NTech Lehrstuhl für Nukleartechnik

GRS

Technische Universität München

Master Thesis

von cand. Ing. JORDI BASSAS
Matrikelnummer: 03283589

Development and implementation of a Nuclear Power Plant steam turbine model in the system code ATHLET

Betreuer TUM: Prof. Dr. Rafael Macián-Juan
Betreuer GRS: Dipl.-Phys. Philipp Schöffel
Ausgegeben: 15.01.2011
Abgegeben: 22.07.2011

Erklärung

Hiermit versichere ich, die vorliegende Arbeit selbstständig und ohne Hilfe Dritter angefertigt zu haben. Gedanken und Zitate, die ich aus fremden Quellen direkt oder indirekt übernommen habe, sind als solche kenntlich gemacht. Diese Arbeit hat in gleicher oder ähnlicher Form noch keiner Prüfungsbehörde vorgelegen und wurde bisher nicht veröffentlicht.

Ich erkläre mich damit einverstanden, dass die Arbeit durch den Lehrstuhl für Nukleartechnik der Öffentlichkeit zugänglich gemacht werden kann.

München, den 22. Juli 2011

JORDI BASSAS

Abstract

In order to improve the simulation of the whole secondary loop with the system code ATHLET a steam turbine model has to be implemented. This paper deals with the development of a thermo-hydraulic model of a Nuclear Power Plant steam turbine and its implementation in the system code ATHLET.

The model is based on Stodola's cone law and simulates the pressure drop and the enthalpy drop along the different turbine stages as well as the steam and water extractions.

The influence of the steam and water extractions on the turbine behaviour as well as the importance of an accurate model for the steam and water extractions are carefully explained.

Heat and mass balances of the Nuclear Power Plant Philippsburg 2 are used for reference purposes as well as for validation purposes of the implemented model. The comparison between steady state simulations and the real plant data indicate a satisfactory accuracy of the model and of the thermodynamic approach used.

List of Contents

List of Figures

List of Tables

Acknowledgments

This paper was developed during my practicum at GRS.

I want to thank my supervisors in GRS, Dipl.-Phys. Philipp Schöffel and Dr. Ing. Fabian Weyermann for their constant support and invaluable help during the development of this paper as well as to Dipl. Ing. (FH) Georg Lerchl for his help with ATHLET. My gratitude also to my tutor Professor Rafael Macián-Juan from the Technische Universität München for his support during my progress and for making the development of my Master Thesis at GRS possible. Thanks also to all the workers and friends at GRS for their readiness to help me at any time and for their support.

I also would like to express to Ms Imogen Helen Sexton Kakuschky my sincere gratitude for her correction of this Master Thesis and for her support.

Finally I want to thank my friends and family for being there, and my parents for making it all possible.

List of Acronyms

A	Area	m^2
BWR	Boiling Water Reactor	-
c	Absolute velocity	m/s
CV	Control Volume	-
f	Frequency	Hz
G	Mass flow through a junction in ATHLET	kg/s
h	Enthalpy	J/kg
h^*	Total enthalpy	J/kg
HP	High Pressure	-
LP	Low Pressure	-
LWR	Light Water Reactor	-
$\dot{m}$	Mass flow	kg/s
M	Mass (in ATHLET equations)	kg
M	Torque	Nm

n	Polytropic exponent	kg/s
N	Rotational Speed	1/min
NPP	Nuclear Power Plant	-
p	Pressure	bar
P	Power	W
PWR	Pressurized Water Reactor	-
Q	Heat	J
s	Entropy	J/(kg·K)
SJP	Single Junction Pipe	-
r	Evaporation Enthalpy	-
$RBMK$	Graphite-moderated boiling water reactor (Russian type)	-
T	Temperature	K
TDV	Time Dependent Volume	-
TFD	Thermo-Fluid dynamic	-
TFO	Thermo-Fluid dynamic Object	-
U	Internal energy	m^3/kg

v	Specific volume	m^3/kg
w	Fluid velocity	m/s
W	Work	J
x	Steam quality	J
α	Correction factor	-
α	Steam void fraction	-
η	Efficiency	-
Θ	Moment of inertia	kg/m^2
κ	Isentropic exponent	-
ρ	Density	kg/m^3
τ	Time constant	s
ψ	Interphase mass exchange per unit volume	$kg/s/m^3$
ω	Angular velocity	rad/s

1 Introduction

All the thermal power plants need an element to transform the heat power into electrical power. In a Nuclear Power Plant the heat produced by the nuclear fission is used to produce high pressure steam. This steam expands through a turbine in which the heat stored in the steam is transformed into mechanical energy used to drive a generator thus producing electricity.

In the field of nuclear safety, so called system codes (e.g. RELAP, TRACE, CATHARE or ATHLET) have been developed to simulate the behaviour of the plant. The aim of this paper is to develop a model for the steam turbine of a Nuclear Power Plant in the computer code ATHLET (acronym for Analysis of Thermal-hydraulics of Leaks and Transients) developed by the company Gesellschaft für Anlagen- und Reaktorsicherheit (GRS).

ATHLET is a 1-D best estimate code and therefore the whole cooling system including the steam turbine should be simulated with the maximum accuracy. In order to do that, and to be able to simulate the behaviour of the plant as a whole in situations such as full and partial load, and abnormal situations such as load rejections, and the operation of the plant supplying energy only for the plant itself isolated from the net, the development of the steam turbine model is necessary. Also the users of ATHLET have being asking for a turbine model in the past.

As the aim of this paper is to model the steam turbine of a Nuclear Power Plant in operation, a short and simple description of the basic NPP features will be given. The chosen thermodynamic approach, with the model delivering a pressure drop and a power extraction makes the presentation of the basic thermodynamic background necessary. The concepts of the Rankine cycle and its particularities for the case of a Nuclear Power Plant as well as the principles behind the operation of steam turbines will also be explained. The system code ATHLET will be presented and described in order to improve the understanding of the chosen approach.

Turbine manufacturers do not publish any relevant data about steam turbines, this makes the development of a model quite complicated. The goal of this paper is to develop a model which requires only data accessible by the final user.

The reference turbine used (the Low Pressure turbine of the Nuclear Power Plant Philippsburg 2) will be described and analysed and the assumptions and hypotheses made will be developed and justified.

The models developed will be explained and justified before alternative approaches are commented. Finally the implementation in ATHLET will be presented as well as the results of the simulations. The application range of the model as well as the possible extensions will be explained in the last part of this paper.

2 Steam turbines

2.1 The Rankine Cycle

The cycle described by the steam in a NPP is known as the Rankine cycle. In the Rankine cycle a working fluid is alternatively condensed at low pressure and evaporated at high pressure, water being the most common working fluid. Water steam is produced in a high pressure boiler and then expanded through a turbine (where the conversion into mechanical work is produced). The low pressure steam is condensed in a condenser. The condensate is then pumped into the boiler thus closing the cycle.

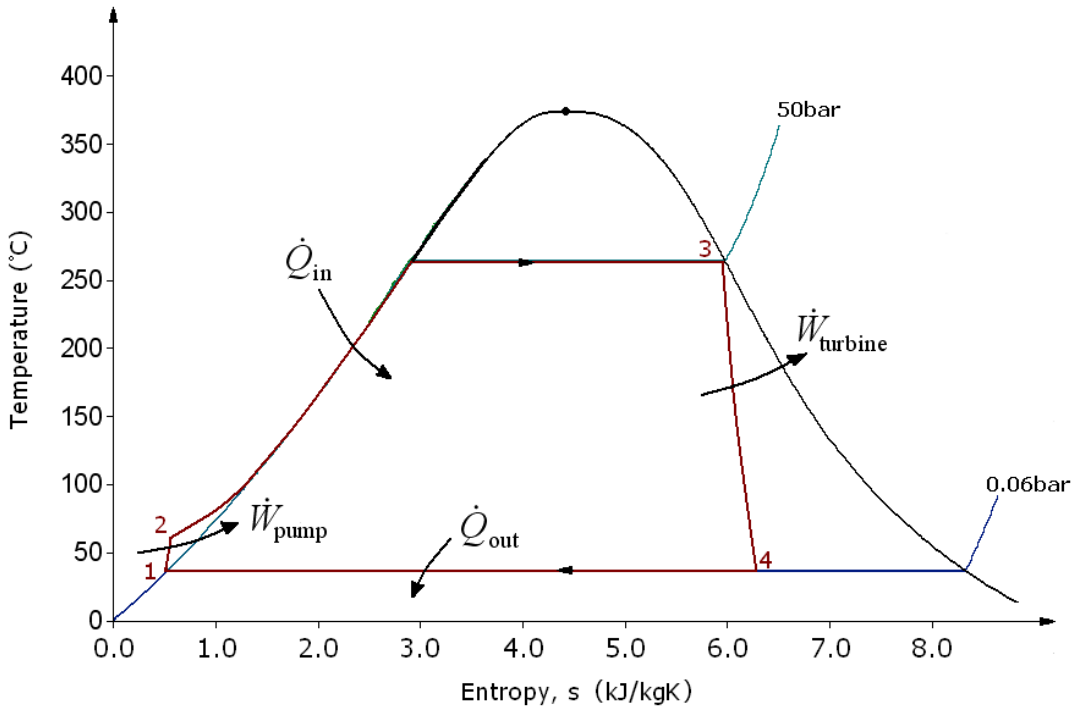

Figure 2.1 T-s diagram of the Rankine cycle (Ainsworth, 2007).

The Rankine cycle consists of four processes; the red numbers in Figure 2.1 indicate the different states:

Process 1-2: The pump compresses the working fluid from the low condenser pressure to the high boiler pressure.

Process 2-3: In the boiler the liquid is heated at constant pressure and evaporates producing saturated steam.

Process 3-4: The saturated steam expands through the turbine converting the heat power into mechanical power. The pressure and the temperature decrease; condensation occurs.

Process 4-1: The wet steam enters the condenser where it is condensed at constant pressure and constant temperature.

In the ideal turbine the expansion would be isentropic; however losses such as the friction between the steam and the turbine increase the entropy thus reducing the isentropic efficiency of the turbine.

The thermodynamic efficiency of the whole process is calculated by the formula:

$$\eta_{th} = \frac{|\dot{W}_{turb}| - |\dot{W}_{pump}|}{|\dot{Q}_{in}|} \tag{2.1}$$

$\dot{W}_{turb}$ is the mechanical power produced in the turbine, i.e. the power delivered by the system, $\dot{W}_{pump}$ the power used by the pump, i.e. the mechanical power consumed by the system and $\dot{Q}_{in}$ the heat power given to the system.

The calculation of the power delivered by the system or to the system is the difference between inlet and outlet enthalpies multiplied by the mass flow.

The formulas for the specific powers are:

$$\frac{\dot{W}_{turb}}{\dot{m}} = h_3 - h_4 \tag{2.2}$$

$$\frac{\dot{W}_{pump}}{\dot{m}} = h_1 - h_2 \tag{2.3}$$

$$\frac{\dot{W}_{pump}}{\dot{m}} = h_1 - h_2 \tag{2.4}$$

$$\frac{\dot{Q}_{in}}{\dot{m}} = h_2 - h_3 \tag{2.5}$$

The power produced by the system is positive and the power received by the system is negative.

The isentropic efficiency of the turbine compares the ideal (isentropic) enthalpy difference and the real enthalpy difference.

$$\eta_i = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{2.6}$$

The sub index s stands for the enthalpy corresponding to the isentropic process.

2.2 Types and construction of turbines

Large power plants use steam turbines for converting heat energy into mechanical energy which is converted into electric energy by the generator. A turbine transforms the internal energy of the fluid through kinetic energy into mechanical energy. Steam with high pressure and high temperature (potential energy) expands through the turbine thus reducing its temperature and pressure. The resulting enthalpy difference between the inlet and the outlet steam is converted into rotational energy on the shaft which spins a generator thus producing electricity.

If steam with high pressure and high temperature expands through a nozzle into a low pressure area, the pressure reduction will increase its velocity. In this way the enthalpy of the steam decreases but its kinetic energy increases, the total enthalpy however remains constant (provided that the expansion occurs without losses). The total enthalpy of a fluid (equation (2.7)) computes the energy content of the fluid as well as the kinetic energy of the streaming fluid.

$$h^* = h + \frac{c^2}{2} \tag{2.7}$$

In a turbine there are stationary blades which are fixed to the casing and moving blades which are fixed to the shaft. A line of stationary blades followed by its corresponding line of moving blades is called a stage (see Figure 2.2).

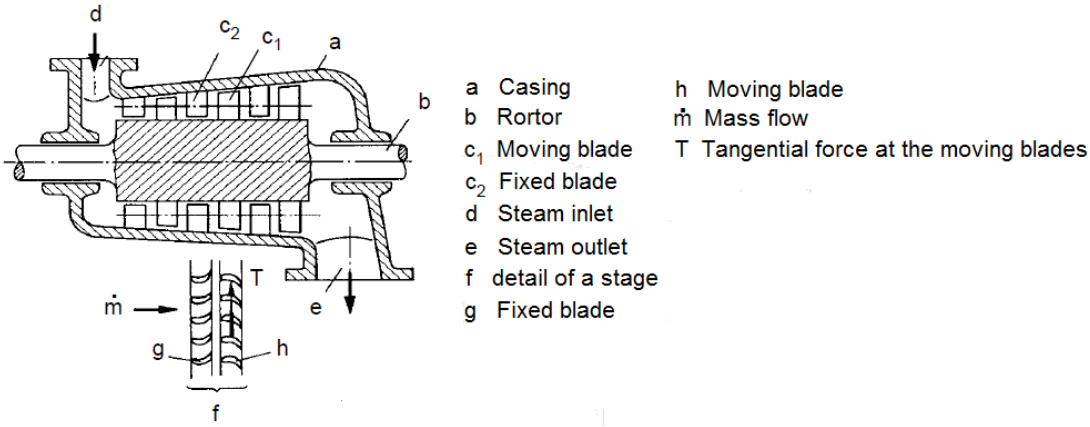

Figure 2.2 Schematic image of a turbine (Lehrstuhl für Energiesysteme , 2010) (Kleinedler, 2002).

The hot steam passes through the stationary blades increasing its velocity and reducing its pressure and then through the moving blades. The direction of the steam is changed both in the stationary as in the moving blades. The resulting reaction force on the moving blades produces the momentum that moves the shaft.

Depending on the part of the stage in which the pressure drop and the acceleration take place there:

- In impulse or action turbines, the complete enthalpy and pressure drop occurs in the stationary blades, in the moving blades only the steam direction changes.
- In reaction turbines the enthalpy and the pressure drops occur both in the fixed and in the moving blades.

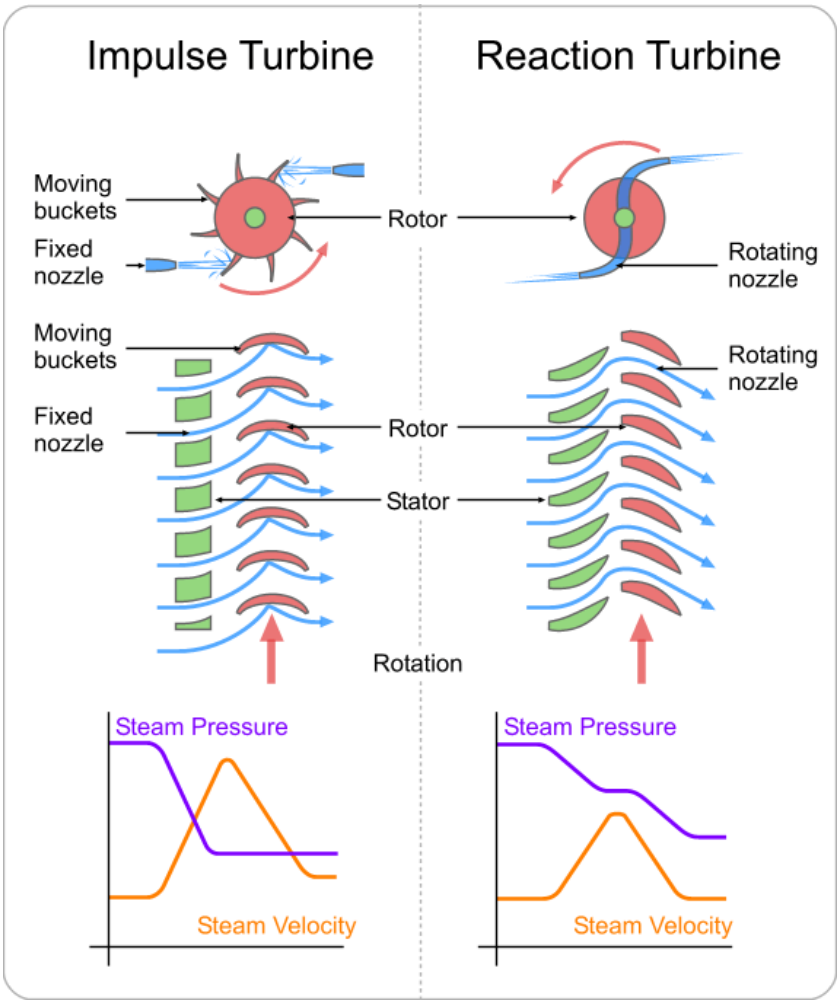

Figure 2.3 Difference between impulse and reaction turbine (Ainsworth, 2007)

Depending on the exhaust conditions turbines can be classified into:

Back pressure turbines, in which the exhaust pressure is atmospheric or higher.

Condensate turbines, in which the exhaust pressure is lower than the atmospheric pressure, normally close to vacuum.

2.3 Particularities of steam turbines in Nuclear Power Plants; the saturated steam process

2.3.1 Steam production in a Nuclear Power Plant

In a Pressurized Water Reactor (see Figure 2.4) the nuclear reaction takes place in the reactor pressure vessel where it heats the primary coolant, the hot primary coolant water goes through the steam generators were its heat is transferred to the lower pressure secondary loop water which evaporates to pressurized steam. The steam produced at the steam generators then expands partially through the HP-turbine. This steam then gets through a moisture separator which increases the steam quality and a re-heater which increases the temperature of the steam. This overheated steam enters into the LP-turbine, and exits to the condenser.

As the volumetric flow rises due to the steam expansion, so does the diameter of the turbine. However, the length of the blades is limited by the speed of sound, therefore, it is usual to divide the low pressure steam between 4 or 6 LP-turbines as it can be seen in Figure 2.4.

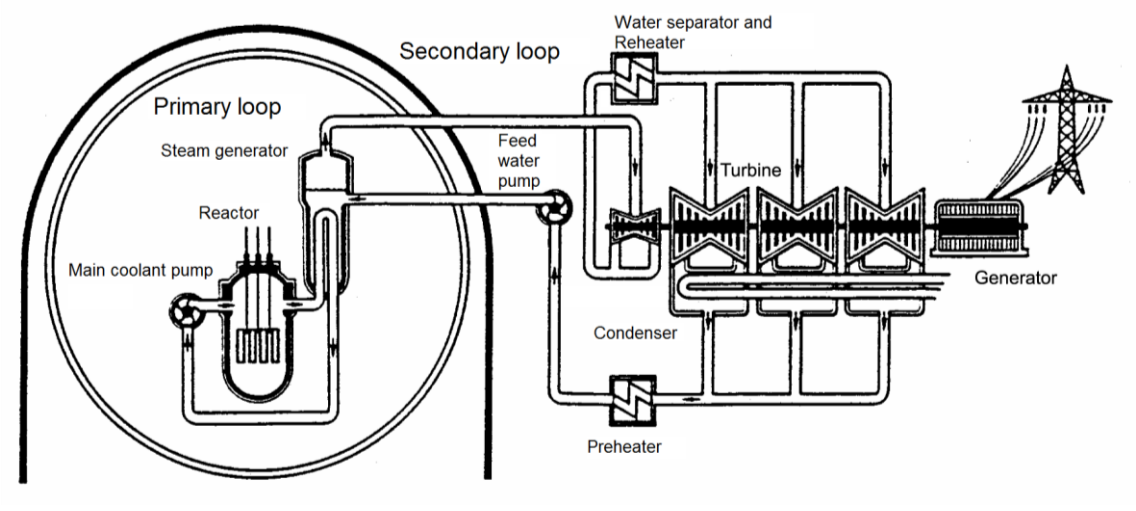

Figure 2.4 Schematic configuration of NPP (PWR type) (Kleinedler, 2002).

