition, 1998.
- Figure 10: “Regulation and control devices of steam turbines”. University of Cantabria; Own edition, 2001.
- Figures 12, 13,14,15,16,17 and 18: Creation of the author.
- Annexes C, D, E, F, G and H: Creation of the author.
- Annexes I and J: Atlas of the turbine blades.

Consulted books:

- “Fundamentals of engineering thermodynamics” Moran, J.; 5th edition, 2006
- “Applied thermodynamics”. Onkar, Singh; 3th edition, 2009.
- “Control and governing of steam turbines”. University of Buenos Aires, Own edition, 1998.
- “Regulation and control devices of steam turbines”. University of Cantabria, Own edition, 2001.
- “Thermal Turbomachines”. Mataix, Claudio; 3th edition, 1991.
- “Fundamentals of themodynamcs” Van Wylen, 6th edition, 2006.
- “Thermodynamics an engineering approach” Çengel, Yunus; 5th edition, 2006.

ANNEX A : STODOLA CONSTANTS

In order to calculate the new pressures for the different mass flows, Stodola equation is used:

$$C := \frac{m}{\sqrt{p_1^{\,2} - p_2^{\,2}}}$$

where p_1 is the pressure of enter at the stage and p_2 is the pressure at the exit. The Stodola constant, C, is different for each part of each turbine, and it keeps constant when the mass flow is changed.

So the process is the following, first Stodola constants are calculated using the nominal mass flow (150 kg/s), and the nominal pressures:

LP TURBINE:

4th Stage:

$p_{27} := 0.175$ bar

$p_7 := 0.04$ bar

$$C_{LP4S} := \frac{m}{\sqrt{p_{27}^{\,2} - p_7^{\,2}}} = 880.451$$

3th Stage:

$p_{26} := 0.39$ bar

$p_{27} := 0.175$ bar

$$C_{LP3S} := \frac{m}{\sqrt{p_{26}^{\,2} - p_{27}^{\,2}}} = 430.376$$

2nd Stage:

$p_{25} := 1.529$ bar

$p_{26} := 0.39$ bar

$$C_{LP2S} := \frac{m}{\sqrt{p_{25}{}^2 - p_{26}{}^2}} = 101.459$$

1 st Stage:

$p_6 := 4.184$ bar

$p_{25} := 1.529$ bar

$$C_{LP1S} := \frac{m}{\sqrt{p_6{}^2 - p_{25}{}^2}} = 38.515$$

IP TURBINE

3th Stage:

$p_{23} := 11$ bar

$p_6 := 4.184$ bar

$$C_{IP3S} := \frac{m}{\sqrt{p_{23}{}^2 - p_6{}^2}} = 14.745$$

2nd Stage:

$p_{22} := 22$ bar

$p_{23} := 11$ bar

$$C_{IP2S} := \frac{m}{\sqrt{p_{22}{}^2 - p_{23}{}^2}} = 7.873$$

2nd Stage:

$p_{22} := 22$ bar

$p_{23} := 11$ bar

$$C_{IP2S} := \frac{m}{\sqrt{p_{22}^{\ 2} - p_{23}^{\ 2}}} = 7.873$$

1st Stage:

$p_5 := 40.57$ bar

$p_{22} := 22$ bar

$$C_{IP1S} := \frac{m}{\sqrt{p_5^{\ 2} - p_{22}^{\ 2}}} = 4.401$$

HP Turbine:

2nd Stage:

$p_{20} := 70$ bar

$p_3 := 46$ bar

$$C_{HP2S} := \frac{m}{\sqrt{p_{20}^{\ 2} - p_3^{\ 2}}} = 2.843$$

1st Stage:

$p_2 := 164.64$ bar

$p_{20} := 70$ bar

$$C_{HP1S} := \frac{m}{\sqrt{p_2^{\ 2} - p_{20}^{\ 2}}} = 1.007$$

Once all the constants are calculated the next step is for each different value of mass flow, to calculate the different new pressures. The pressure at the end of the low pressure turbine is the same in all cases, this pressure is 0.04 bar, then the calculation starts from the last stage of this turbine and continues until arrive at the first stage of the high pressure turbine.

So, for the mass flow 130 kg/s:

LP Turbine:

4th Stage:

$p_7 := 0.04$ bar

$$p_{27'} := \sqrt{\frac{m^2}{C_{LP4S}^{\,2}} + p_7^{\,2}} = 0.153 \quad \text{bar}$$

3 th Stage:

$$p_{26'} := \sqrt{\frac{m^2}{C_{LP3S}^{\,2}} + p_{27'}^{\,2}} = 0.339 \quad \text{bar}$$

2 nd Stage:

$$p_{25'} := \sqrt{\frac{m^2}{C_{LP2S}^{\,2}} + p_{26'}^{\,2}} = 1.325 \quad \text{bar}$$