2.3.2 The saturated steam process

The main difference between a NPP and conventional thermal power plants is the fact that steam of NPP enters in the turbine as saturated steam and not as superheated steam. This has implications in the efficiency of the whole thermodynamic cycle and in the design of the turbine. These particularities apply to PWR as well as to BWR.

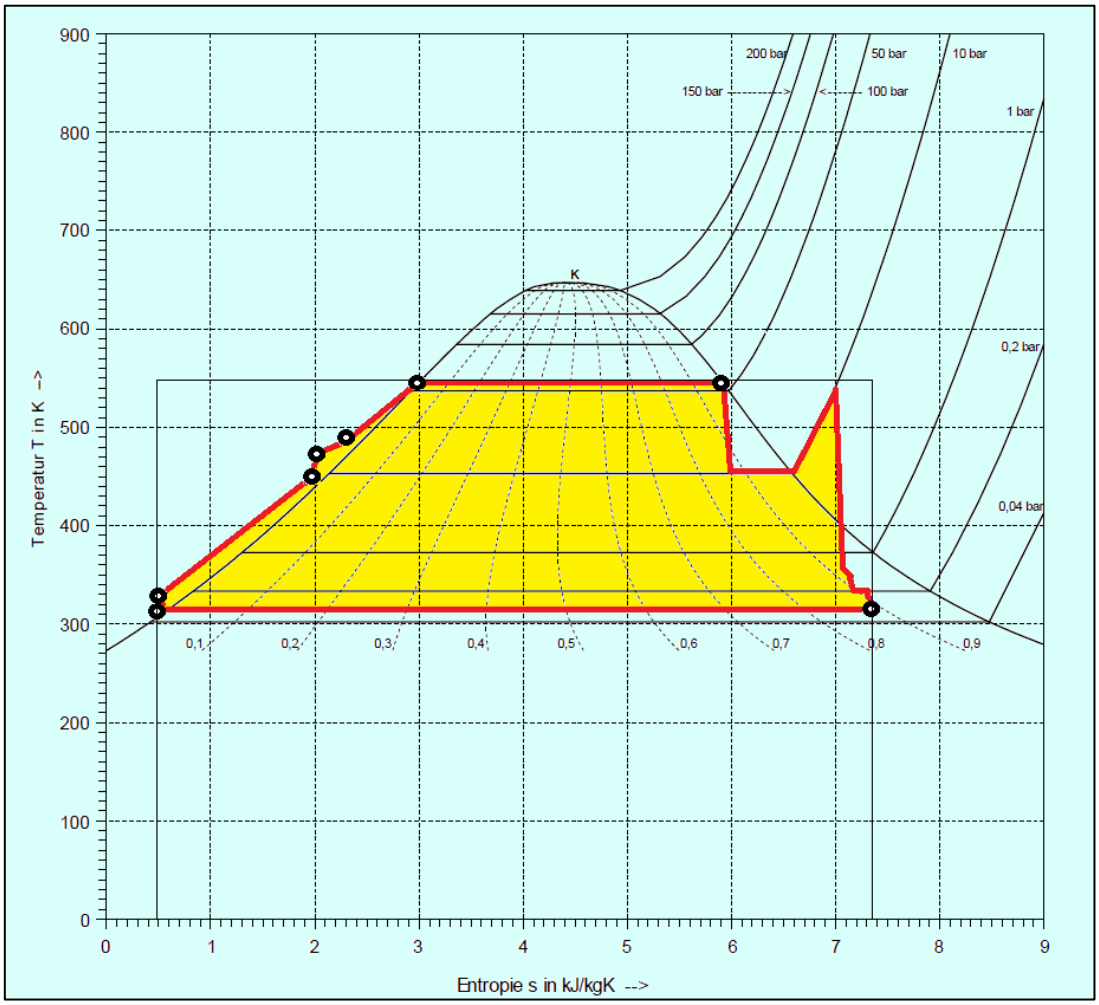

Figure 2.5 Schematic T-s diagram of the Rankine cycle in a NPP.

As it can be seen in Figure 2.5, the expansion occurs mostly in the saturated steam area. The saturated steam exits the steam generator and enters the HP-turbine. It ex-

pands and its thermal energy is transformed into kinetic energy and then into mechanical energy in the turbine as detailed before. This loss of heat of the steam results in condensation of the steam thus decreasing the steam quality. The humidity in form of water drops causes energy losses and, should it be allowed to increase, could result in a rapid erosion of the turbine blades. In order to avoid this, the water is partly extracted after every stage. This water extraction results in an increase of the specific enthalpy of the remaining steam as the water extracted has a much lower enthalpy than the steam. The fluid enthalpy, that is, the specific enthalpy multiplied by the steam mass has decreased in the amount of the absolute enthalpy of the extracted water.

Before entering the LP-turbine the steam gets through a moisture separator and a reheater, entering into the turbine as superheated steam. This increases the efficiency of the cycle; however this improvement is minimal (Strauß, 2006) as part of the high quality steam from the steam generator has to be used for this re-heating instead of expanding through the turbine. The main objective of the reheating is to minimize the appearance of moisture in the LP-turbine thus reducing the erosion of the blades.

A much more effective measure to improve the efficiency of the plant is the feed water preheating. This is done by extracting steam from the turbine and using it to preheat the feed water. The steam extracted from the low pressure stages from the turbine has all its condensation heat but only a fraction of its original capacity to perform work at the turbine.

3 Physical models

In order to simulate the turbine a series of key physical models have to be developed. The development of these models has to be directed towards the proper representation of those variables relevant for the purpose of the modelling. In this case, the model is expected to represent the turbine behaviour in the thermo-hydraulics simulation code ATHLET. In order to integrate this model into ATHLET, it has to provide a series of variables as pressure drop across the turbine, enthalpy drop across the turbine, power output, pressure at the extraction lines, etc.

Given the data available (see Chapter 4) and that only the above stated variables are necessary a detailed fluid dynamics model of the behaviour of the fluid through the moving and fixed blades is not necessary. Instead of that, a simpler thermodynamic approach is used.

3.1 Stodola's cone law

Stodola's Cone Law (Stodola, 1922), and its different versions (Traupel, 2001) display, given the design parameters, the relationship between the inlet- and outlet pressure at the turbine and the mass flow through the turbine.

The cone law equation is (Traupel, 2001):

$$\frac{\dot{m}}{\dot{m}_0} = \frac{p_a}{p_{a,0}} \sqrt{\frac{p_{a,0}\, v_{a,0}}{p_a\, v_a}} \sqrt{\frac{1 - \left(\frac{p_b}{p_a}\right)^{\frac{n+1}{n}}}{1 - \left(\frac{p_{b,0}}{p_{a,0}}\right)^{\frac{n+1}{n}}}} \tag{3.1}$$

'n' being the polytrophic exponent, 'p' the pressure and 'v' the specific volume. The sub index "a" stands for the inlet value, "b" for the outlet value and "0" for the design values.

For wet steam the calculation of the polytrophic exponent is (Traupel, 2001):

$$n = \frac{\kappa}{1 + \frac{\kappa\, p\left(v_{steam} - v_{liquid}\right)}{r}\left(1 - \eta_p\right)} \tag{3.2}$$

"κ" is the isentropic exponent, "v_{steam}" and "v_{liquid}" the specific volume of steam and water respectively, "r" the evaporation enthalpy and "η_p" the overall efficiency of the turbine.

The term $\frac{p\left(v_{steam} - v_{liquid}\right)}{r}$ depends only on the pressure (Traupel, 2001), taking a mean value of p for the range of pressures of the reference turbine:

$$\frac{p\left(v_{steam} - v_{liquid}\right)}{r} = 0{,}85$$

For wet steam and considering that the steam quality is never under 0.8, κ=1.135 (Grote, 2009) (Traupel, 2001).

The expansion polytrophic exponent for dry steam is (Ray, 1980), (Traupel, 2001):

$$\frac{n+1}{n} = \frac{\kappa\left(2 - \eta_p\right) + \eta_p}{\kappa} \tag{3.3}$$

For dry steam and for the range steam parameters in a NPP-turbine, κ=1.3.

If the approximation $n \approx 1$ is done (Grote, 2009), (Ray, 1980), (Stodola, 1922), (Traupel, 2001), (Zimmer, 2008) and considering the steam an ideal gas, the equation (3.1) can be simplified to:

$$\frac{\dot{m}}{\dot{m}_0} = \frac{p_a}{p_{a,0}} \sqrt{\frac{T_{a,0}}{T_a}} \sqrt{\frac{1 - \left(\frac{p_b}{p_a}\right)^2}{1 - \left(\frac{p_{b,0}}{p_{a,0}}\right)^2}} \tag{3.4}$$

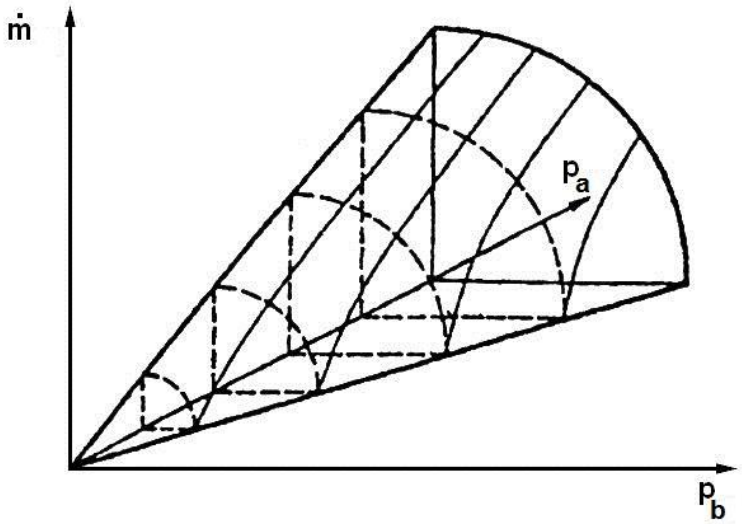

Figure 3.1 Graphic representation of the cone law (Stodola, 1922)

If the temperature varies moderately from the design temperature, the influence of the temperature is rather limited (about 5%). The same analogy can be done without considering the ideal gas simplification for the product of pressure and specific volume. However, in order to maintain the accuracy it has been chosen not to neglect the influence of temperature variations.

In order to describe the changes in the operating conditions, the mass flow has to be constant throughout the whole group of stages, making the application of equation (3.4) only possible in those sections of the turbine with the same mass flow i.e. stages between two consecutive extractions[1]. Steam turbines in NPP have several extraction lines which extract steam and/or condensate for feed water preheating and also to limit the quantity of condensate in the turbine (see Figure 3.2). This means that in order to describe faithfully the behaviour of the whole turbine, several interconnected sections will be necessary.

[1] For practical reasons, every group of stages will be referred to as a stage (see section 4.2).

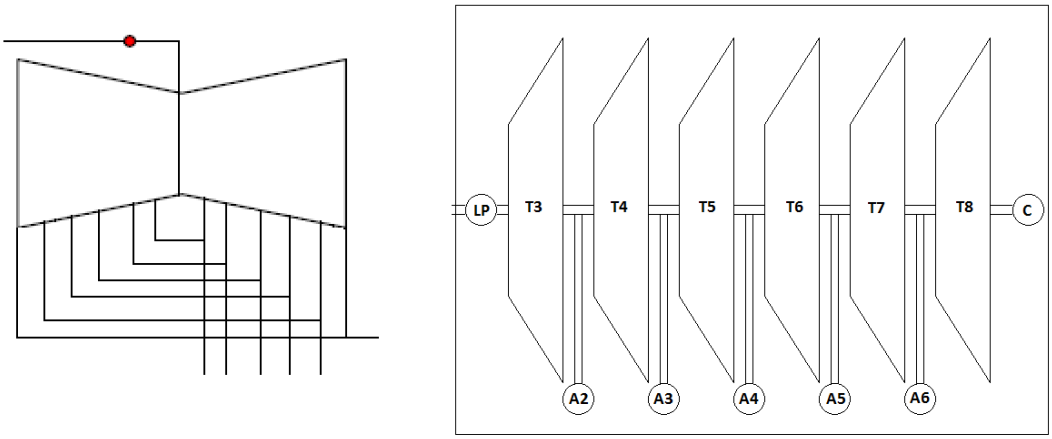

Figure 3.2 Turbine with several extraction lines, on the right the subdivision in sections can be seen.

3.2 Steam properties across the turbine

The thermodynamic state of water can be defined by two thermodynamic properties. For overheated steam or undercooled pressure and temperature give a definite state of the steam, for humid vapour, pressure and temperature are dependent on each other thus making the use of a third variable necessary, e.g. steam quality or specific volume.

The steam expansion through a turbine is a polytrophic process; therefore the isentropic enthalpy drop Δh_s has to be multiplied by an internal efficiency factor η_i, called isentropic efficiency (see equation (2.6)).

$$h_a - h_b = \eta_i (h_a - h_{bs}) \tag{3.5}$$

Or, what is the same:

$$\Delta h = \eta_i \, \Delta h_s \tag{3.6}$$

An isentropic process occurs at constant entropy, whereas in the real process the entropy increases. This implies that the final enthalpy is higher than the isentropic enthalpy (see Figure 3.3). At the end of the real process this can be seen as a higher tem-

perature in the case of overheated steam or a higher steam quality in the case of wet steam.

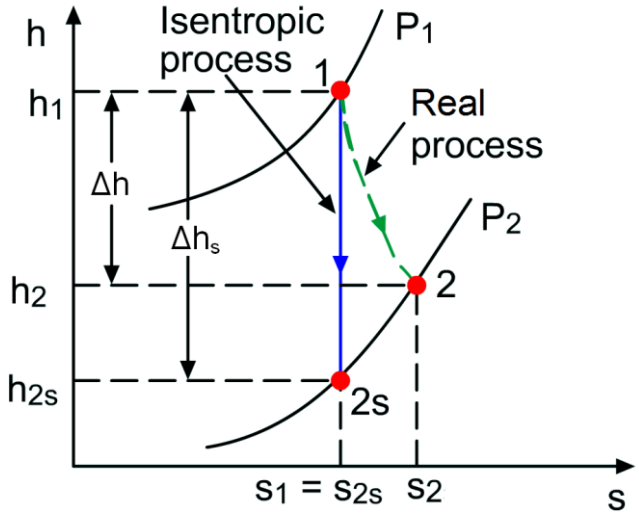

Figure 3.3 Comparison between the real process and the ideal process in the hs-Diagram

The isentropic efficiency can be calculated for every given turbine, once the pressure and the enthalpy at every reference point are known (e.g. see Figure 4. and Figure 4.3). The process would be analogue to the one described above.

Given the steam properties at the inlet of the turbine and knowing the isentropic efficiency of every stage, only the pressure at every point of the turbine is necessary to know all the thermodynamic properties at that point. With the entropy and the pressure drop, the isentropic enthalpy difference can be calculated. Multiplying the isentropic enthalpy difference by the isentropic efficiency, the real enthalpy drop can be calculated; and knowing the pressure and the enthalpy at a certain point, all the properties are known (see Figure 3.4).

So, for a given turbine stage and given η_i, p_a, h_a, T_a, x_a and p_b, the algorism above described would be:

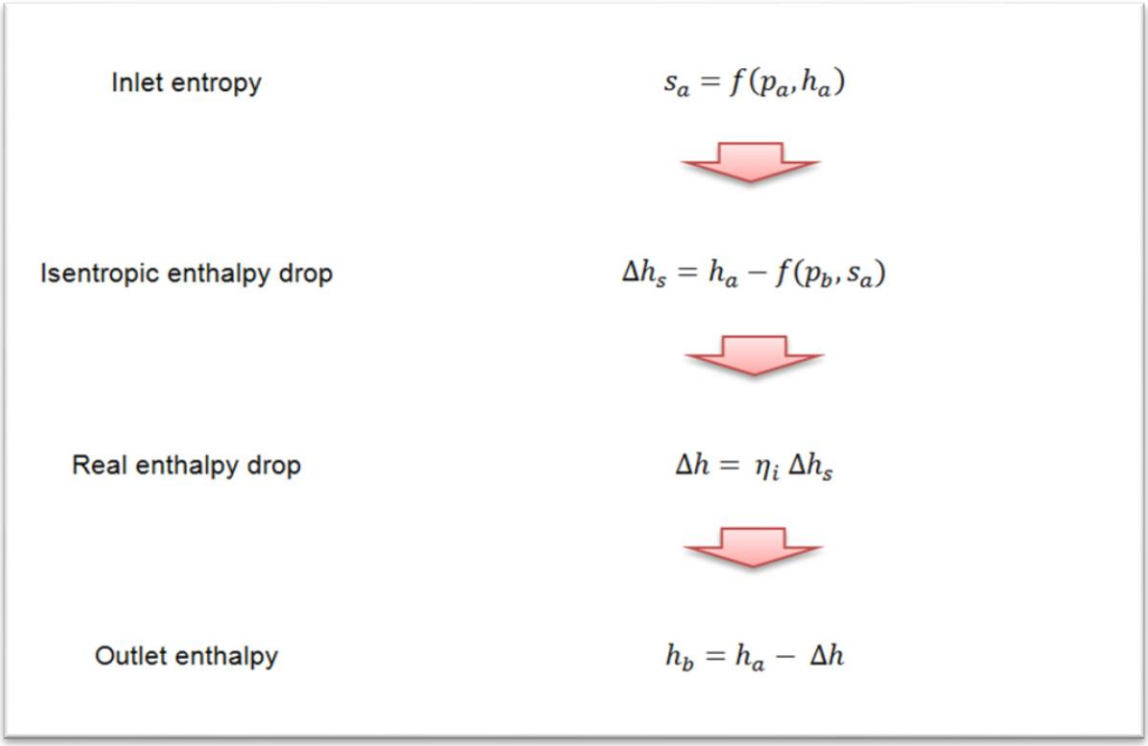

Figure 3.4 Calculation of the enthalpy at the exhaust of a turbine when inlet properties (sub index a), the exhaust pressure (sub index b) and the isentropic efficiency of the turbine are known.

As stated above, the internal efficiency can be calculated, provided that the rest of the parameters are given. However, when simulating off design operation, these parameters are not known. The internal efficiency is influenced by many design factors including blade construction and operation point and it reaches its maximum at nominal load. Equation (3.7) is a semi-empirical formula that describes the variations of the internal efficiency as a function of the angular velocity; the design efficiency and the isentropic enthalpy drop (Ray, 1980).

$$\eta_i = \eta_{i,0} - \alpha \left[\frac{N/\sqrt{\Delta h_s}}{N_0/\sqrt{\Delta h_{s,0}}} - 1 \right]^2 \tag{3.7}$$

Where α is a positive constant. For the purpose of this paper it can be considered that α=2 and $\eta_{i,0}$=0.87.

In (Grote, 2009) equation (3.8) is used; however he quotes (Ray, 1980).

$$\eta_i = \eta_{i,0} - \alpha \left[\frac{N}{N_0} \sqrt{\frac{\Delta h_s}{\Delta h_{s,0}}} - 1 \right]^2 \tag{3.8}$$

Although both equations behave similarly in the surroundings of the design point beyond a certain point, they give very different results. The simplest hypothesis is that there was a spelling mistake.