1 st Stage:

$$p_{6'} := \sqrt{\frac{m^2}{C_{LP1S}^{\,2}} + p_{25'}^{\,2}} = 3.626 \quad \text{bar}$$

IP turbine:

3 th stage;

$$p_{23'} := \sqrt{\frac{m^2}{C_{IP3S}^2} + p_{6'}^2} = 9.533 \quad \text{bar}$$

2 nd Stage:

$$p_{22'} := \sqrt{\frac{m^2}{C_{IP2S}^2} + p_{23'}^2} = 19.067 \quad \text{bar}$$

1 st Stage:

$$p_{5'} := \sqrt{\frac{m^2}{C_{IP1S}^2} + p_{22'}^2} = 35.161 \quad \text{bar}$$

It is important to notice the loss of 2% of pressure in the regulation valve between states 4 and 5, and the loss of pressure of 4.6 bar in the reheater, so:

$$p_{4'} := \frac{p_{5'}}{0.98} = 35.878 \quad \text{bar}$$

$$p_{3'} := p_{4'} + 4.6 = 40.478 \quad \text{bar}$$

HP Turbine:

2 nd Stage:

$$p_{20'} := \sqrt{\frac{m^2}{C_{HP2S}^2} + p_{3'}^2} = 61.07 \quad \text{bar}$$

1 st Stage:

$$p_{2'} := \sqrt{\frac{m^2}{C_{HP1S}^{\ 2}} + p_{20'}^{\ 2}} = 142.86 \quad \text{bar}$$

Then the same process is done but with mass flows equal to 110 kg/s:

LP Turbine:

4th Stage:

$$p_7 := 0.04 \quad \text{bar}$$

$$p_{27'} := \sqrt{\frac{m^2}{C_{LP4S}^{\ 2}} + p_7^{\ 2}} = 0.131 \quad \text{bar}$$

3 th Stage:

$$p_{26'} := \sqrt{\frac{m^2}{C_{LP3S}^{\ 2}} + p_{27'}^{\ 2}} = 0.287 \quad \text{bar}$$

2 nd Stage:

$$p_{25'} := \sqrt{\frac{m^2}{C_{LP2S}^{\ 2}} + p_{26'}^{\ 2}} = 1.122 \quad \text{bar}$$

1 st Stage:

$$p_{6'} := \sqrt{\frac{m^2}{C_{LP1S}^{\ 2}} + p_{25'}^{\ 2}} = 3.068 \quad \text{bar}$$

IP turbine:

3 th stage;

$$p_{23'} := \sqrt{\frac{m^2}{C_{IP3S}^{\,2}} + p_{6'}^{\,2}} = 8.067 \quad \text{bar}$$

2 nd Stage:

$$p_{22'} := \sqrt{\frac{m^2}{C_{IP2S}^{\,2}} + p_{23'}^{\,2}} = 16.133 \quad \text{bar}$$

1 st Stage:

$$p_{5'} := \sqrt{\frac{m^2}{C_{IP1S}^{\,2}} + p_{22'}^{\,2}} = 29.751 \quad \text{bar}$$

It is important to notice the loss of 2% of pressure in the regulation valve between states 4 and 5, and the loss of pressure of 4.6 bar in the reheater, so:

$$p_{4'} := \frac{p_{5'}}{0.98} = 30.359 \quad \text{bar}$$

$$p_{3'} := p_{4'} + 4.6 = 34.959 \quad \text{bar}$$

HP Turbine:

2 nd Stage:

$$p_{20'} := \sqrt{\frac{m^2}{C_{HP2S}^{\,2}} + p_{3'}^{\,2}} = 52.147 \quad \text{bar}$$

1 st Stage:

$$p_{2'} := \sqrt{\frac{m^2}{C_{HP1S}^2} + p_{20'}^2} = 121.084 \quad \text{bar}$$

ANNEX B : ITERATION PROCESS

Once the states of the extractions are calculated for the different types of regulation and different mass flow value, the following step is to calculate the quantity of the mass extraction in each stage, and the states of the other points of the cycle. To get this, a iteration process of calculation was used. It has been used three iterations for each cycle. Before to start the process, the thing to do is to establish the enthalpies of the exits of the closed exchangers, the pressure of work of these exchangers are known, so as at the exits of the exchangers there is saturated liquid, the enthalpies are also known, and these states are then completely defined.