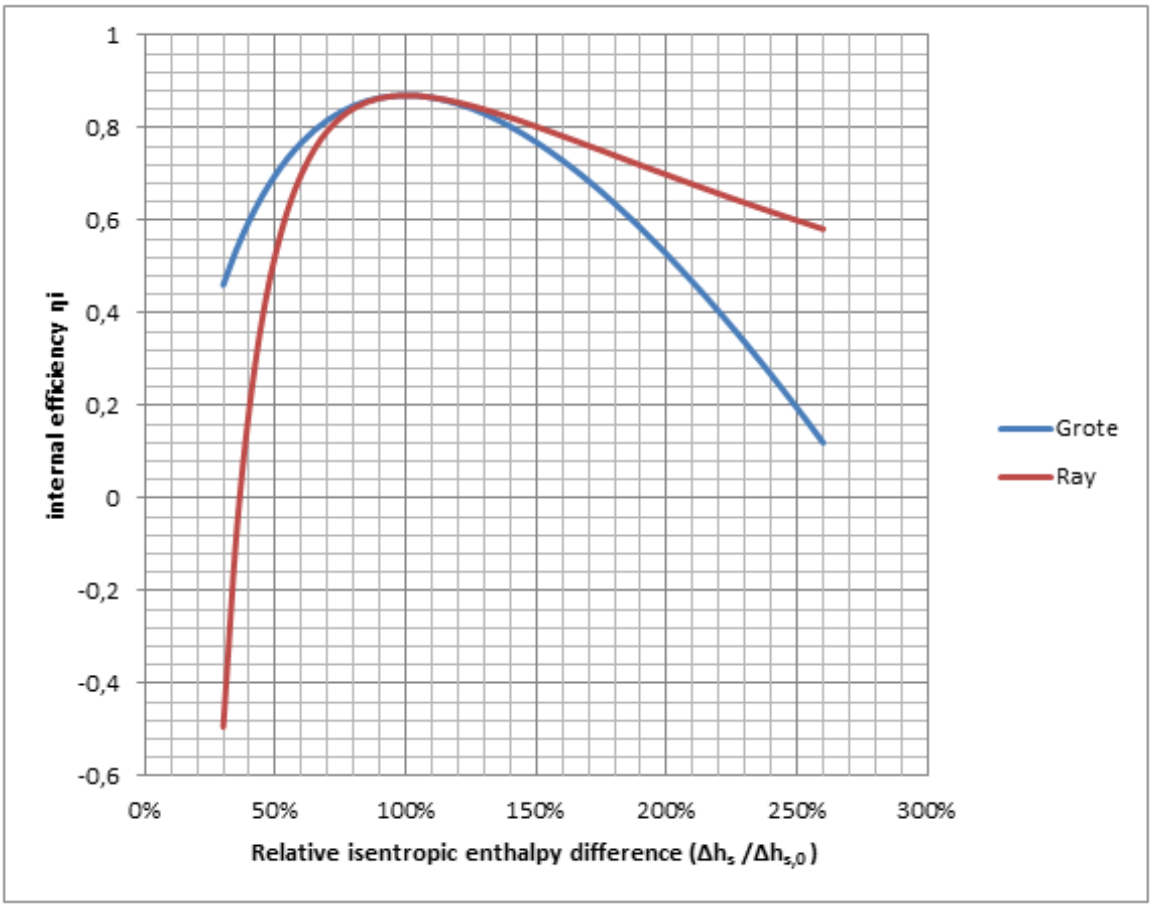

Figure 3.5 Evolution of the isentropic efficiency depending of the isentropic enthalpy difference for constant angular velocity.

In Figure 3.5 both equations plotted. The red line corresponds to equation (3.7) and the blue one to equation (3.8). Notice that if instead of plotting after (Δh_s /$\Delta h_{s,0}$) the plot is done after ($\Delta h_{s,0}$ / Δh_s), the plot resulting then is identical but corresponding the red line to equation (3.8) and the blue one to equation (3.7).

For the purpose of this paper, equation (3.7) will be used and the above stated hypothesis will be accepted.

3.3 Extractions

Extractions are of great importance to this paper (see section 4.2). It will be considered that there are two different kinds of extractions, the steam extraction and the water extraction. The extractions increase the efficiency of the cycle by preheating the feed water and extracting the condensed water of the turbine keeping the quality of the steam in it inside the margins thus avoiding erosion problems in the blades and minimizing the efficiency losses due to condensation. The extraction of water increases the quality of the remaining steam in the turbine, which increases the enthalpy. This steam with higher quality is partly extracted by the steam extraction and the rest of it enters the following stage.

4 Data base

An unexpected difficulty was the unavailability of reliable and abundant turbine data. There were a few heat balances of NPP available, but mostly only for the full power output configuration, so reference data was partly available, but no possibility to compare the results with real data.

Manufacturers are very reserved with their data. Considering the fact that the potential user of ATHLET is expected to have a very limited access to relevant data; it has been decided to develop a model which relies as much as possible on data obtainable by the user.

This lack of detailed data had a great influence in the development of this work making the first fluid dynamic approaches developed unpractical. All the geometry based models had to be abandoned as the geometry was completely unknown. Even if access to detailed geometry would have been granted, ATHLET is a 1-D code and a blade geometry based solution would have required a 3-D approach.

4.1 Reference plant

The reference data for this paper has mainly been the heat and mass balances of the Nuclear Power Plant Phillipsburg 2 with the old turbine. For this NPP we have data about more operation points that for any other, namely for 100%, 80%, 60% and 40% power output. The plant consists of one 2-flow HP turbine and 3 identical 2-flow LP turbines.

For comparison purposes the reference turbine in (Grote, 2009) has also been used. It is an industrial extraction turbine in a steel mill in Salzgitter used for the production of electricity and process steam, it was installed in 2006 by MAN Turbo and its generator has an electrical power output of 45-55 MW.

4.2 Available data on the reference plant

As stated before, only the heat balance plans are known for the reference Plant. The pressure, the enthalpy, the mass flow and the steam quality at some significant points are known (see Figure 4.).

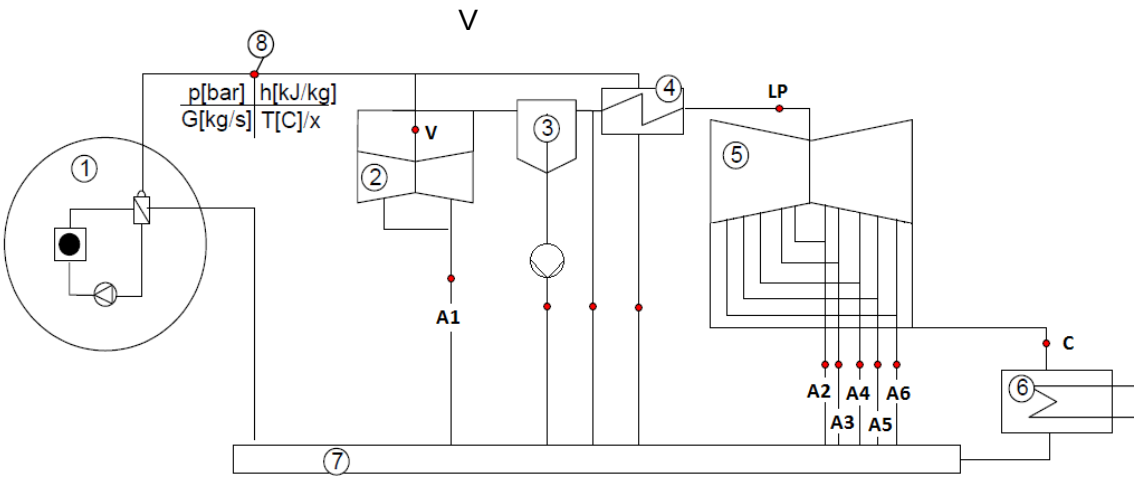

Figure 4.1 Heat balance plan of the reference plant being: (1) the primary loop; (2) HP turbine; (3) Moisture separator; (4) Re-heater; (5) LP turbine; (6) Condenser; (7) Feed water preheater lines (8) Points of which the thermodynamic values (pressure, enthalpy, mass flow and temperature/steam quality) are known.

The values at the marked points are known for the plant working at 100%, 80%, 60% and 40% of the nominal output.

This data is schematic and its accuracy is not certain. The reason for this is that the exact points where these values are measured are not known, and significant changes can be expected depending on where the measurements take place. At the HP turbine inlet (see Figure 4.1Figure 4., point V) there are values given, however these are not coherent with the measurements in the rest of the points whenever the plant works below nominal power.

In fact the pressure in all the steam extractions as well as in the inlet of the LP turbine is proportional to the power output; so for instance, the pressure in A1 (see Figure 4.)

when the plant is working at a 40% of its nominal power is approximately 40% of the pressure in A1 when the plant is working at nominal power. This is not the case for the pressure in point V at the HP-turbine inlet; in fact, the pressure at that point for the plant working at 40% of its capacity is almost 30% higher than the pressure with the plant working at nominal power (see Table 4.1).

Table 4.1 Pressures at the different load points divided by to the nominal pressures ($p_i/_{pi0}$).

Power	100,00%	80,00%	60,00%	40,00%
V	1	1,1102107	1,21880065	1,27714749
A1	1	0,78111588	0,57081545	0,35450644
LP	1	0,79174312	0,58990826	0,38440367
A2	1	0,78947368	0,5812357	0
A3	1	0,79487179	0,58974359	0,39487179
A4	1	0,7993921	0,59878419	0,40273556
A5	1	0,79619565	0,59782609	0,40217391
A6	1	0,80379747	0,60759494	0,41772152

The reason for this is because the available data have been taken in front of the Main Steam Valve (see Figure 4.2). So that the values are taken before the steam gets throttled through the valve and do not represent the state of the steam at the inlet of the HP-turbine. For operation points different than nominal power, the main steam valve is partially closed so that a critical flow takes place limiting the mass flow through the valve regardless of the pressure at the inlet of the valve.

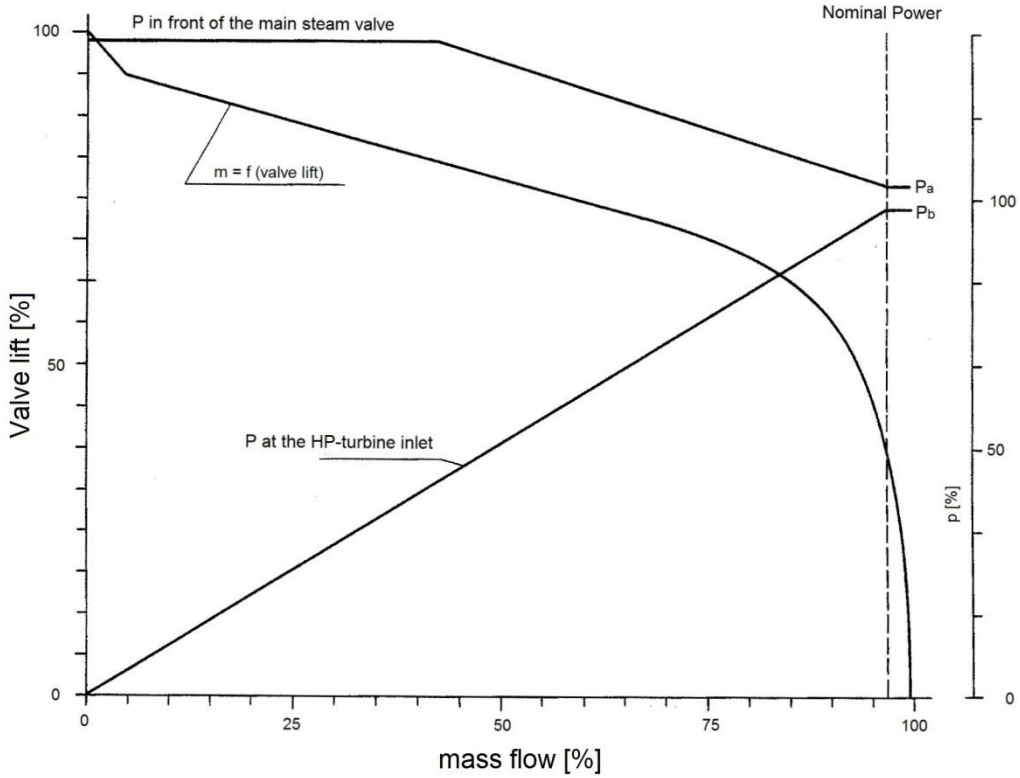

Figure 4.2 Qualitative representation of the Steam pressure and mass flow behaviour through the main steam valve (Siemens AG Bereich Energieerzeugung).

Another point in which the state of the steam is not given is at the HP-turbine outlet, before the moisture separator. However, since the properties and mass flow after the moisture separator are known for both steam and water, it can be calculated. This calculation is based on the hypothesis that there is no pressure drop in the moisture separator.

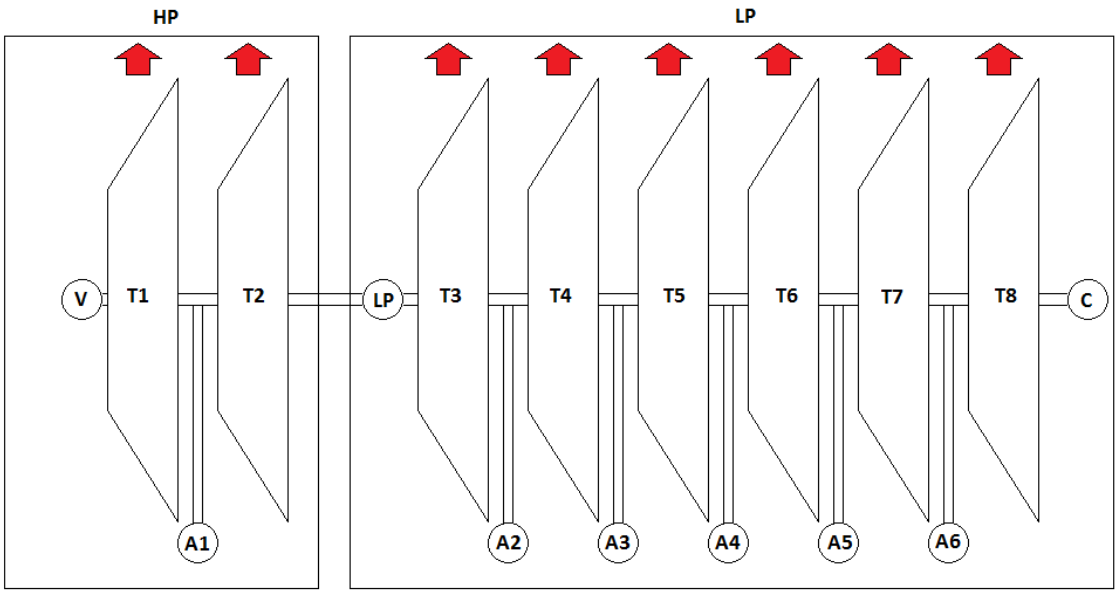

Figure 4.3 Schematic of the turbine. The circles are points in which the mass flow, the enthalpy, the pressure and the steam quality or temperature are known. T1-8 are the indexes for every stage.

The turbine has several stages and several extractions, however the exact number of stages is not known. In order to simplify the explanation of the calculations, every turbine section between two extractions will be called a stage, so for the purpose of this work, the HP turbine will have two stages (T1 and T2, see Figure 4.3) and the LP turbine will have 6 stages (T3 to T8).

Given the numerous extractions, the mass flow through the turbine is not constant, so that the flow through every stage is constant throughout that particular stage but different from the flow through the other stages.

The mass flow through every stage is:

$$\dot{m}_{T1} = \dot{m}_V \tag{4.1}$$

$$\dot{m}_{T2} = \dot{m}_{T1} - \dot{m}_{A1} \tag{4.2}$$

$$\dot{m}_{T3} = \dot{m}_{LP} \tag{4.3}$$

$$\dot{m}_{T4} = \dot{m}_{T3} - \dot{m}_{A2} \tag{4.4}$$

$$\dot{m}_{T5} = \dot{m}_{T4} - \dot{m}_{A3} \tag{4.5}$$

$$\dot{m}_{T6} = \dot{m}_{T5} - \dot{m}_{A4} \tag{4.6}$$

$$\dot{m}_{T7} = \dot{m}_{T6} - \dot{m}_{A5} \tag{4.7}$$

$$\dot{m}_{T8} = \dot{m}_{T7} - \dot{m}_{A6} = \dot{m}_{C} \tag{4.8}$$

At extraction A5 (see Figure 4.3) only water is extracted, so that the steam properties at that point are not known. However the water mass extracted at A5 ($\dot{m}_{A5}$) is about 1.5% of the mass flow through the next stage ($\dot{m}_{T7}$) so that it could be neglected and stages T6 and T7 considered as one single stage. Another approach is to interpolate the steam quality at the point A5, knowing that the pressure of the steam is the same as the pressure of the extracted water. With the interpolated steam quality after the extraction, all the properties of the steam are known. The latter has been the one chosen in the frame of this paper.

Once the enthalpy drop and the mass flow through every stage are known, the power output of every stage can be calculated. However, once this calculation is done, the resulting global power (see equation (4.9)) is below the nominal power of the generator. The way of considering the extraction A5 and the stage T6 has little influence on this result, the difference between both approaches being minimal.

$$P = \sum_{i=T1}^{T8} \dot{m}_i \, (h_i^{in} - h_i^{out}) \tag{4.9}$$

The reason for this is because the improvement of the specific enthalpy due to the water extractions has not been taken into account. As stated before, the water extractions at every stage result in an increase of the specific enthalpy, this means that the steam entering a given stage has a higher enthalpy that the steam that leaves the preceding stage.

For the plant working at full load, water is extracted at all the extractions with the exception made of extraction A2, where only overheated steam is extracted.

The exact quality improvement at every stage is not known so that a few assumptions have to be made. The most important assumption made is that the enthalpy of the steam extracted is equal to that of the steam that goes through the next stage. So, to all the practical effects, the steam extracted has identical properties as the steam in the turbine. This assumption implies another, namely that the water extraction takes place before the steam extraction.

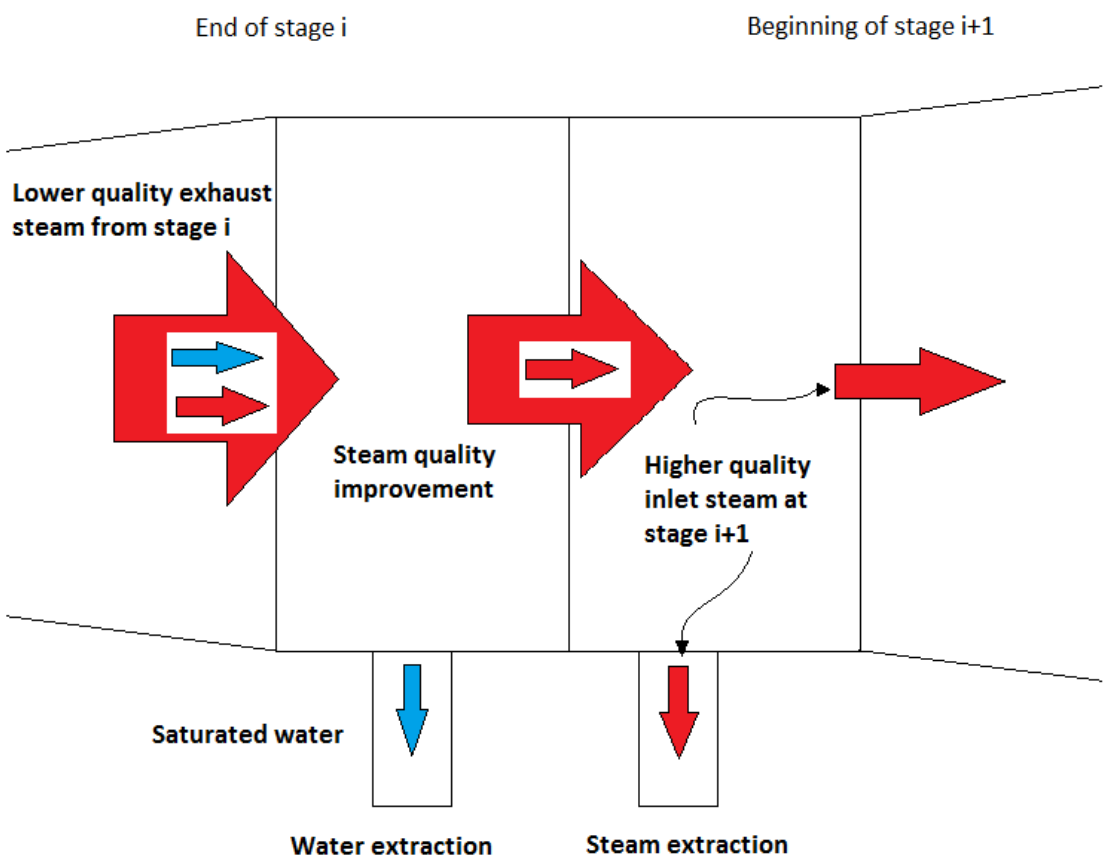

Figure 4.4 Schematic water and steam extraction between two stages.

A graphic representation of that assumption is shown in Figure 4.4. The steam coming out of the stage i gets the water extracted thus increasing its quality. After this quality improvement and before entering the next stage, a part of the “good quality” steam is extracted, the rest of the steam enters stage i+1.

Taking this into account, the specific enthalpy after every stage has to be recalculated. At every extraction a known quantities of steam and water are extracted. The pressure, the enthalpy, and the mass flow of both steam and water are known. The mass flow through stage i (the stage before the extractions) is also known. So to calculate the specific enthalpy of the steam coming out of stage T1 (in the HP turbine):

$$h_{out} = \frac{\dot{m}_{water}\, h_{water} + (\dot{m}_{T1} - \dot{m}_{water}\,)\, h_{steam}}{\dot{m}_{T1}} \tag{4.10}$$

Being:

h_{out} the enthalpy at the outlet of stage T1.

$\dot{m}_{T1}$ the mass flow through T1 (stage before the extraction).

$\dot{m}_{water}$ the mass flow of water extracted at A1

h_{water} the enthalpy of the water extracted at A1

h_{steam} the enthalpy of the steam extracted at A1

After recalculating the enthalpy after every stage, the power output is slightly above the generator power. The resulting efficiency of the turbine and the generator together is of 98%. This efficiency is very high but not much higher than the expected efficiency (95 to 97%). Although it is considered acceptable, there are some factors which could explain this high efficiency.

The main reason is the assumption that the quality of the steam extracted is identical to the quality of the steam remaining in the turbine, so that only the water extraction itself is responsible for the steam quality improvement. This assumption does not take into consideration the centrifugal forces in the turbine. Due to these forces, it is expected that the steam extracted has a lower quality than the steam remaining in the turbine. This fact means that the steam quality improvement at every stage is greater than calculated.

The internal efficiency of the turbine stages calculated after these assumptions varies greatly depending on the stage. For stage T5 it almost reaches a value of 1, which is impossible. This is because the enthalpy after stage T5 is actually higher than the enthalpy calculated supposing an identical enthalpy of the steam extracted and the steam in the turbine. At the last stage, the internal efficiency is 0,4, the reason for this is that the steam coming out of the last stage (as well as the steam along the turbine) has a high velocity and therefore a high kinetic energy and as it enters the condenser it loses its velocity which then results in an enthalpy increase (see equation (2.7)). Despite of this, it has been decided to maintain the steam enthalpy at the last stage equal to the enthalpy of the steam in the condenser.

The fact that the results are within the margins, and that there is no way to determine the real quality improvement, are considered sound arguments in favour of maintaining all the aforenamed assumptions and considerations in this paper.

An interesting fact is that the total power output of the turbine remains constant no matter what steam quality value is used in the extraction A5, as a higher steam quality brings an increase of the power output at the stage T7 but this increase is compensated by the decrease of the power output in the stage T6 and vice versa.

This seems to be true for all the extractions, as what are important for the power calculation are the total mass extraction and the quality improvement resulting of the extraction.

In (Grote, 2009) a large number of geometrical data of the reference turbine was available, such as Volumes of the spaces between stages, exact enthalpies and pressure at the inlet and at the outlet of every stage and isentropic efficiencies of every stage at one of the design points, however, the information for other design points was limited (Grote, 2009) and it could not be used as reference data to be compared with the results provided by the implemented models.

5 ATHLET

The turbine model is to be developed in ATHLET, so a general description of ATHLET is necessary in order to justify the solutions chosen. For those areas necessary to understand the development of the model, a detailed description is provided.

For further detail see (GRS, 2009).

5.1 Description of ATHLET

The thermal-hydraulic computer code ATHLET (Analysis of Thermal-Hydraulics of LEaks and Transients), developed by the Gesellschaft für Anlagen- und Reaktorsicherheit (GRS), aims to cover the whole spectrum of design basis and beyond design basis accidents (without core degradation) for light water reactors such as PWR, BWR, VVER and RBMK.

5.1.1 Modules of ATHLET

ATHLET is composed of several basic modules which simulate the phenomena involved in the operation of LWR. These basic modules are:

Thermo-fluid dynamics: This module is based upon a 5-equation model with a mixture momentum equation, and separate conservation equations for vapour and liquid, or a 2-fluid model with 6-equations which has a momentum equation for vapour and another for liquid.

Heat Transfer and Heat Conduction: This module allows the simulation of the heat conduction in all those components needed.

Neutron Kinetics: Models the nuclear heat generation.

General Control Simulation Module: Is a block-oriented simulation language for the description of control, protection and auxiliary systems. GCSM allows the representa-

tion of fluid dynamic systems in a very simplified way requiring very little computation time to do so. So far, the turbine has been modelled by a large number of GCSM signals.

The solution of the differential equation system is performed implicitly by the ODE-solver FEBE. The coupling of other independent modules can easily be performed in the general interface.

Although major plant components can be modelled by connecting TFOs and HCO via input data, some of them are available as special objects. These special objects are simplified and compact models.

Additional models for the simulation of valves, pumps, accumulators, steam separators, single ended breaks, double ended breaks, fills, leaks and boundary conditions for pressure and enthalpy, are provided. The purpose of this paper is to add a turbine to this list.

5.1.2 The Thermo-Fluid dynamic Module

The leading module in ATHLET is the thermo-fluid-dynamic (TFD) module. Given that the turbine model is to be a thermodynamic model, the TFD module has to be explained comprehensibly in order to fully understand the proposed solution.

The basic equations describing the thermal-hydraulic behaviour of the system are based on the conservation laws of mass, energy and momentum. They are time and space dependant partial differential equations which have to be solved numerically, as it is not possible to solve them analytically.

The system configuration to be simulated is modelled connecting basic thermo-fluid dynamic objects (TFO) and heat conduction objects (HCO) via input data. There are different TFOs categories; however only the pipe objects are relevant for this paper.

Pipe objects apply for a one-dimensional TFD-Model with partial differential equations describing the transport of fluid. In the input data the nodalization is defined. Beyond that point a pipe object is treated as consecutive Control Volumes united to each other

by junctions. The control domain of every junction is defined by the CVs centres at its right and left (see Figure 5.1). The momentum differential equations provide the mass flow rates at the boundaries of each CV. A single junction pipe consists of a single junction without any control volumes.

The mass and energy based partial differential equations are integrated using the CVs as integration domain, the pressure, the vapour and the liquid temperatures, and the steam quality being the solution variables. For the momentum based partial differential equations the integration domain is the junction (staggered grid), the mass flow rate being the solution in the 5-equation model and the phase mass velocities in the 6-equation model. This is known as a staggered grid (see Figure 5.1). The quantities resulting from these integrations represent the local average physical state and are only time dependant.

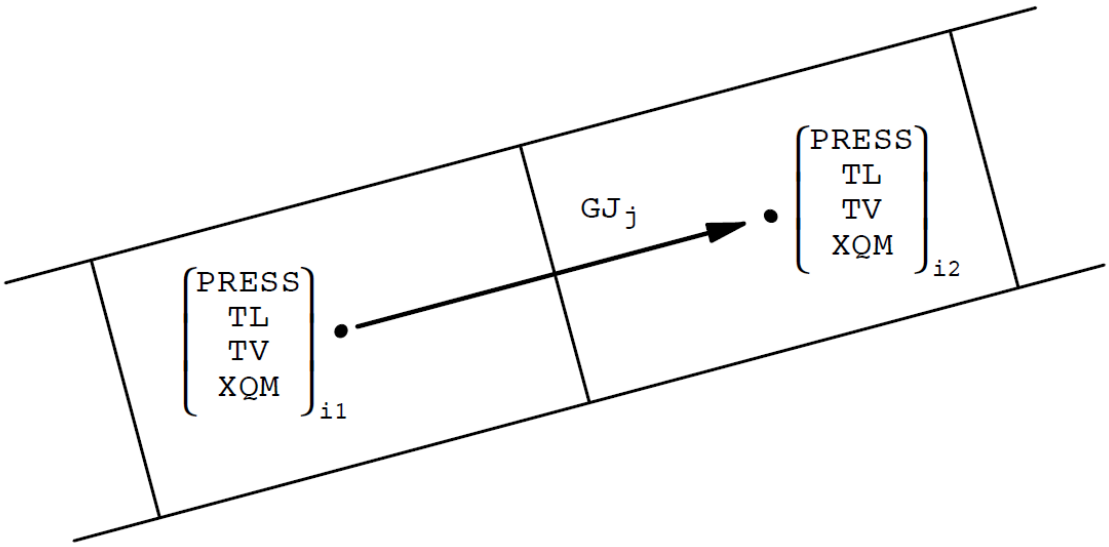

Figure 5.1 Staggered grid with CV and junctions (GRS, 2009)

For the 5-equation model the mass and energy balances for vapour and liquid in the CV are solved separately. The solution variables in the CV are: The mass quality, the liquid temperature, the vapour temperature and the pressure.

Integrating the mass conservation equations over a CV V_i, liquid mass balance equation (5.1) and vapour mass balance equation (5.2) are obtained.

$$\frac{dM_L}{dt} = \sum_{in} G_L - \sum_{out} G_L - \psi \tag{5.1}$$

$$\frac{dM_V}{dt} = \sum_{in} G_V - \sum_{out} G_V + \psi \tag{5.2}$$

With:

$$G_L = (1-\alpha)\rho_L w_L A \tag{5.3}$$

$$G_V = \alpha \rho_V w_V A \tag{5.4}$$

With:

$$\alpha = \frac{V_V}{V} = \frac{M_V v_V}{M_V v_V + M_L v_L} \tag{5.5}$$

From the phase mass balances above the differential equation for the mass quality is derived:

$$\frac{dx_M}{dt} = \frac{M_L \frac{dM_V}{dt} - M_V \frac{dM_L}{dt}}{(M_L + M_V)^2} \tag{5.6}$$

With

$$M_L = \frac{1 - x_M}{x_M v_V + (1 - x_M) v_L} V \tag{5.7}$$

$$M_L = \frac{x_M}{x_M v_V + (1 - x_M) v_L} V \tag{5.8}$$

Integrating the energy balance equations over the CV V_i and after making some simplifications, the ordinary differential equations for the phase temperatures (5.9) and (5.10) are obtained.

$$\frac{dT_L}{dt} = \frac{1}{c_{p,L}}\frac{E_L}{M_L} + \frac{1}{c_{p,L}}\left(v_L - \left.\frac{\partial h_L}{\partial p}\right|_{T_L}\right)\frac{dp}{dt} \tag{5.9}$$

$$\frac{dT_V}{dt} = \frac{1}{c_{p,V}}\frac{E_V}{M_V} + \frac{1}{c_{p,V}}\left(v_V - \left.\frac{\partial h_V}{\partial p}\right|_{T_V}\right)\frac{dp}{dt} \tag{5.10}$$

Where

$$\begin{aligned} E_L = \sum_{in} G_L\left(h_L + \frac{w_L^2}{2}\right) - \sum_{out} G_L\left(h_L + \frac{w_L^2}{2}\right) - \frac{dM_L}{dt}\left(h_L + \frac{w_L^2}{2}\right) \\ - M_L w_i \frac{dw_i}{dt} - QEI + QI_L \end{aligned} \tag{5.11}$$

$$\begin{aligned} E_V = \sum_{in} G_V\left(h_V + \frac{w_V^2}{2}\right) - \sum_{out} G_V\left(h_V + \frac{w_V^2}{2}\right) - \frac{dM_V}{dt}\left(h_V + \frac{w_V^2}{2}\right) \\ - M_V w_i \frac{dw_i}{dt} + QEI + QI_V \end{aligned} \tag{5.12}$$

QEI is the interfacial heat exchange due to condensation or evaporation, QI the heat source to the control volume and w_i the average fluid velocity in the CV.

The differential equation for the pressure is:

$$\frac{dp}{dt} = -\frac{Z_1}{Z_2} \tag{5.13}$$

With:

$$Z_1 = \frac{dM_V}{dt} v_V + \left.\frac{\partial v_V}{\partial T_V}\right|_p \frac{E_V}{c_{p,V}} + \frac{dM_L}{dt} v_L + \left.\frac{\partial v_L}{\partial T_L}\right|_p \frac{E_L}{c_{p,L}}$$

And

$$Z_2 = M_V \left[\left.\frac{\partial v_V}{\partial p}\right|_{T_V} + \frac{1}{c_{p,V}} \left.\frac{\partial v_V}{\partial T_V}\right|_p \left(v_V - \left.\frac{\partial h_V}{\partial p}\right|_{T_V} \right) \right] + M_L \left[\left.\frac{\partial v_L}{\partial p}\right|_{T_L} + \frac{1}{c_{p,L}} \left.\frac{\partial v_L}{\partial T_L}\right|_p \left(v_L - \left.\frac{\partial h_L}{\partial p}\right|_{T_L} \right) \right]$$

In the junction a mixture momentum balance is solved. (5.14) is the differential equation for the mixture flow rate over a junction j connecting CVs i_1 and i_2.

$$\frac{dG}{dt} = \frac{1}{\int \frac{1}{A} ds} \left[\Delta p_s + \Delta p_{MF} + \Delta p_{WR} + \Delta p_{grav} + \Delta p_{fric} + \Delta p_\rho + \Delta p_I \right] \tag{5.14}$$

$$\Delta p_s = -\int \frac{\partial p}{\partial s} ds \; = \; p(i_2) - p(i_1) \tag{5.15}$$

Δp_I is the pressure difference between the CVs at both sides of the junction

Δp_{MF} is the momentum flux term

Δp_{WR} is the relative velocity term

Δp_{grav} is the elevation term

Δp_{fric} is the friction and loss pressure drop

Δp_ρ is the density derivative term

Δp_I is the external source term, e.g. pump differential pressure term

From all these terms and in the context of this paper, only the external source term, the friction and loss pressure drop term, and the momentum flux term need a further analysis.

The external source term and its influence and importance in the solution chosen will be explained in chapter 6.2.

The friction and loss pressure drop term is:

$$\Delta p_{fric} = \int f_{W,m} \, ds \tag{5.16}$$

The momentum flux is calculated as follows:

$$\Delta p_{MF} = -\int \rho_m \, \vec{w}_m \, \frac{\partial \, \vec{w}_m}{\partial s} \, ds = 0.5 \, (T_1 - T_2) G^2 \tag{5.17}$$

Where:

$$T_1 = \frac{1}{\rho_m(i_1) \, A^2(i_1)} - \frac{1}{\rho_m(i_2) \, A^2(i_2)}$$

And

$$T_2 = \frac{T_1}{3}, if \quad |\rho_m(i_2) \, A^2(i_2) - \rho_m(i_1) \, A^2(i_1)| \leq 10^{-4} \left(\rho_m(i_1) \, A^2(i_2) + \rho_m(i_2) \, A^2(i_1)\right)$$

$$T_2 = \frac{\Delta\rho_m}{\left(\Delta(\rho_m \, A)\right)^2} \left[\frac{(\Delta\rho_m)^2}{\rho_m(i_1)\rho_m(i_2)} + \frac{(\Delta A)^2}{A(i_1)A(i_2)} + \frac{2T_c \Delta\rho_m \Delta A}{\Delta(\rho_m \, A)}\right], \qquad else$$

With

$$\Delta\rho_m = \rho_m(i_2) - \rho_m(i_1)$$

$$\Delta A = A(i_2) - A(i_1)$$

$$\Delta(\rho_m A) = \rho_m(i_1)A(i_2) - \rho_m(i_2)A(i_1)$$

$$T_c = ln\left[\frac{\rho_m(i_1)A(i_2)}{\rho_m(i_2)A(i_1)}\right]$$

Two models relevant for this paper are FILLs and Pressure-Enthalpy Boundary Component.

Fills are junction related models used for the simulation of mass sources and sinks. If it is to be a mass source, the mass flow to be injected in the system has to be defined in the GCSM module as well as the total specific enthalpy. If a sink is to be simulated, the mass flow has to have a negative sign and the enthalpy does not need to be defined as it is calculated as in normal junctions, i.e. they are calculated from the upstream conditions.

Pressure-Enthalpy Boundary Component - also referred to as 'time dependent volume (TDV)' – is a CV related model which permits to establish, via GCSM signals, a pressure enthalpy boundary at the edge of the system. In this way and depending upon the conditions in the system, mass will flow into the TDV or from the TDV into the system.

6 Implementation of the turbine model

The implementation of the turbine model has been chosen taking into consideration the characteristics of ATHLET in order to make it coherent with the solving strategy of ATHLET and as simple as possible.

Various approaches for the simulation of a turbine are proposed in literature and outlined in section 6.1. Taking into account the characteristics of the ATHLET solution strategy a new method had to be developed that is based on a finite volume discretization and a staggered grid method with solution variables defined in control volumes and junctions. The method, as well as the details of the implementation, is thoroughly discussed in section 6.2.

6.1 Alternative implementation strategies

An approach suggested by some authors (Plavšić, 2008) is to develop a model taking a detailed geometry configuration such as turbine blade angles into account. In order to develop such a model, detailed information about blade geometries and fluid velocities in the turbine are needed, this information however is only in the power of turbine manufacturers and the access to it is highly limited as it is regarded as industrial secret. Besides, the solution of such a model would need to be performed in 3-D and ATHLET is a 1-D code.

Several authors have proposed different modelling strategies for simulating steam turbines (Grote, 2009), (Zimmer, 2008). The thermodynamic basis and the physical models are the ones exposed in chapter 3.

The thermodynamic systems are designed as storage-throttle-systems where the internal volume of the turbine is divided into several steam storages (internal volumes of pipes and between turbine stages) which are linked together by throttle devices (turbines and valves) governed by valve or turbine models, depending on the case (see Figure 6.1).

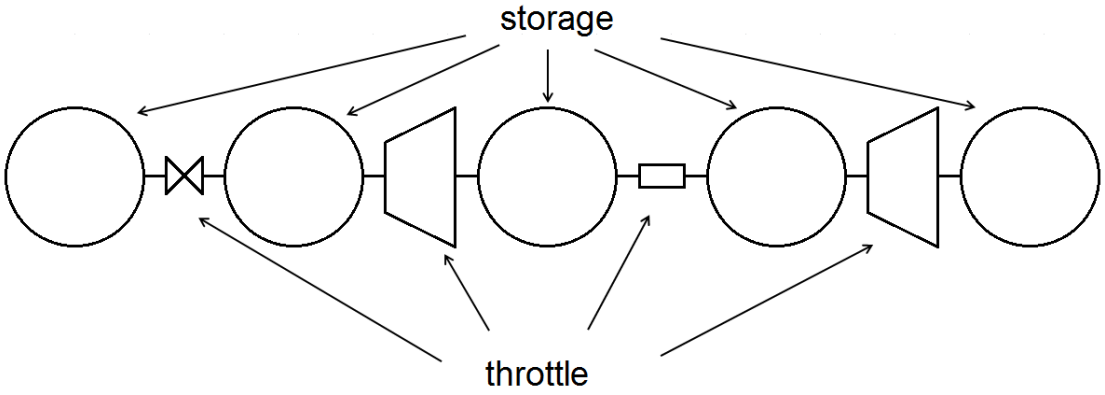

Figure 6.1 Turbine as a storage-throttle system

For the storages the following equations for mass and energy conservation apply:

$$m(t) = m_0 + \int_0^t \left(\dot{m}_{in}(\tau) - \dot{m}_{out}(\tau)\right) d\tau \tag{6.1}$$

$$U(t) = U_0 + \int_0^t \left(\dot{Q}(\tau) + \dot{m}_{in}(\tau)\, \dot{h}_{in}(\tau) - \dot{m}_{out}(\tau)\, \dot{h}_{out}(\tau)\right) d\tau \tag{6.2}$$

Knowing the volume of all the storages, the specific volume and internal energy of the steam in any given storage are known, thus, with the help of the water steam properties, all the thermodynamic properties are known.

Together with equation (3.4) for the mass flow and with the algorism displayed in Figure 3.4 for the energy extraction in the turbine throttle, the whole turbine can be modelled. Other types of throttles such as valves and pipes are also modelled; however these models are not relevant for the purpose of understanding the general idea behind this strategy. The boundary conditions are the pressure at the inlet and at the outlet of the system e.g. pressure at the steam generator and condenser. A detailed description of this approach can be found in (Grote, 2009) and (Zimmer, 2008).