One iteration of the process is:

Calculation of the mass flow extracted in the different extraction points:

Exchanger number 8:

$$m_{HP1S} := m - m_{FB} - m_{HP2S} - m_{IP1S} - m_{IP2S}$$

Exchanger number 7:

$$m_{HPS2} := \frac{m \cdot (h_{18'} - h_{17'}) - m_{HP1S} \cdot h_{18'} + h_{29'} \cdot m_{HP1S}}{h_{21'} - h_{29'}}$$

Exchanger number 6:

$$m_{IP1S} := \frac{m \cdot (h_{17'} - h_{16'}) - h_{29'} \cdot (m_{HP1S} + m_{HP2S}) + h_{30'} \cdot (m_{HP1S} + m_{HP2S})}{h_{22'} - h_{30'}}$$

Open feedwater:

$$m_{FB} := m - (m_{HP1S} + m_{HP2S} + m_{IP1S}) - m_{IP2S}$$

$$m_{IP2S} := \frac{m \cdot h_{15'} - m_{FB} \cdot h_{14'} - h_{30'} \cdot (m_{HP1S} + m_{HP2S} + m_{IP1S})}{h_{23'}}$$

Exchanger number 4:

$$m_{IP3S} := \frac{m_{FB} \cdot (h_{13'} - h_{12'})}{h_{24'} - h_{31'}}$$

Exchanger number 3:

$$m_{LP1S} := \frac{m_{FB} \cdot (h_{12'} - h_{11'}) - m_{IP3S} \cdot h_{31'} + m_{IP3S} \cdot h_{32'}}{h_{25'} - h_{32'}}$$

Exchanger number 2:

$$m_{LP2S} := \frac{m_{FB} \cdot (h_{11'} - h_{10'}) - h_{32''} \cdot (m_{IP3S} + m_{LP1S}) + h_{33''} \cdot (m_{IP3S} + m_{LP1S})}{h_{26'} - h_{33'}}$$

Exchanger number 1:

$$m_{LP3S} := \frac{m_{FB} \cdot (h_{10'} - h_{9'}) - h_{33''} \cdot (m_{IP3S} + m_{LP1S} + m_{LP2S}) + h_{34''} \cdot (m_{IP3S} + m_{LP1S} + m_{LP2S})}{h_{27'} - h_{34'}}$$

Once all the mass flow extractions are calculated, the next step is to calculate the different points around the exchangers, using these new values of mass flow. As the pressures are known, because them had been calculated before, with only calculating the enthalpies of the states, it is enough to define completely these states. It is important to notice that the enthalpies in the extraction points are already calculated. So the equations used are:

First pump:

In the first pump the enthalpies of the states 8 and 9 are going to be the same in all the cases, because in the state 8, the exit of the condenser, there is always saturated liquid at 0.04 bar. The efficiency of the pump will be the same in the different circuits, as it was said in the assumptions; so as the exit pressure is the same too, 25 bar, the state 9 it would be the same always.

$$h_{8'} := 121.4 \quad kJ/(kg.K)$$

$$h_{9'} := 124.7 \quad kJ/(kg.K)$$

Exchanger number 1:

$$h_{10'} := \frac{h_{33''} \cdot (m_{IP3S} + m_{LP1S} + m_{LP2S}) + m_{LP3S} \cdot h_{27'} - (m_{IP3S} + m_{LP1S} + m_{LP2S} + m_{LP3S}) \cdot h_{34'}}{m_{FB}} + h_{9'}$$

Exchanger number 2:

$$h_{11'} := \frac{h_{32''}\cdot(m_{IP3S} + m_{LP1S}) + h_{26''}\cdot(m_{LP2S}) - h_{33''}\cdot(m_{IP3S} + m_{LP1S} + m_{LP2S})}{m_{FB}} + h_{10'}$$

Exchanger number 3:

$$h_{12'} := \frac{m_{IP3S}\cdot h_{31'} + m_{LP1S}\cdot h_{25'} - h_{32''}\cdot(m_{IP3S} + m_{LP1S})}{m_{FB}} + h_{11'}$$

Exchanger number 4:

$$h_{13'} := \frac{m_{IP3S}\cdot(h_{24'} - h_{31'})}{m_{FB}} + h_{12'}$$

Open feedwater:

The pressure of the open feedwater depends on the case, this pressure is the pressure of the second extraction in the intermediate pressure turbine, and it is already calculated. So the enthalpy of the 15 it depends only in that pressure because as it was before said, in this point there is saturated liquid. The enthalpy of the state 14 it is the same than the one of the state 13, the difference between both states is the pressure because between both states there is an isenthalpic valve.

Second pump:

The efficiency of this pump is known η_{SP}= 0.7554, and the pressure of exit too, 210 bar; so knowing the state 15, it is possible to know the ideal state 16. The enthalpy of state number 15 depends on the pressure of the open feedwater. So there is:

$$h_{16'} := \frac{h_{16'id} - h_{15'}}{n_{SP}} + h_{15'}$$

Exchanger number 6:

$$h_{17'} := \frac{(m_{HP1S} + m_{HP2S})\cdot h_{29'} + m_{IP1S}\cdot h_{22'} - (m_{HP1S} + m_{HP2S} + m_{IP1S})\cdot h_{30'}}{m} + h_{16'}$$

Exchanger number 7:

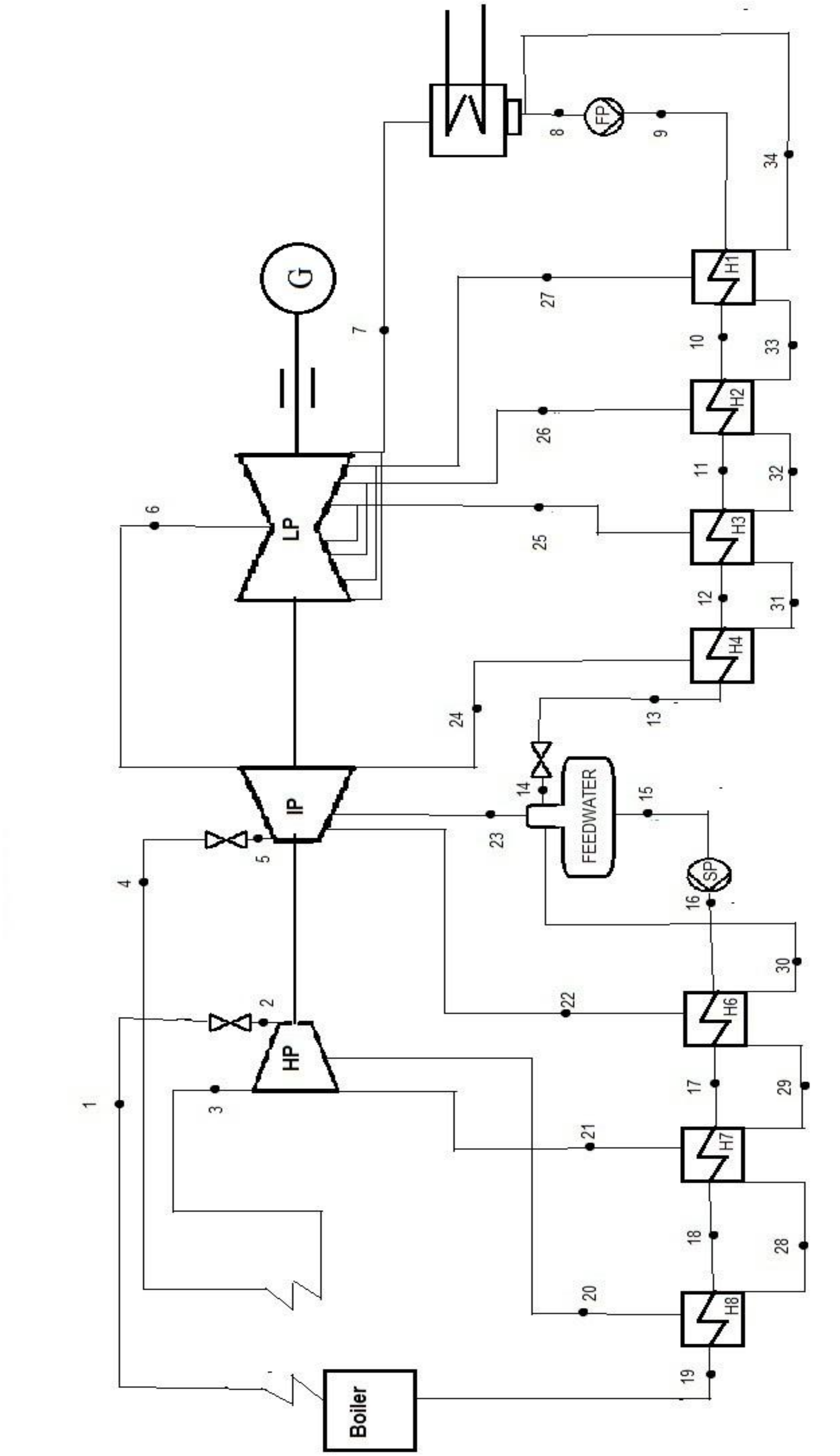
$$h_{18'} := \frac{m_{HP1S}\cdot h_{28'} + m_{HP2S}\cdot h_{21'} - (m_{HP1S} + m_{HP2S})\cdot h_{29'}}{m} + h_{17'}$$

Exchanger number 8:

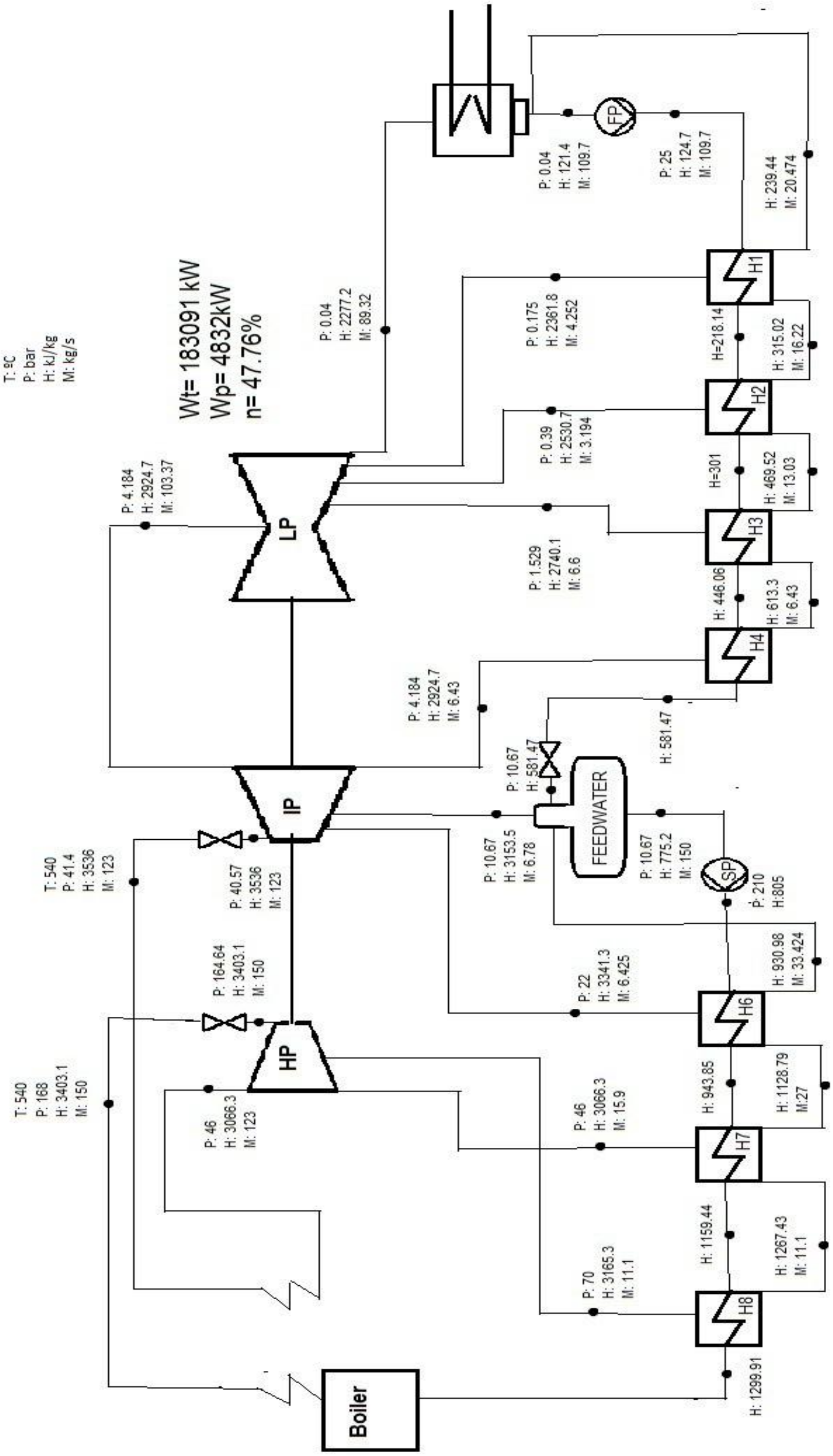
$$h_{19'} := \frac{m_{HP1S} \cdot (h_{20'} - h_{28'})}{m} + h_{18'}$$

To start this iterative process, in the first iteration the values of mass flow of the nominal cycle are used. Three iterations per cycle have been made.