An attempt to develop a model according to the strategy mentioned was made, but, once the basic TFOs were coupled, several stability problems arose.

The configuration was of two TDV (with the pressure and enthalpy before and after the turbine) connected to each other by a pipe object representing the turbine. In the middle of the pipe object a modified junction was to provide the mass flow depending on the pressures. Even fixing the pressure and enthalpy drop in this primitive turbine junction via GCSM signals, the simulation proved to be unstable. This, together with the modelling difficulty resulted in abandoning of this strategy.

6.2 Chosen modelling strategy

As described chapter 5, ATHLET equation system consists of balance equations solved in control volumes and junctions. In ATHLET, several so-called junction-based component models already exist (e.g. pump…). These models basically provide additional source terms, that are added to the right hand side of the presented equations (see subsection 5.1.2). The idea is to model a turbine stage by a pressure drop across a junction and by a power extraction from the adjacent control volume. Adding the terms to the corresponding equations, a turbine stage could be simulated by a junction of a basic thermo-fluid pipe object. Corresponding models are presented in the following chapters. The extraction lines have to be modelled separately.

The boundary conditions of the chosen strategy are: fill providing mass flow and enthalpy in the main steam line, and a TDV (p-h-boundary) for the pressure in the condenser.

6.2.1 Pressure drop model

Given the limited data available (see chapter 4) and the characteristics of ATHLET, the chosen strategy has been to develop the turbine model taking the pump model as a basis.

For the turbine model a new junction type which adds a negative pressure in the term Δp_l of equation (5.14) has been developed. To do that, the mass flow has been considered an input data, it being the steam mass flow from the steam generator, considering valid the hypothesis exposed in chapter 4.

So, defining Δp_I as:

$$\Delta p_I = p_a - p_b \tag{6.3}$$

And isolating p_b in equation (3.4):

$$p_b = p_a \sqrt{1 - \left(\frac{\dot{m}\, p_{a0}\sqrt{T_a}}{\dot{m}_0\, p_a\sqrt{T_{a0}}}\right)^2 \left(1 - \left(\frac{p_{b0}}{p_{a0}}\right)^2\right)} \tag{6.4}$$

So, in the turbine junction, between CVs i_1 and i_2 the term Δp_I in the differential equation (5.14) is:

$$\Delta p_I = p_a - p_a \sqrt{1 - \left(\frac{\dot{m}\, p_{a0}\sqrt{T_a}}{\dot{m}_0\, p_a\sqrt{T_{a0}}}\right)^2 \left(1 - \left(\frac{p_{b0}}{p_{a0}}\right)^2\right)} \tag{6.5}$$

The terms p_a and p_b are the pressure terms in CVs i_1 and i_2 respectively (see Figure 5.1).

6.2.2 Power extraction model

The enthalpy drop in the turbine has been modelled as an energy extraction over the junction. However the entropy in a given control volume is not one of the solution variables and could not be obtained, so the algorism of Figure 2.3 could not be applied. Thus an alternative way to calculate the energy extraction has been chosen:

The second law of thermodynamics can be formulated as follows:

$$dS \geq \frac{dQ}{T} \tag{6.6}$$

Adding an ideal heat quantity dQ’ to equation (6.6):

$$dS = \frac{dQ + dQ'}{T} \tag{6.7}$$

Equation (6.7) is explained by exchanging the process in equation (6.6) by a reversible process which departs from the same starting point and ending point. In that case and dQ being the heat added in the irreversible process, a reversible heat quantity dQ' has to be added.

In the case of a closed volume in which a fluid changes its volume exchanging heat with the environment and under a given pressure, the first law of thermodynamics can be formulated as follows:

$$\mathrm{dU} = -\mathrm{p\ dV} + \mathrm{dA_\eta} + \mathrm{dQ} \tag{6.8}$$

The term pdV is the work done by the pressure on the surface of the element, the term dA_η is the dissipated work (due to friction, deformation, etc.) and dQ the heat added. Given the fact that work gets dissipated, the process is irreversible. If the process is conducted with infinite slowness, no work is dissipated, so this new process becomes reversible. If this substitute reversible process is to achieve the same state modification as the irreversible one (dU has to be the same for both processes), and the dissipation work no longer being present, an additional heat quantity dQ' (equal to dA_η of the real process) has to be added. So, equation (6.8) can be rewritten as:

$$dU + p\ dV = dQ' + dQ \tag{6.9}$$

With equation (6.7) and dividing all the extensive units by the mass (U, S and V):

$$\mathrm{ds} = \frac{\mathrm{du} + \mathrm{p\ dv}}{\mathrm{T}} \tag{6.10}$$

The definition of enthalpy is:

$$\mathrm{h} \equiv \mathrm{u} + \mathrm{pV} \tag{6.11}$$

Deriving equation (6.11):

$$\mathrm{dh} = \mathrm{du} + \mathrm{p\,dv} + \mathrm{v\,dp} \tag{6.12}$$

$$\mathrm{du} + \mathrm{p\,dv} = \mathrm{dh} - \mathrm{dp} \tag{6.13}$$

With equation (6.10)

$$\mathrm{ds} = \frac{\mathrm{dh} - \mathrm{v\,dp}}{\mathrm{T}} \tag{6.14}$$

Assuming an isentropic process, i.e. ds=0:

$$\mathrm{dh}_s = \mathrm{v\,dp} \tag{6.15}$$

Equivalent to:

$$\mathrm{dh}_s = \frac{1}{\rho}\,\mathrm{dp} \tag{6.16}$$

Assuming constant density before and after the turbine stage (Ray, 1980), and integrating equation (6.16), one obtains:

$$\Delta\mathrm{h}_s = \frac{1}{\rho}\,\Delta\mathrm{p} \tag{6.17}$$

Taking for ρ the average of density before and after the stage and Δp being the pressure drop of the turbine Δp_I (equation (6.5) difference between CVs i_1 and i_2.

With equation (3.6) and assuming that η_i is known, the heat extracted by the turbine junction to the fluid can be approximated by the equation (6.18):

$$\dot{\mathrm{Q}} = \eta_i\,\dot{\mathrm{m}}\,\frac{1}{\rho}\,\Delta p_I \tag{6.18}$$

This approach relies on the use of an average density. While this can be accepted in the case of incompressible fluids where the density variation can be neglected, the density variation throughout a turbine stage is considerable[2]. This density variation makes the assumption implied in equation (6.17) quite bold. However, the accuracy of the achieved results endorses the applicability of the proposed approach.

6.3 Implementation in ATHLET

Once the equations have been developed, the next step has been the implementation in ATHLET of the proposed model. Before starting with the implementation in ATHLET, all the equations have been tried in MATLAB in order to observe the response of the model. Although the results are not free of errors, the decision has been taken to carry on the implementation in ATHLET and to make any further modifications there.

The TFD system chosen is a fill junction connected to a pipe which is connected to a time dependent volume (see Figure 6.2).

As it is a first approach a pipe with a constant diameter is user in order to minimize any influence beside than the one of the turbine junction.

[2] For example in the case of the first stage in the LP turbine $\rho_a > 2\rho_b$

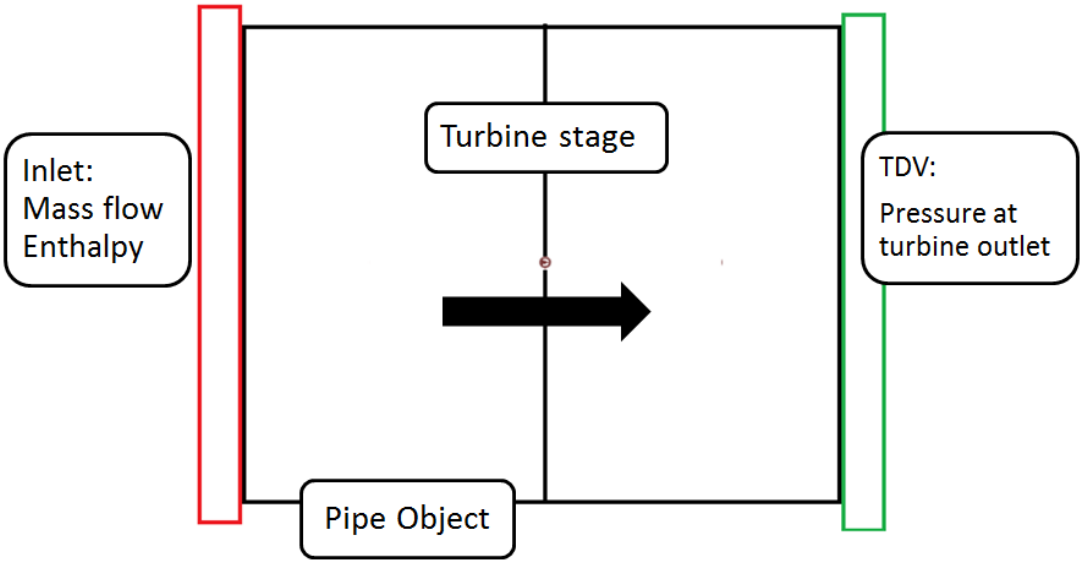

Figure 6.2 TFD system. Fill on the left, pipe in the middle and TDV on the right. The turbine junction is in the middle of the pipe.

In order to observe the behaviour of the turbine junction and of the whole TFD, the first simulation has been made by setting in the turbine junction, the pressure and enthalpy drop. After observing the adequacy of the configuration, further modifications have been done in order to implement the model.

6.3.1 The pump model as a basis

The turbine model has been developed taking the pump model as a basis. What the pump junction does is to introduce a pressure difference as part of the momentum equation of the junction (see equation (5.14)) and adding a pump power to the fluid in both adjacent CVs (see equations (5.11) and (5.12)).

The development of the turbine model taking the pump model as a basis is much simpler and takes much less modifications in the code than choosing the approach suggested in 6.1. That approach, besides of the initial instability problems explained, would have taken major modifications in the code including the development of entirely new subroutines.

All what is required is to modify the power extraction to the fluid and the way the pressure drop is inserted by introducing the equations (6.5) and (6.18) in the corresponding subroutines.

6.3.1.1 The modifications in the pump model

In the pump junction the power of the pump is added at both adjacent CV. In the turbine junction, instead of adding power, it is subtracted thus only a sign modification is necessary to represent the work performed by the steam in the shaft.

In a steam turbine the power transfer from the steam to the shaft takes place at that stage; however the present model does not model that stage internally. Instead it models a new junction type which is then accommodated in a pipe object. The pressure drop is added as part of the momentum equation of the junction; however the power extraction from the fluid cannot take place in the junction as the energy balance equation is not solved in the junction (see chapter 5). The power must therefore be extracted from the fluid in the CV after the turbine. The difference with the energy added in the pump model is that in the latter the pump energy is added to the CV before the pump and to the CV after the pump; in the turbine model all the energy is extracted from one single CV after the turbine junction.

In the turbine the work is done only by the steam; water in the turbine has a lower velocity than steam and actually receives energy from the turbine. Therefore, instead of extracting power from the liquid and steam phase, the energy extraction takes place only in the steam phase.

The modifications done in order to achieve this can be seen in Figure 6.3. The energy extraction done by the pump in the CV left of the junction (index JILJ) as well as the energy extraction to the liquid (QLI) in both CV have been turned off. Instead all the power extraction takes place in the CV right of the turbine junction and only to the steam.

```
93   C         STEADY STATE CALCULATION
94             ITES = 0 ! Unklar ob wir das brauchen
95             CALL KTUSS(IPUMP,ITYP,J,IWRIT,ROPU,PH1,QNP,GPU,Z(JU),QFLUID)
96   C
97         ELSE
98   C
99   C         TRANSIENT CALCULATION
100            UMN   = Z (JU) ! Pump-Speed
101            DUDT  = DZ(JU) ! Derivative of Pump Speed
102
103               PHE    = Z(JP+JILJ)
104               TVE    = TV(JILJ)
105  !       WRITE(*,*)'dkturb:',GPU,TVE,PHE
106  C
107            CALL KTUTR(IPUMP,ITYP,J2MEQ(J),IWRIT,T,ALFPU,ROPU,ROLM,ROVM,
108        &              PH1,QNP,QNPL,QNPV,GPU,UMN,DUDT,QFLUID,
109        &              CLIMV(JUFE),TOPD(JU),INXUMN,PHE,TVE)
110  C
111               Z(JU) = UMN
112               DZ(JU) = DUDT
113  C
114        ENDIF
115  C
116  C --- ADD PUMP POWER TO FLUID
117  C
118  !      QFH      = 0.5D0*QFLUID
119        QFH      = QFLUID
120  C     IF PUMP DEFINED AT THE BEGIN OR AT THE END OF A PIPE
121        IF(J.EQ.IJLOA.OR.J.EQ.IJROA) QFH = QFLUID
122  C
123        XQM2      = XQM(JILJ) * XQM(JILJ)
124  !      QI (JILJ) = QI (JILJ) + QFH
125  !      QLI(JILJ) = QLI(JILJ) + QFH*(1.D0-XQM2)
126  ! Nur dem Dampf wird Energie entzogen
127  !      QVI(JILJ) = QVI(JILJ) + QFH*XQM2
128   !     QVI(JILJ) = QVI(JILJ) + QFH
129        XQM2      = XQM(JIRJ) * XQM(JIRJ)
130        QI (JIRJ) = QI (JIRJ) + QFH
131  !      QLI(JIRJ) = QLI(JIRJ) + QFH*(1.D0-XQM2)
132  !      QVI(JIRJ) = QVI(JIRJ) + QFH*XQM2
133        QVI(JIRJ) = QVI(JIRJ) + QFH
134  C
135        RETURN
136        END
```

Figure 6.3 Part of the subroutine dkturb.f. Comments are in green.

The calculation of the pressure drop and the power extraction is calculated in subroutine ktutr.f (see Figure 6.4). In order to perform the power drop calculation after the

equation (6.5) the pressure and temperature in the CV right of the turbine junction have to be known. This data is available in a variables array.

```
64   !      Eingabeparameter
65          INTEGER, INTENT(IN) :: ITURB,KEY2M,IWRITE,ITYP,TOPDU
66          DOUBLE PRECISION, INTENT(IN)     :: T,PPU,ALFPU,QNP,UPU,ROLPU,
67         &                    ROVPU,QNPL,QNPV,DUDT,CLIMU,ROPU,GPU,PHE,TVE
68          LOGICAL, INTENT(IN) :: INXU
69          DOUBLE PRECISION, INTENT(OUT)    :: QPU
70   !       DOUBLE PRECISION, DIMENSION(5,2):: TURBDAT
71          DOUBLE PRECISION :: TERMA,TERMB,p0e,p0a,m0,T0in,eta0
72
73          p0e  = TURBDAT(1,ITURB)
74          p0a  = TURBDAT(2,ITURB)
75          m0   = TURBDAT(3,ITURB)
76          T0in = TURBDAT(4,ITURB)
77          eta0 = TURBDAT(5,ITURB)
78
79          QPU    = 0.D0
80          RHOPU(ITURB) = ROPU
81   !       WRITE(*,*)'ktutr1:',GPU,TVE,PHE
82          TERMA=(((GPU*p0e*sqrt(TVE+273.15))/
83         &        ((m0*PHE*SQRT(T0in))+1.D-6))**2)
84          TERMB=(1-(p0a/p0e)**2)
85   C
86          DPP(ITURB)= -(PHE -PHE*SQRT(1-TERMA*TERMB))
87   C
88   !          WRITE(*,*)'ktutr:',ITURB,DPP(ITURB)
89              IF(KEY2M.GT.0) THEN
90                 DPPL(ITURB) = DPP(ITURB)
91                 DPPV(ITURB) = DPP(ITURB)
92              ENDIF
93   !
94   C
95   ! Die Energie darf nur dem Dampf entzogen werden !!!
96   C          PUMP POWER TO FLUID
97              QPU=eta0*GPU*DPP(ITURB)/ROPU
98   !           WRITE(*,*)'ktutr:',T,ITURB,DPP(ITURB)
99              GOTO 250
100  C
101  C
102  C --- WRITE RESULTS IF REQUIRED
```

Nominal values of the turbine junction

Δp_i calculation after equation (6.5)

Power extraction after equation (6.18)

Figure 6.4 Subroutine ktutr.f calculates the pressure drop and the power extraction.

For the internal efficiency, given the considerations stated in section 4.2, a constant value has been chosen for all the stages. A typical value of 0.87 has been chosen (Ray, 1980) for the simulations.

6.3.2 Modelling of the water and steam extractions

In order to couple several stages, steam and water extractions have to be included in the simulation. This has been done adding two fill junctions after every stage; one for the steam and the other for the water. Given the fact that a fill must be always be directed toward the TFD system, the only way to add a fill between two stages (i.e. not at the leftmost junction of a pipe) is via a Single Junction Pipe (for a complete description of the SJP see (GRS, 2009)).

Between every stage, the leftmost extraction is the water extraction and the rightmost extraction is the steam extraction (see Figure 4.4). In order to simplify the explanation, the concept stage will hereinafter comprehend the set of a turbine junction and the extractions after that junction (at the right of that junction).

The mass flow to be extracted by every fill is set by the user in the input data set (see Figure 6.5). Notice that there are two values for every extraction. The purpose of this is to avoid that extraction takes place before a semi-stationary main mass flow is achieved as it could result in instability and, in the case of water extraction, in the paradox of extracting more water than the moisture present in a given stage resulting in an abrupt end of the simulation.

```
WGSTART2 = 0.0D+0
WGENDE2 = -#D+00
WGSTART3 = 0.0D+0
WGENDE3 = -#D+0
WGSTART4 = 0.0D+0
WGENDE4 = -#D+0
WGSTART5 = 0.0D+0
WGENDE5 = -#0D+0
GSTART1 = 0.0D+0
GENDE1  = -#D+0
GSTART2= 0.0D0
GENDE2 = -#D+00
GSTART3= 0.0D0
GENDE3 = -#D+0
GSTART5= 0.0D0
GENDE5= -#D+0
```

Figure 6.5 Detail of input data set. Mass flows in every extraction. WGSTART/ GSTART and WGENDE/GENDE are the mass flows of water/steam extractions at the beginning and after a given time of the simulation.

The fill junctions in every extraction can be divided in water and steam extractions. The fills of the steam extractions are not different from any fill. Given the fact that it is an extraction, the mass flow has to be set negative. This mass flow extracted will have exactly the same properties as the steam in the turbine.

The water extractions however cannot be represented as ordinary fills, because, as explained in chapter 5, the properties of the fluid extracted will be the ones of the fluid in the adjacent CV (i.e. the turbine). In order to extract only water and to achieve the steam quality improvement effect described by Figure 4.4, some modifications had to be made to the fill subroutine.

In Figure 6.6 a detail of subroutine dfk1ha.f can be seen. For a regular discharge fill junction, the steam quality through that junction (XXM) is given by the steam quality in the CV upstream of that junction. By multiplying the steam quality and the extraction mass flow, the quantity of water and steam to extract is obtained.

```
496            if (TURBTEIL(J)==2) then
497                write(*,*)'Fill ist Anzapfung',TURBTEIL(J)
498                XXM=0
499            endif
500            Z2      = ZG*XXM
501            GLJ(J) = ZG-Z2
502            GVJ(J) = Z2
```

Figure 6.6 Detail of subroutine dfk1ha.f where the quality of the steam to be extracted can be set.

The modification consists in adding a new input camp in the declaration of a fill junction in the input data set. This input value set by the user is to be one in the case of a water extraction and zero for all other cases.

In the case of a water extraction, the steam quality is automatically set to zero, thus extracting only water, being the mass flow of water extracted determined by GCSM signals, i.e. by the user.

The real water extractions occur as a consequence of the internal conditions in the turbine and the operation mode, therefore a more realistic option has been developed as an alternative (see Figure 6.7 and Figure 6.8) where the user only has to fill in the input data of the percentage of water in a stage to be extracted (i.e. the steam quality improvement).