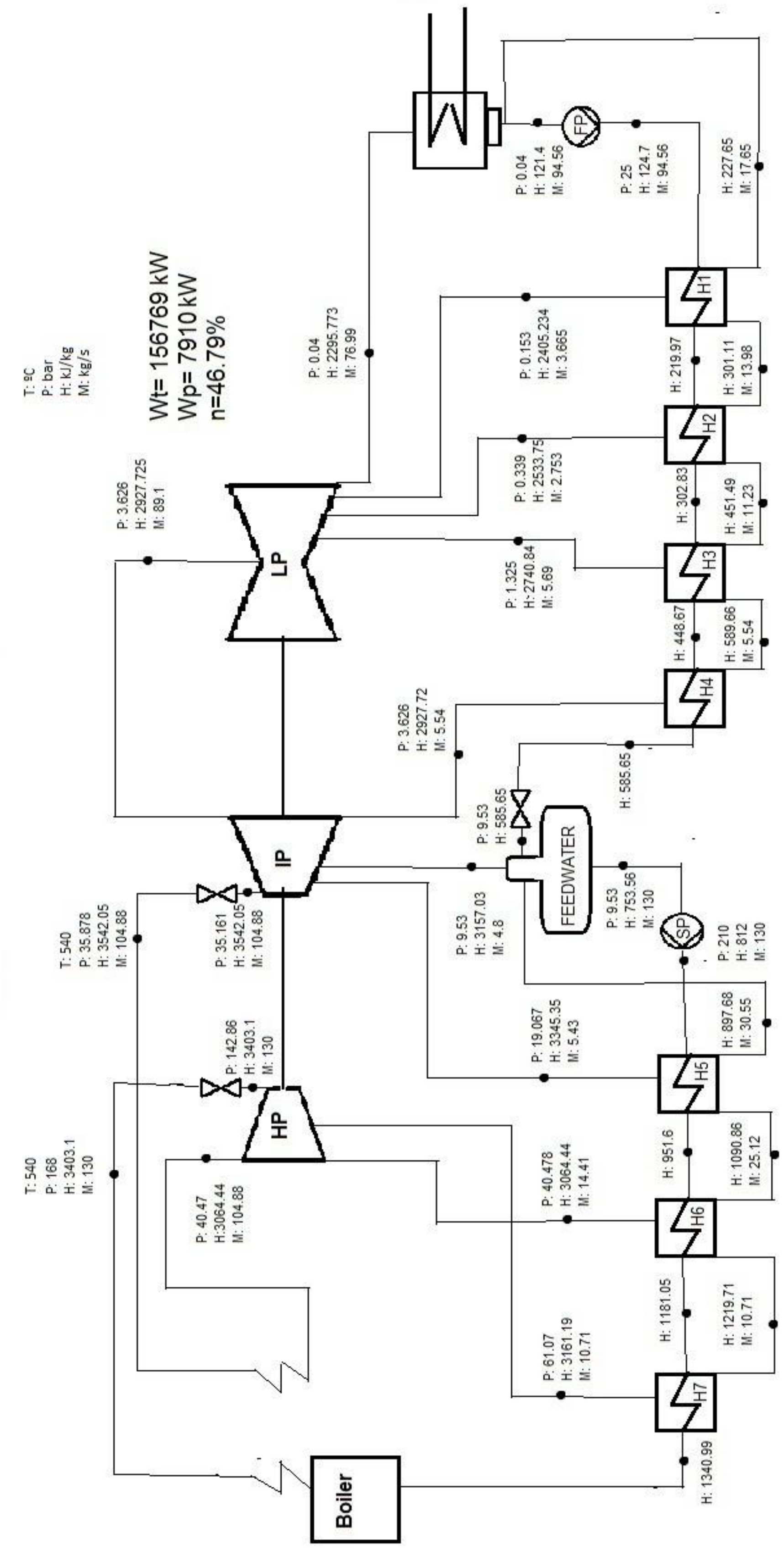
ANNEX C: NUMBERED STATES



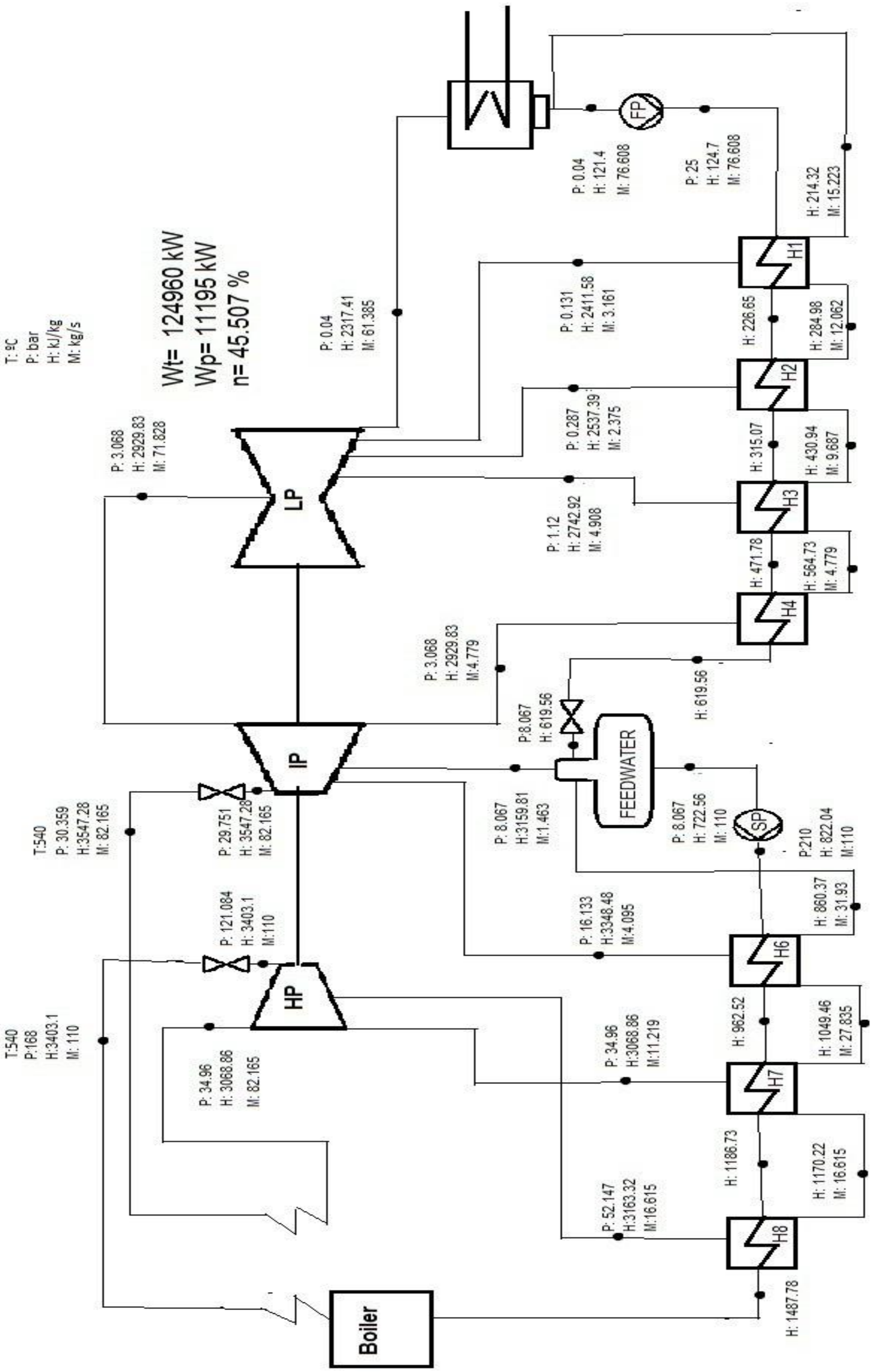