The way this model operates is quite simple; the steam quality in the CV at the outlet of the closest (upstream) turbine junction together with the mass flow through that same turbine junction is read by the subroutine thus calculating the water mass flow. The mass flow to be extracted results from the product of the water mass flow and the ABGRAD variable set by the user (proportion of water)

A model for these percentages is still to be developed so that the user does not have to set these values for every operation point.

```
K---- WEXTRA4
@  SGFLOW     SGENTH        ISANZ    ABGRAD
    'SGFLOW4W'    'DEFAULT'    1      0.xxx
@
```

Figure 6.7 **Detail of the input data set for a water extraction. ISANZ equal to one implies that it is a water extraction and ABGRAD is the percentage of water in the stage to be extracted.**

```
                if (TURBTEIL(J)==2) then
                    KKK=JTFOBJ(J)
                    XXM= 0
                    ZG = - ABGRAD(J)*GLJ(IJL(iqtemp(2*KKK)))
!                    write(*,*)'Abzapfgrad:',ABGRAD(J)
                    Z(IG)   = ZG
                    GJ(J)   = ZG
                endif
                Z2      = ZG*XXM
                GLJ(J)  = ZG-Z2
                GVJ(J)  = Z2
```

Figure 6.8 **Detail of subroutine dfk1ha.f where the water mas flow to be extracted is set by a percentage of the water flow through the stage.**

6.3.3 Momentum Flux term

Several stages including their corresponding steam and water extractions have been coupled. At the end the whole LP-turbine has been coupled. The steam and water extractions have been set only in the stages needed. So for instance, the steam after the first stage of the LP-turbine is overheated steam thus making a water extraction unnecessary.

The geometry of the ‘turbine pipe’ has been set so that critical flow is not been achieved. In order to do that, the first approach has been to use a pipe with a constant diameter which is big enough to avoid critical flow at any point. This geometry however is not realistic. In a real turbine the diameter increases after every stage. In order to achieve this, the diameters increase along the turbine in order to have similar Mach

numbers at every stage. The geometry resulting is similar to a cone as it can be seen in Figure 6.9.

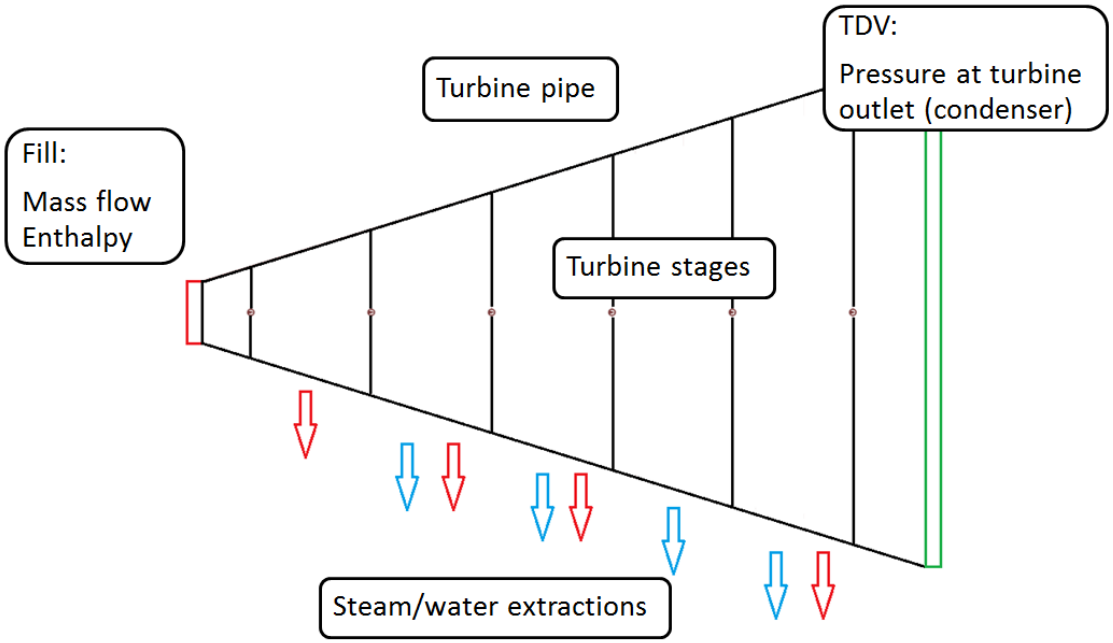

Figure 6.9 Final configuration of the 'turbine pipe'. The vertical black lines with the brown dots represent the turbine junctions and the arrows represent the steam and water extractions.

The basis of the turbine junction and the turbine model is that the only terms different to zero in the momentum balance of equation (5.14) are Δp_s and the source term Δp_l. The former being the pressure difference between CV i and CV i+1 and the latter the turbine differential pressure calculated in the turbine junction according to equation (6.5).

In order to achieve this the momentum flux term and the friction term have been set to zero (the friction term has been set close to a zero value in order to avoid instabilities). The conic geometry proposed above implies a flow cross section increase.

This cross section increase causes a pressure recovery when the momentum flux is calculated, in order to avoid this, in the initial calculation of the simulation; the friction term is adjusted to compensate this pressure recovery. After the initial calculation however and given that the JDPA term has been set to zero (which means that the momen-

tum flux term is not to be calculated during the simulation), the momentum flux term is zero, the modified friction term however, remains modified thus adding a unwanted pressure term due to the friction losses.

In order to avoid any pressure drop due to factors different from the turbine junction, the initial calculation subroutines had to be modified in order to avoid the initial calculation of the pressure recovery in the case of the turbine thus avoiding the friction term adjustment.

6.3.4 Basic thermo-fluid dynamic models used

The thermo fluid dynamics module chosen is the one based on the 5-equation model. This has been done because of the small amount of water in the turbine and because given the lack of information about the turbine's internal geometry, a proper modelling of the phase frontier is not possible. The effects of moisture in the performance of the turbine are considered in the overall efficiency factor.

The friction throughout the turbine model (i.e. pipe and turbine junction) is set to a very low factor close to zero. This is done because the internal geometry of the turbine (number of blades, hydraulic diameters, and length) is not known. So, setting the friction factor to a close to a zero value eliminates any pressure drop other than the one due to the turbine junction. This minimizes the influence of the error committed by the approximation of values such as the diameter and the length of the turbine.

6.3.5 Turbine data required by user input

The nominal values for equation (6.5) (mass flow, steam temperature, and pressure before and after the turbine, and stage efficiency) have to be input by the user in the input dataset (see Figure 6.10).

```
@ @   TURBINE MODEL DATA                                                @
@                                                                       @
@                                                                       @
C---- TURBINE                                                          @
@
@     ATURB (KEYWORD)
K---- PUMP1
@
@    P0E        P0A          M0          T0IN        ETA0
```

Figure 6.10 **Detail of input dataset. Input of a turbine stage nominal values being P0E the pressure before the stage, P0A the pressure after the stage, M0 the mass flow, T0IN the temperature before the stage, and ETA0 the efficiency of the stage.**

7 Results

As explained in chapter 4 the database is limited to the heat mass flow charts of the NPP Phillipsburg 2. Therefore, and considering valid all the assumptions made in chapter 4, the output variables are the pressure, the enthalpy, the temperature, the mass flow and the steam quality.

All the simulations presented below have been carried out for steady state conditions

Figure 7.1 displays the simplest configuration of the model, (Fill, pipe object with turbine junction in the middle and TDV, with no extractions)

The mass flow is determined as boundary condition by the input data, as are the enthalpy at the turbine inlet (both variables determined by the fill junction) and the pressure at the turbine outlet (determined by the pressure set by the user at the TDV). The nominal values of every turbine stage are to be provided by the user in the input data set.

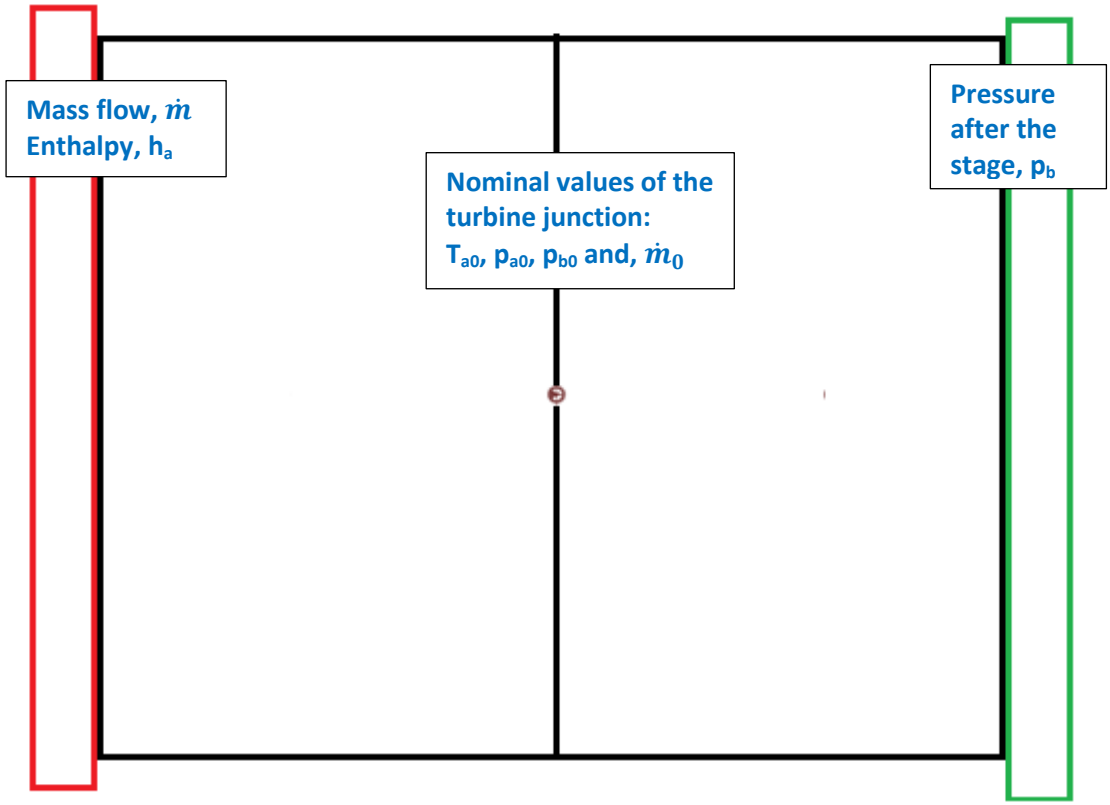

Figure 7.1 Fill, pipe-with-turbine, TDV system with input variables in blue.

The input data can be divided into two categories: The system input data and the simulation input data.

The system input data are only dependant on the reference turbine, so, once the system input variables are set for a given turbine stage or group of stages different simulations can be run without further modifications. The system dependant input variables are T_{a0}, p_{a0}, p_{b0}, $\dot{m}_0$, and η_i (i.e. the nominal values) for every turbine stage in the system (i.e. for every turbine stage in the system a complete set of system input variables has to be set)

The simulation input variables are the boundary conditions of the simulation and have to be defined for every simulation (e.g. a stage or group of stages working at 80% of the nominal power need a different set of simulation variables than the simulation of that same stage or group of stages working at 60% of the nominal power). The simula-

tion input variables are $\dot{m}$ (steam entering the turbine), h_a (defined at the fill junction), p_b (defined in the TDV) and $\dot{m}_{steam\ out}$ (steam mass flow extracted), and $\dot{m}_{water\ out}$ (water mass flow extracted) or water in the stage proportion to be extracted. The extracted water and steam mass flows have to be defined for every extraction in the system and only if the system includes extractions.

The solution variables are the pressure before each turbine stage, the enthalpy after each enthalpy stage, and, for the last simulation with the quality improvement oriented water extractions also the extracted water mass flow.

GRS does not have permission of KWU to publish the data of the reference power plant and therefore all the data will be represented divided by the parameters of the steam at the inlet of the LP turbine (see point LP in Figure 4.). So instead of having values such as p_1, p_2, h_1, h_2 etc. the results will be $p_1/p_{1,100}$, $p_2/p_{1,100}$, $h_1/h_{1,100}$, $h_2/h_{2,100}$ etc. All the values presented in this paper are normalized values (hereinafter referred to as ‘values’)

The per cent error has been calculated after the formula:

$$\%\ error = \frac{calculated\ value - real\ value}{real\ value} \times 100 \tag{7.1}$$

For the HP-turbine the inlet steam is wet steam and its pressure is not known. So, given that it only has two stages and that the conditions at the inlet are not known for simulation purposes the HP-turbine has only one stage because it is not possible to compare the results of the simulation with real values, besides even the nominal values of the first stage of the HP-turbine seem to be inaccurate (see Figure 4.2). At the beginning it was not clear that the cone law with its simplification for ideal gas (see equation (3.4)) would work for wet steam and therefore it was decided to make the first trials with stages working with overheated steam and where the errors implied in the initial assumptions made (see section 4.2) would have as little influence in the results as possible; it is for this reason that the first trials and simulations haven been made with the first stages of the LP-turbine, where the steam is overheated and where all the parameters of the inlet steam are known.

7.1 Simulation of all the stages separately

The first step has been to model every stage of the turbine separately without any extractions in order to observe solely the behaviour of the turbine junction for steady state conditions (Figure 7.1 is a graphic representation of the system).

7.1.1 Pressure at the inlet of every stage

All the stages of the LP turbine have been simulated separately. The results for the pressure and the error committed at every stage are shown in Table 7.1, Table 7.2 and Table 7.3, for 80%, 60% and 40% of nominal power, respectively. A graphic representation of the pressure and the enthalpy along the stages T3, T4 and T5 of the LP-turbine for the 80% case can be seen in Figure 7.2 and Figure 7.3.

Table 7.1 Pressure at the inlet of the LP-turbine stages, plant working at 80% of the nominal power.

Stage	Simulation inlet pressure	Real inlet pressure	Per cent error
T3	0,791788991	0,79174312	0,006%
T4	0,319001468	0,31651376	0,786%
T5	0,141114495	0,14220183	-0,765%
T6	0,047221495	0,04825688	-2,146%
T7	0,026403156	0,02688073	-1,777%
T8	0,011666459	0,01165138	0,129%

Table 7.2 Pressure at the inlet of the LP-turbine stages, plant working at 60% of the nominal power.

Stage	Simulation inlet pressure	Real inlet pressure	Per cent error
T3	0,58711156	0,58990826	-0,474%
T4	0,2379122	0,23302752	2,096%
T5	0,10404303	0,10550459	-1,385%
T6	0,03512752	0,03614679	-2,820%
T7	0,0184393	0,02018349	-8,642%
T8	0,00791484	0,00880734	-10,134%

Table 7.3 Pressure at the inlet of the LP-turbine stages, plant working at 40% of the nominal power.

Stage	Simulation inlet pressure	Real inlet pressure	Per cent error
T5	0,06948744	0,0706422	-1,635%
T6	0,02341212	0,02431193	-3,701%
T7	0,01208187	0,01357798	-11,019%
T8	0,00533608	0,00605505	-11,874%

Notice that the pressure at stages T3 and T4 has not been simulated; the reason for this is that there is no real data available about these stages at 40% power output and comparison with the simulation is therefore not possible.

As it can be seen the error is only significant for the last stages for the 60% and 40 % operation points. As a first approximation, however, it is considered acceptable.

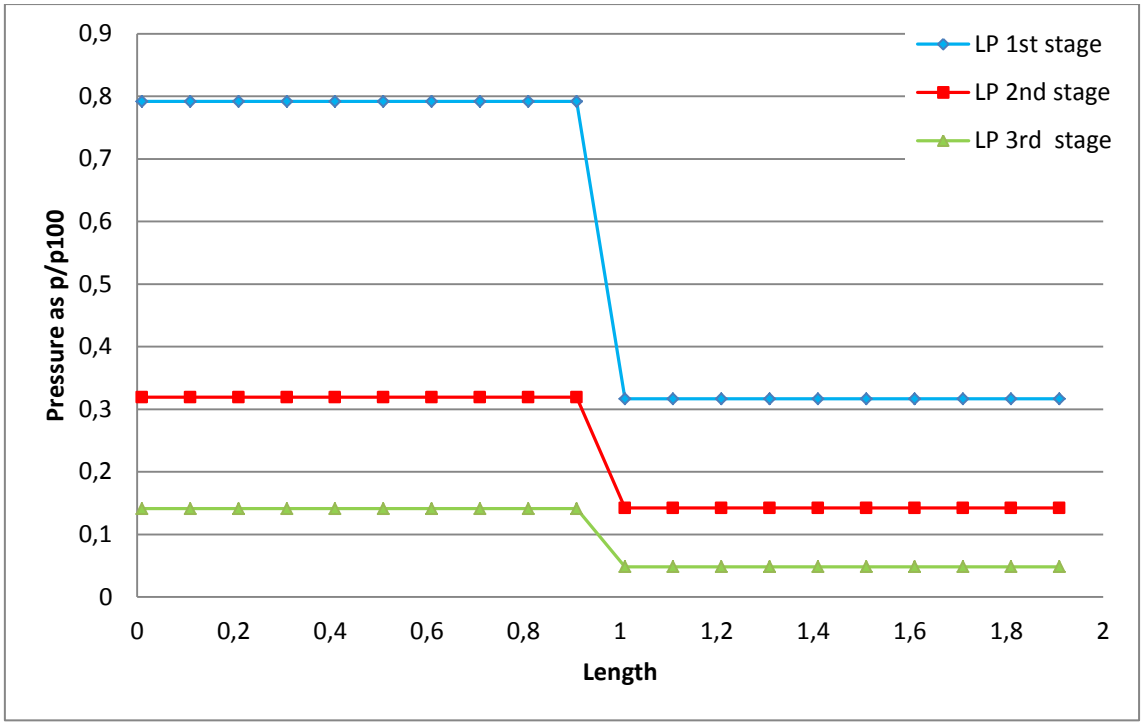

Figure 7.2 Calculated pressure behaviour along stages T3, T4 and T5 of the LP turbine

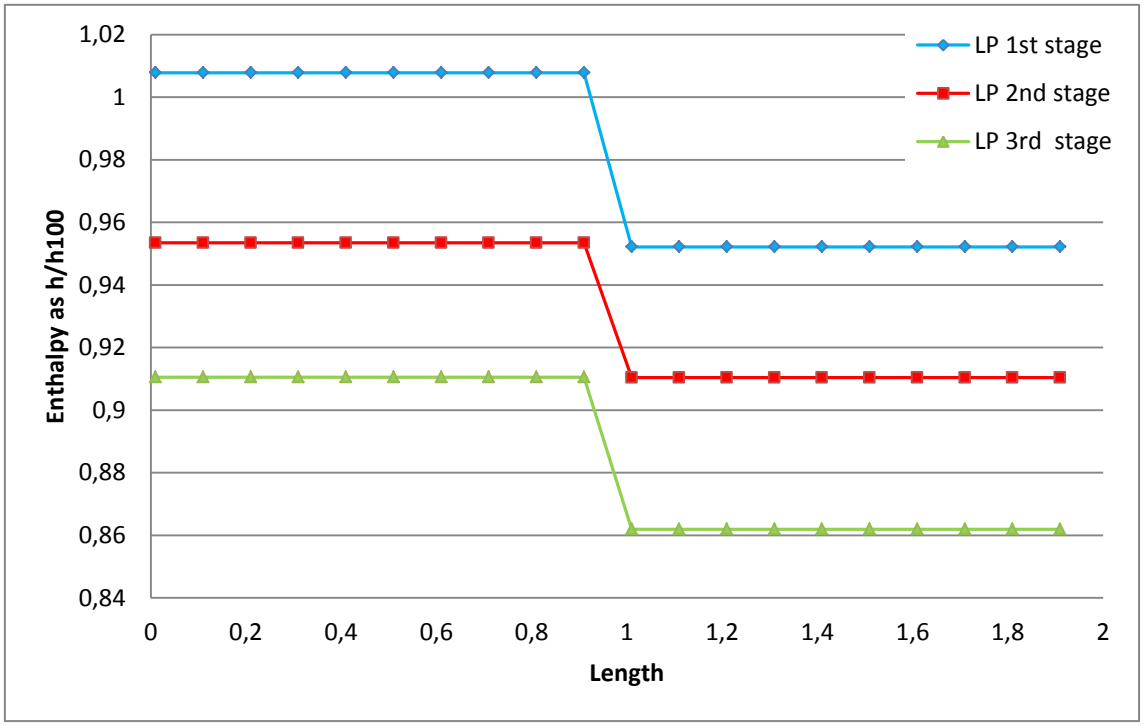

Figure 7.3 Calculated enthalpy along stages T3, T4 and T5 of the LP turbine

7.1.2 Enthalpy at the outlet of every stage

The results for the enthalpy after every stage can be seen in Table 7.4, Table 7.5 and Table 7.6, for 80%, 60% and 40%, respectively.