ANNEX D: NOMINAL CYCLE m=150 kg/s



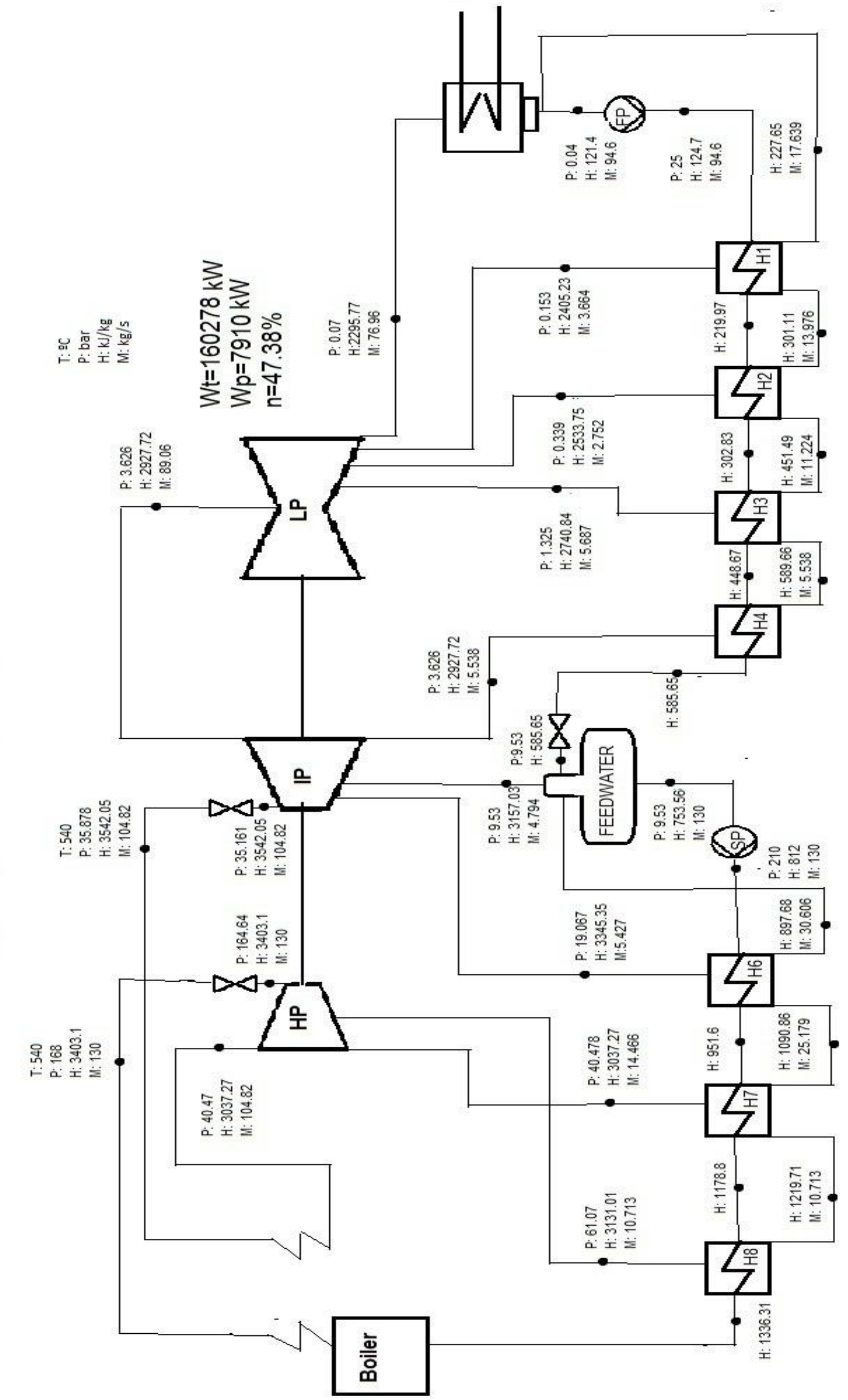
ANNEX E: THROTTLING GOVERNING m=130 kg/s