Table 7.4 Enthalpy at the outlet of the LP-turbine stages for the plant working at 80% of the nominal power.

Stage	Simulation outlet enthalpy	Real outlet enthalpy	Per cent error
T3	0,95214298	0,95340749	-0,133%
T4	0,91037398	0,91053411	-0,018%
T5	0,8619126	0,85296453	1,049%
T6	0,83476118	0,82358954	1,356%
T7	0,79694549	0,8055026	-1,062%
T8	0,75333379	0,79496539	-5,237%

Table 7.5 Enthalpy at the outlet of the LP-turbine stages for the plant working at 60% of the nominal power.

Stage	Simulation outlet enthalpy	Real outlet enthalpy	Per cent error
T3	0,959080547	0,96029478	-0,126%
T4	0,91702779	0,91690485	0,013%
T5	0,868961741	0,86247058	0,753%
T6	0,772795206	0,77378909	-0,128%
T7	0,805454733	0,80837214	-0,361%
T8	0,778101519	0,79606736	-2,257%

Table 7.6 Enthalpy at the outlet of the LP-turbine stages for the plant working at 40% of the nominal power.

Stage	Simulation outlet enthalpy	Real outlet enthalpy	Per cent error
T5	0,87594752	0,873446055	0,286%
T6	0,84850339	0,759540481	11,713%
T7	0,81307931	0,811627977	0,179%
T8	0,7861073	0,799269947	-1,647%

The error for the enthalpy is also acceptable, except after stage T6 operating at a 40% of nominal power. The real value of the enthalpy after stage T6 is not known as in extraction A5 only water is extracted (see section 4.2). Therefore it can be only said that there is a deviation from the interpolated value, not necessarily an error.

Although the errors in the simulation of every separate stage cannot be neglected, it can be accepted that the proposed turbine junction model is accurate enough to proceed to the coupling of several stages with their corresponding water and steam extractions.

7.2 Simulation of the whole LP-turbine with steam and water extractions, and constant diameter of the "turbine pipe"

A new TFD system has been developed including all the stages and its corresponding extractions. As it can be seen in Figure 7.4, the turbine pipe configuration remains unchanged except by the addition of the turbine junctions and the extractions.

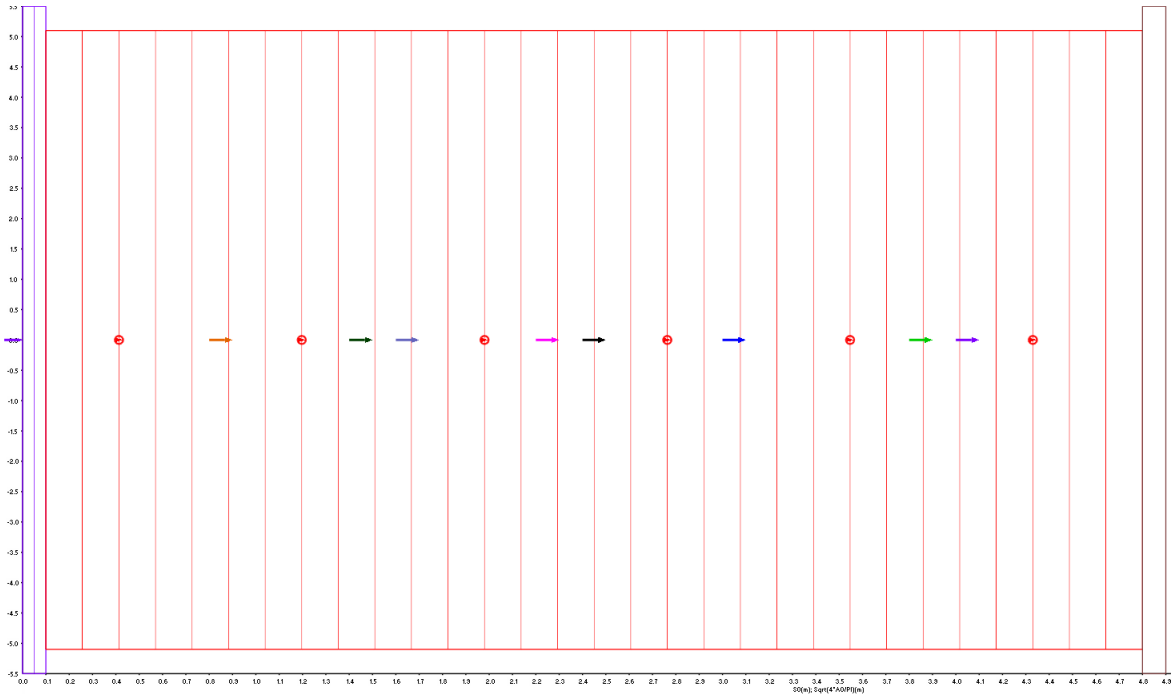

Figure 7.4 **TFD system. The points are the turbine junctions and the arrows the steam and water extractions.**

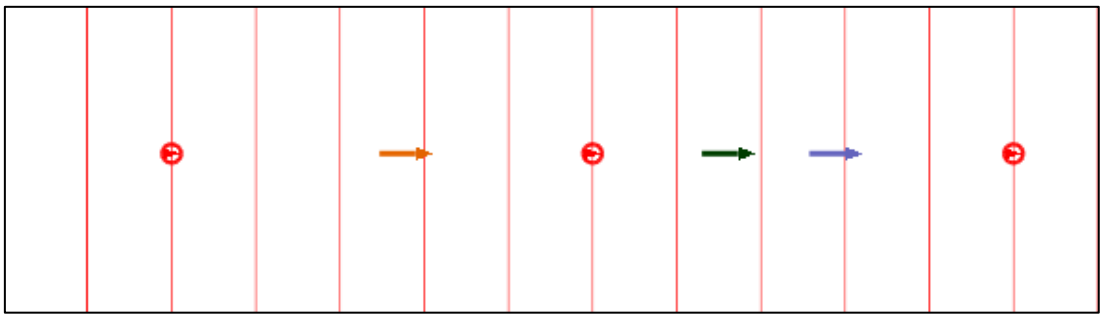

Figure 7.5 **Detail of Figure 7.4. Turbine junctions and extractions can be seen clearly. Between some stages, there is only one extraction as only steam (between stages T3 and T4) or only water (between stages T6 and T7) is extracted.**

7.2.1 Pressure evolution along the LP-turbine

Because the pressure drop occurs only at the turbine junctions (i.e. at the stages), the pressure after one stage is exactly the same as the pressure before the next stage.

The pressure after stage T8 (i.e. in the condenser) is not displayed as it is a boundary condition and is not calculated by the TFD system, but set by the user.

Table 7.7 Pressure at the LP-turbine stages inlets, plant working at 80%, 60% and 40% of the nominal power

	80%			60%			40%		
Stage	Simulation pressure	Real pres-sure	Per cent error	Simulation pressure	Real pressure	Per cent error	Simulation pressure	Real pres-sure	Per cent error
T3	0,7937	0,7917	0,251%	0,5897	0,5899	-0,038%	0,3843	0,3844	-0,037%
T4	0,3217	0,3165	1,647%	0,2397	0,2330	2,870%	0,1609	N/A	N/A
T5	0,1488	0,1422	4,653%	0,1096	0,1055	3,882%	0,0732	0,0706	3,684%
T6	0,0480	0,0483	-0,532%	0,0357	0,0361	-1,276%	0,0238	0,0243	-2,073%
T7	0,0270	0,0269	0,399%	0,0201	0,0202	-0,356%	0,0135	0,0136	-0,568%
T8	0,0119	0,0117	1,723%	0,0089	0,0088	0,596%	0,0061	0,0061	0,217%

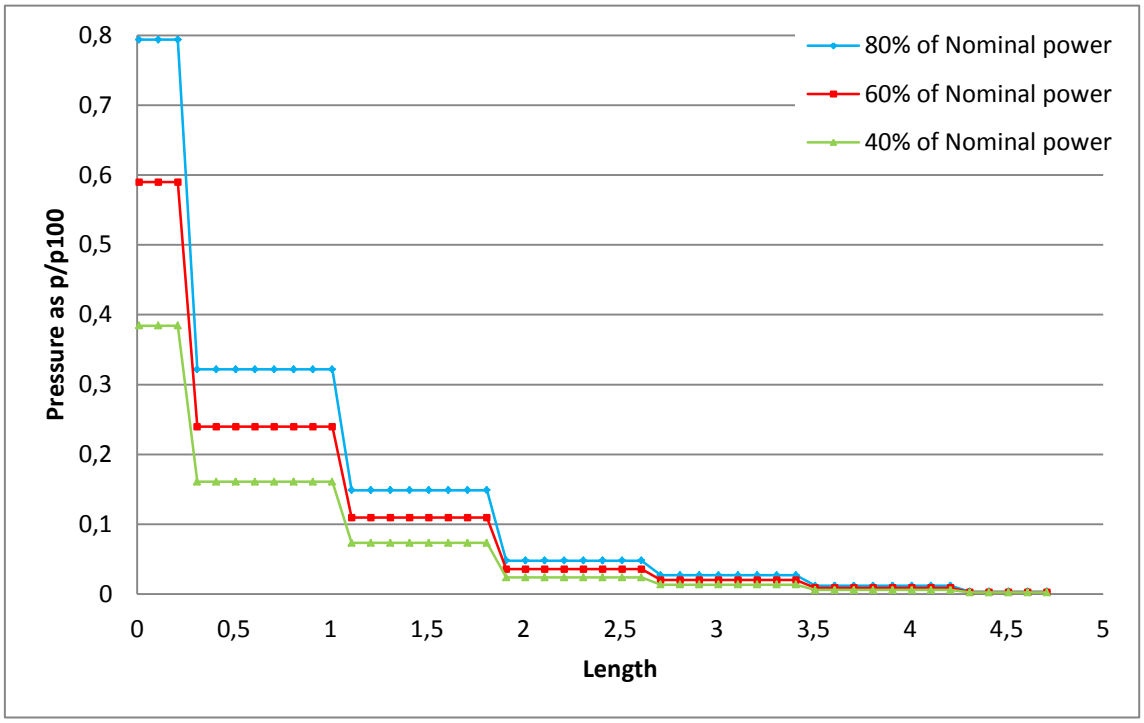

Figure 7.6 Calculated pressure behaviour along the LP turbine, plant working at 80%, 60% and 40% of the nominal power.

As can be seen in Table 7.7 the results are significantly better for the simulation of the whole turbine than for the simulation of the stages separately, the error being below 5% in all the three simulations. Figure 7.6 displays clearly the pressure drop after every stage. Notice that the main pressure drop takes place at the first stage.

7.2.2 Enthalpy evolution along the LP-turbine

The enthalpy variations in the turbine take place along the stages and during the water extractions. Therefore the enthalpy has to be displayed before and after every stage, as if a water extraction takes place, it is no longer true as it is for the pressure, that the enthalpy after a stage is equal to the enthalpy before the following stage.

In Table 7.8 the enthalpy at the inlet and at the outlet of every stage is given. Hence between two given stages two enthalpies are displayed, the one at the outlet of the first stage and the other at the inlet of the following stage or, accordingly, one enthalpy value before the water extraction and another after the water extraction.

Table 7.8 Enthalpy before and after every stage of the LP-turbine, plant working at 80%, 60% and 40% of the nominal power

	80%			60%			40%		
Stage	Simulation enthalpies	Real enthalpies	Per cent error	Simulation enthalpies	Real enthalpies	Per cent error	Simulation enthalpies	Real enthalpies	Per cent error
T3	1,0078	1,0078	0,000%	1,0166	1,0166	0,000%	1,0252	1,0252	0,000%
	0,9529	0,9534	-0,057%	0,9604	0,9603	0,006%	0,9692	N/A	N/A
T4	0,9529	0,9534	-0,057%	0,9604	0,9603	0,006%	0,9692	N/A	N/A
	0,9122	0,9105	0,188%	0,9185	0,9169	0,175%	0,9260	0,9242	0,190%
T5	0,9122	0,9105	0,188%	0,9185	0,9169	0,175%	0,9260	0,9242	0,190%
	0,8610	0,8530	0,945%	0,8678	0,8625	0,614%	0,8744	0,8734	0,111%
T6	0,8681	0,8600	0,938%	0,8716	0,8662	0,616%	0,8744	0,8734	0,111%
	0,8420	0,8236	2,237%	0,8455	0,7738	9,263%	0,8485	0,7595	11,706%
T7	0,8515	0,8331	2,215%	0,8536	0,8384	1,807%	0,8542	0,8444	1,161%
	0,8146	0,8055	1,130%	0,8165	0,8084	1,006%	0,8178	0,8116	0,758%
T8	0,8281	0,8189	1,124%	0,8286	0,8203	1,011%	0,8281	0,8218	0,768%
	0,7597	0,7950	-4,440%	0,7635	0,7961	-4,094%	0,7732	0,7993	-3,260%

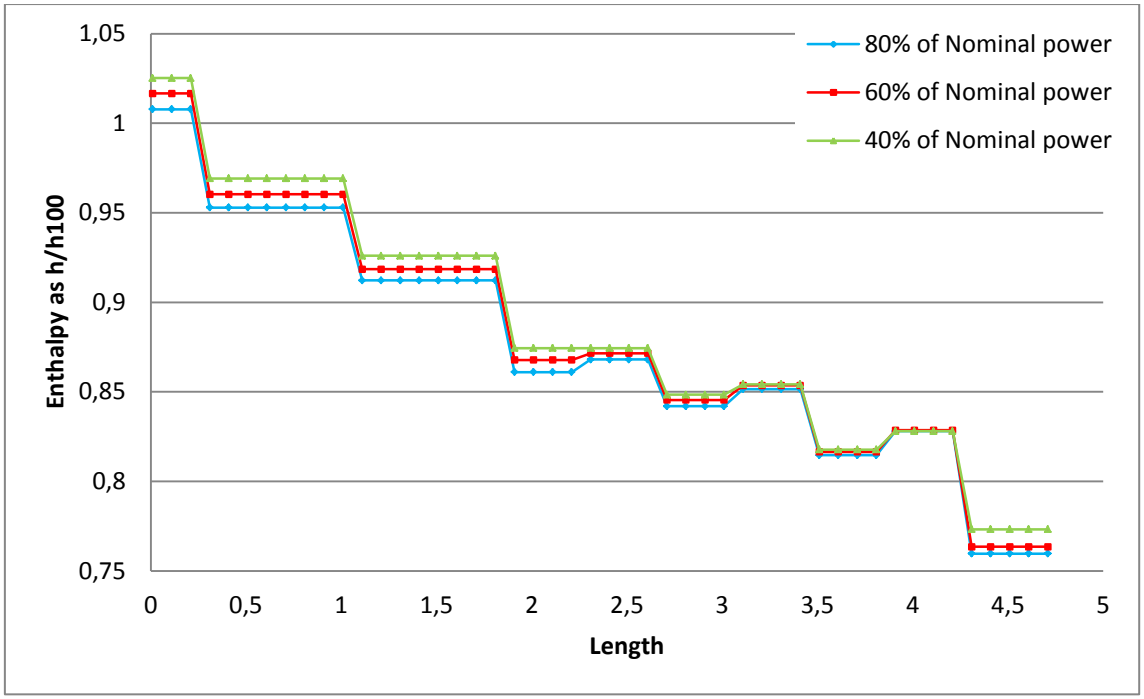

Figure 7.7 Calculated enthalpy behaviour along the LP-turbine, plant working at 80%, 60% and 40% of the nominal power.

In Figure 7.7 the effect of the water extractions can be clearly seen after stages T5, T6 and T7, and how the importance of these extractions in the enthalpy increase after every extraction cannot be neglected.

7.2.3 Evaluation of results

Table 7.9 shows the error made at every stage. Compared with the results obtained from the simulation of the stages separately (section 7.1), there is a substantial improvement.

Table 7.9 Compared error in the enthalpy (red) and enthalpy (red) results for the different operation points.

Stage	Per cent error at 80%		Per cent error at 60%		Per cent error at 40%	
	Enthalpies	Pressure	Enthalpies	Pressure	Enthalpies	Pressure
T3	0,000%	0,251%	0,000%	-0,038%	0,000%	-0,037%
	-0,057%		0,006%		0,000%	
T4	-0,057%	1,647%	0,006%	2,870%	0,000%	
	0,188%		0,175%		0,190%	
T5	0,188%	4,653%	0,175%	3,882%	0,190%	3,684%
	0,945%		0,614%		0,111%	
T6	0,938%	-0,532%	0,616%	-1,276%	0,111%	-2,073%
	2,237%		9,263%		11,706%	
T7	2,215%	0,399%	1,807%	-0,356%	1,161%	-0,568%
	1,130%		1,006%		0,758%	
T8	1,124%	1,723%	1,011%	0,596%	0,768%	0,217%
	-4,440%		-4,094%		-3,260%	

The error in the pressure calculation stays below 5% in all the simulations and is only above 3% between stages T4 and T5. The magnitude of the error at that point for all the simulations (compared with the other errors) could be explained by some inaccuracy in the measurements at that point or by the assumptions made. The real cause, however, remains unclear.

The error in the enthalpies is quite small for all the points but two. At the condenser (after stage T8) it is due to the deceleration of the steam which transforms its kinetic energy into heat thus increasing the enthalpy, the steam enthalpy at the condenser in the heat balance is considered after this deceleration thus being higher than the actual enthalpy at the outlet of the turbine itself where the steam still has a considerable kinetic energy.

At the outlet of stage T6 the largest error is made by far which might be explained by the fact that the real enthalpy at that point is not known and the value used as reference has been interpolated (see section 4.2).

These considerations being made, the performance of the model is considered quite satisfactory.

7.3 Simulation of the whole LP-turbine with steam and water extractions and conic geometry

The next step has been to adjust the geometry of the turbine pipe to a cone adjusting it to the changing conditions of the steam along it. As explained in subsection 6.3.3 the diameters at the inlet and at the outlet of the “turbine pipe” have been set in order to avoid a critical flow at any point.

The reference power plant has a 6 flow LP-turbine, so that there are 6 LP-turbines. This has also been reflected in the model, the dimensions of it being the corresponding to only one of these 6 LP-turbines. The simulation has been set so that the steam mass flow is divided between 6 identical turbines.

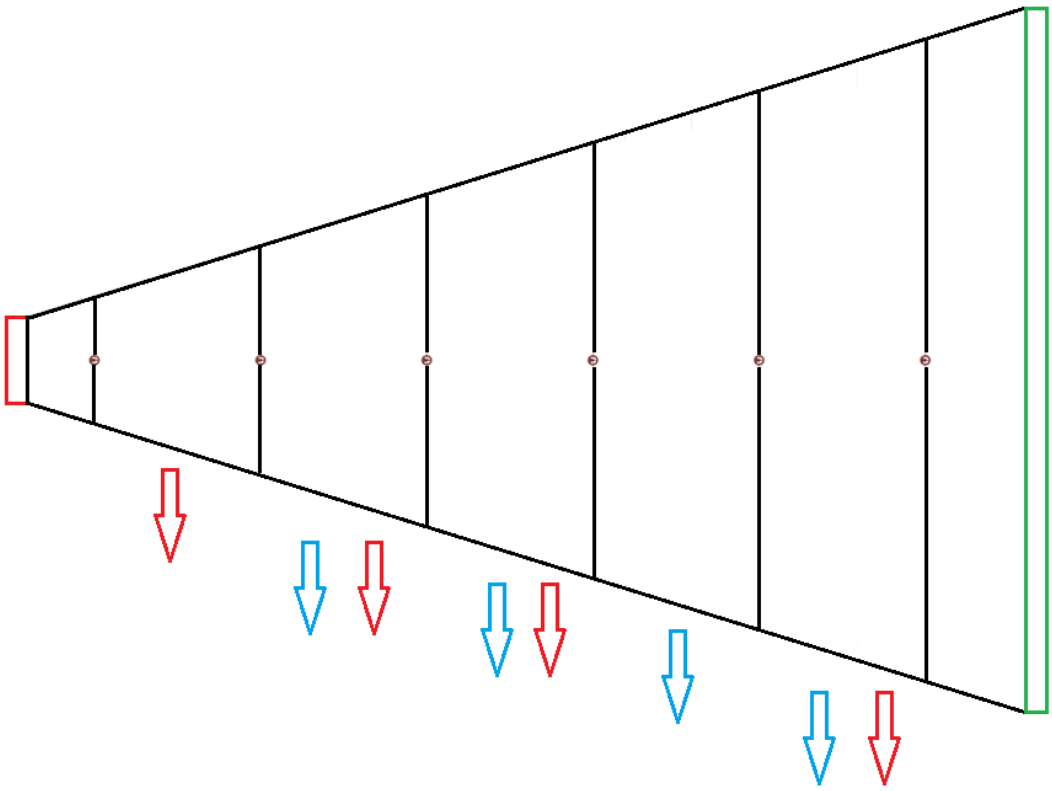

Figure 7.8 Image of the geometry used. The fill and the TDV can be seen at the left and at the right end respectively.