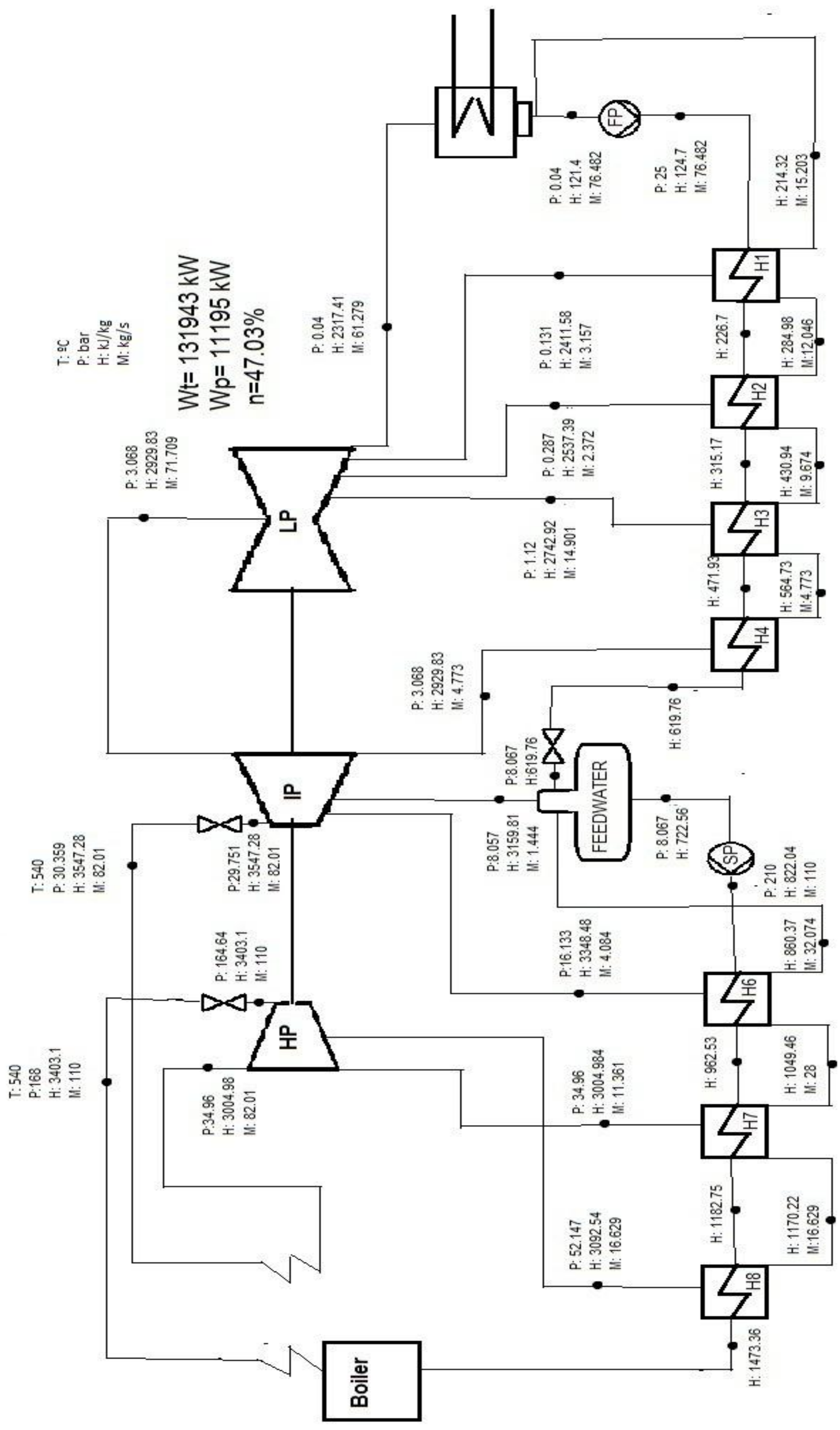
ANNEX F: THROTTLING GOVERNING m=110 kg/s



ANNEX G: NOZZLE GOVERNING m=130 kg/s



ANNEX H: NOZZLE GOVERNING m=110 kg/s



ANNEX I: STATOR BLADE C-9009A GRAPHICS

ANNEX J: ROTOR BLADE P-2617A GRAPHICS