7.3.1 Pressure evolution along the LP-turbine

Table 7.10 Pressure at the LP-turbine stages inlets, plant working at 80%, 60% and 40% of the nominal power

	80%			60%			40%		
Stage	Simulation pressure	Real pres-sure	Per cent error	Simulation pressure	Real pressure	Per cent error	Simulation pressure	Real pres-sure	Per cent error
T3	0,7932	0,7917	0,188%	0,5893	0,5899	-0,105%	0,3840	0,3844	-0,104%
T4	0,3217	0,3165	1,647%	0,2397	0,2330	2,873%	0,1609	%	
T5	0,1488	0,1422	4,646%	0,1096	0,1055	3,891%	0,0733	0,0706	3,702%
T6	0,0479	0,0483	-0,669%	0,0357	0,0361	-1,340%	0,0238	0,0243	-2,073%
T7	0,0269	0,0269	-0,061%	0,0201	0,0202	-0,596%	0,0135	0,0136	-0,634%
T8	0,0116	0,0117	-0,402%	0,0088	0,0088	-0,422%	0,0061	0,0061	0,021%

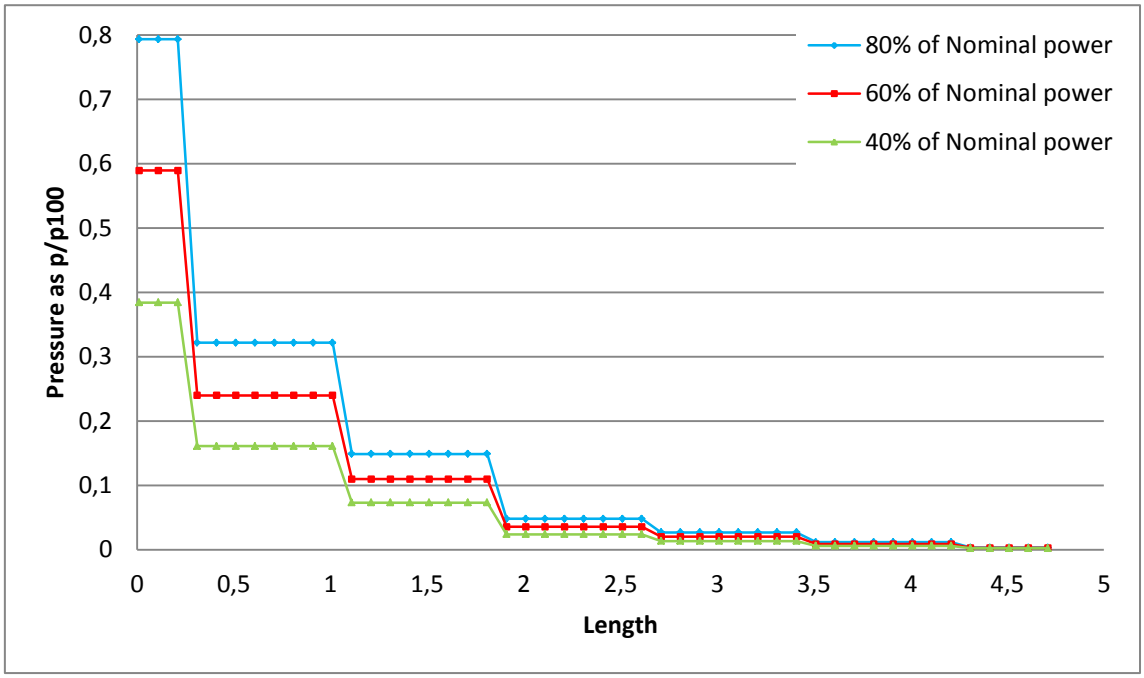

Figure 7.9 Calculated pressure behaviour along the LP turbine, plant working at 80% of the nominal power.

7.3.2 Enthalpy evolution along the LP-turbine

Table 7.11 Enthalpy before and after every stage of the LP-turbine, plant working at 80%, 60% and 40% of the nominal power.

	80%			60%			40%		
Stage	Simulation enthalpies	Real enthalpies	Per cent error	Simulation enthalpies	Real enthalpies	Per cent error	Simulation enthalpies	Real enthalpies	Per cent error
T3	1,0078	1,0078	0,000%	1,0166	1,0166	0,000%	1,0252	1,0252	0,000%
	0,9529	0,9534	-0,057%	0,9604	0,9603	0,007%	0,9692	N/A	N/A
T4	0,9529	0,9534	-0,057%	0,9604	0,9603	0,007%	0,9692	N/A	N/A
	0,9124	0,9105	0,200%	0,9186	0,9169	0,187%	0,9261	0,9242	0,201%
T5	0,9124	0,9105	0,200%	0,9186	0,9169	0,187%	0,9261	0,9242	0,201%
	0,8607	0,8530	0,902%	0,8674	0,8625	0,571%	0,8745	0,8734	0,124%
T6	0,8682	0,8600	0,949%	0,8717	0,8662	0,628%	0,8745	0,8734	0,124%
	0,8419	0,8236	2,225%	0,8454	0,7738	9,260%	0,8485	0,7595	11,711%
T7	0,8518	0,8331	2,246%	0,8539	0,8384	1,848%	0,8546	0,8444	1,210%
	0,8151	0,8055	1,196%	0,8175	0,8084	1,131%	0,8191	0,8116	0,923%
T8	0,8291	0,8189	1,248%	0,8301	0,8203	1,195%	0,8299	0,8218	0,995%
	0,7754	0,7950	-2,462%	0,7796	0,7961	-2,068%	0,7864	0,7993	-1,613%

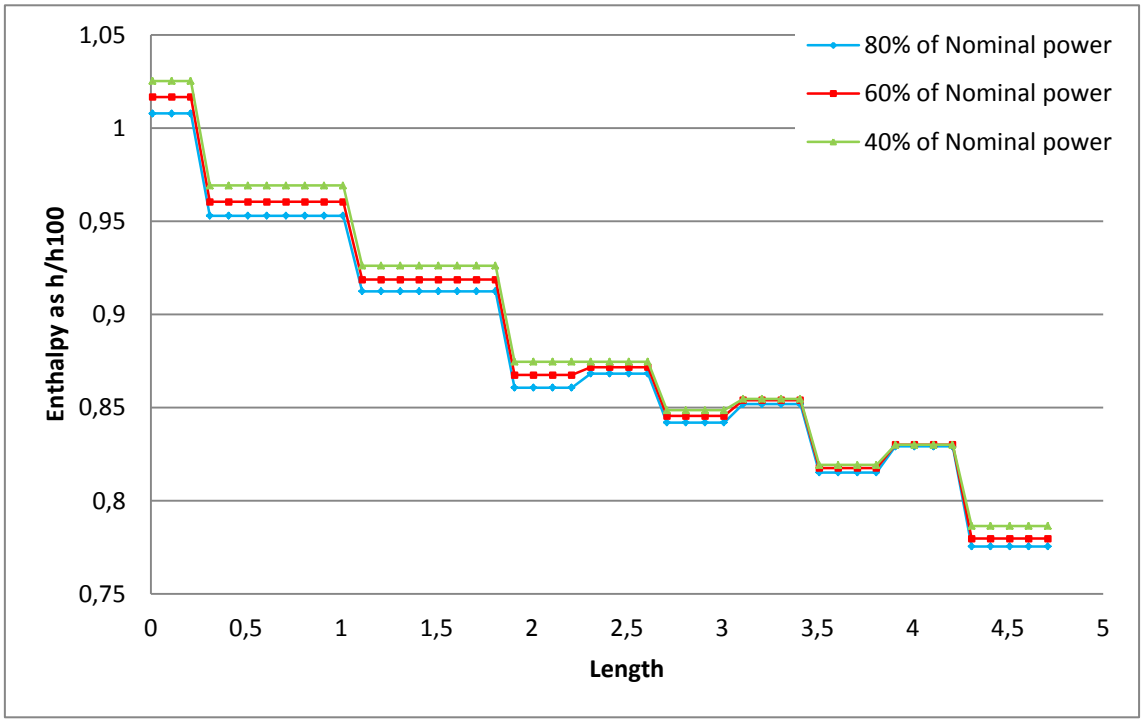

Figure 7.10 Enthalpy along the LP turbine, plant working at 80% of the nominal power.

7.3.3 Evaluation of results

Comparing Table 7.12 and Table 7.9 it is clear that there are no significant differences. Therefore, the elimination of the influence of the increase of the diameter along the turbine can be considered achieved.

Table 7.12 Compared error in the enthalpy (red) and enthalpy (red) results for the different operation points.

Stage	Per cent error at 80%		Per cent error at 60%		Per cent error at 40%	
	Enthalpies	Pressure	Enthalpies	Pressure	Enthalpies	Pressure
T3	0,000%	0,188%	0,000%	-0,105%	0,000%	-0,104%
	-0,057%		0,007%		0,000%	
T4	-0,057%	1,647%	0,007%	2,873%	0,000%	
	0,200%		0,187%		0,201%	
T5	0,200%	4,646%	0,187%	3,891%	0,201%	3,702%
	0,902%		0,571%		0,124%	
T6	0,949%	-0,669%	0,628%	-1,340%	0,124%	-2,073%
	2,225%		9,260%		11,711%	
T7	2,246%	-0,061%	1,848%	-0,596%	1,210%	-0,634%
	1,196%		1,131%		0,923%	
T8	1,248%	-0,402%	1,195%	-0,422%	0,995%	0,021%
	-2,462%		-2,068%		-1,613%	

7.4 Simulation of the whole LP-turbine with steam and water extractions and conic geometry with qualitative water extractions

The last simulation is done with the second water extraction mode, i.e. with a percentage of the water in the stage extracted instead of a fix quantity. Instead of introducing in the input data the water mass flow to be extracted, the user introduces a percentage. From the water present in the CV at the left of the extraction, ATHLET extracts the introduced percentage. For this simulation the mass flow in the extraction lines becomes an additional solution variable.

7.4.1 Pressure evolution along the LP-turbine

Table 7.13 Pressure at the LP-turbine stages inlets, plant working at 80%, 60% and 40% of the nominal power.

	80%			60%			40%		
Stage	Simulation pressure	Real pres-sure	Per cent error	Simulation pressure	Real pressure	Per cent error	Simulation pressure	Real pres-sure	Per cent error
T3	0,79331	0,79174	0,198%	0,58936	0,58991	-0,094%	0,38404	0,38440	-0,094%
T4	0,32176	0,31651	1,658%	0,23975	0,23303	2,886%	0,16093	N/A	N/A
T5	0,14885	0,14220	4,675%	0,10964	0,10550	3,922%	0,07328	0,07064	3,727%
T6	0,04806	0,04826	-0,411%	0,03576	0,03615	-1,082%	0,02384	0,02431	-1,923%
T7	0,02701	0,02688	0,464%	0,02020	0,02018	0,072%	0,01355	0,01358	-0,171%
T8	0,01167	0,01165	0,157%	0,00882	0,00881	0,153%	0,00607	0,00606	0,324%

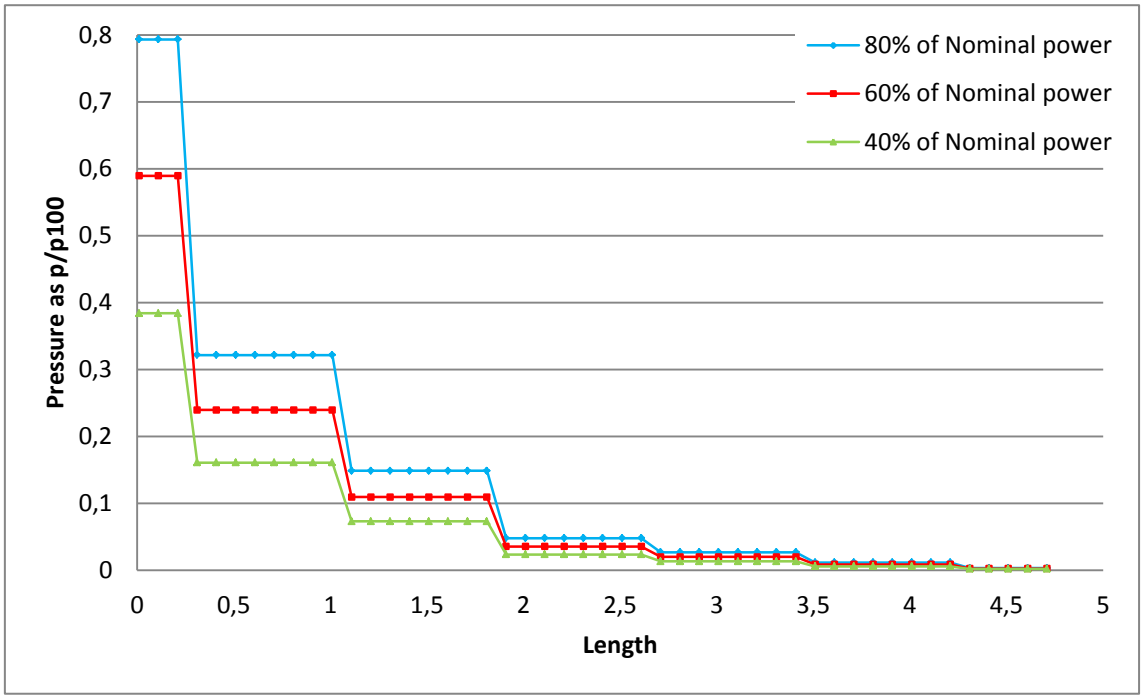

Figure 7.11 Pressure behaviour along the LP turbine, plant working at 80% of the nominal power.

7.4.2 Enthalpy evolution along the LP-turbine

Table 7.14 Enthalpy before and after every stage of the LP-turbine, plant working at 80%, 60% and 40% of the nominal power:

	80%			60%			40%		
Stage	Simulation enthalpies	Real enthalpies	Per cent error	Simulation enthalpies	Real enthalpies	Per cent error	Simulation enthalpies	Real enthalpies	Per cent error
T3	1,00782	1,00782	0,000%	1,01660	1,01660	0,000%	1,02521	1,02521	0,000%
	0,95291	0,95341	-0,052%	0,96041	0,96029	0,012%	0,96920	N/A	N/A
T4	0,95291	0,95341	-0,052%	0,96041	0,96029	0,012%	0,96920	N/A	N/A
	0,91241	0,91053	0,206%	0,91867	0,91690	0,193%	0,92615	0,92424	0,206%
T5	0,91241	0,91053	0,206%	0,91867	0,91690	0,193%	0,92615	0,92424	0,206%
	0,86080	0,85296	0,918%	0,86753	0,86247	0,587%	0,87462	0,87345	0,135%
T6	0,86741	0,86002	0,860%	0,87135	0,86621	0,593%	0,87462	0,87345	0,135%
	0,84129	0,82359	2,150%	0,84532	0,77379	9,245%	0,84872	0,75954	11,741%
T7	0,84848	0,83309	1,847%	0,84925	0,83842	1,292%	0,85108	0,84440	0,791%
	0,81200	0,80550	0,806%	0,81305	0,80837	0,579%	0,81569	0,81163	0,500%
T8	0,82532	0,81886	0,789%	0,82603	0,82028	0,702%	0,82768	0,82176	0,721%
	0,77170	0,79497	-2,927%	0,77561	0,79607	-2,570%	0,78424	0,79927	-1,880%

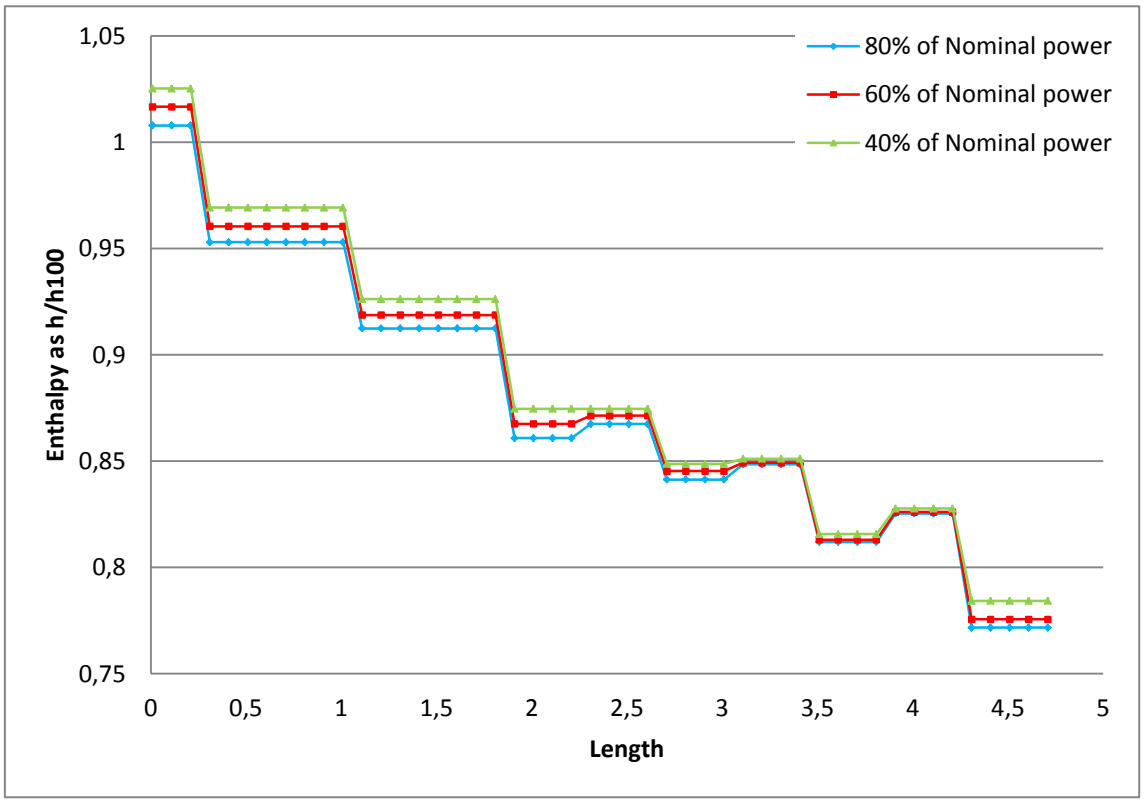

Figure 7.12 Enthalpy along the LP turbine, plant working at 80% of the nominal power.

7.4.3 Water extractions

The water mass flow extracted is calculated by ATHLET. It has been normalized dividing the calculated and real mass flows by the water extracted at the third extraction at nominal power.

Table 7.15 Water extracted for the plant working at 80%, 60% and 40% of the nominal power

	80%			60%			40%		
Extraction	Simulation mass flow	Real mass flow	Per cent error	Simulation mass flow	Real mass flow	Per cent error	Simulation mass flow	Real mass flow	Per cent error
A3	0,4894	0,5606	-12,700%	0,1914	0,2168	-11,715%	0,0000	0,0000	0,000%
A5	0,6319	0,7137	-11,469%	0,3920	0,4455	-12,018%	0,1895	0,2087	-9,205%
A6	0,9144	0,9748	-6,194%	0,5971	0,6450	-7,429%	0,3383	0,3812	-11,241%

The water mass flows extracted in the simulations of sections 7.3 and 7.2 are exactly the same as the real mass flows in Table 7.15 as they are set by the user.

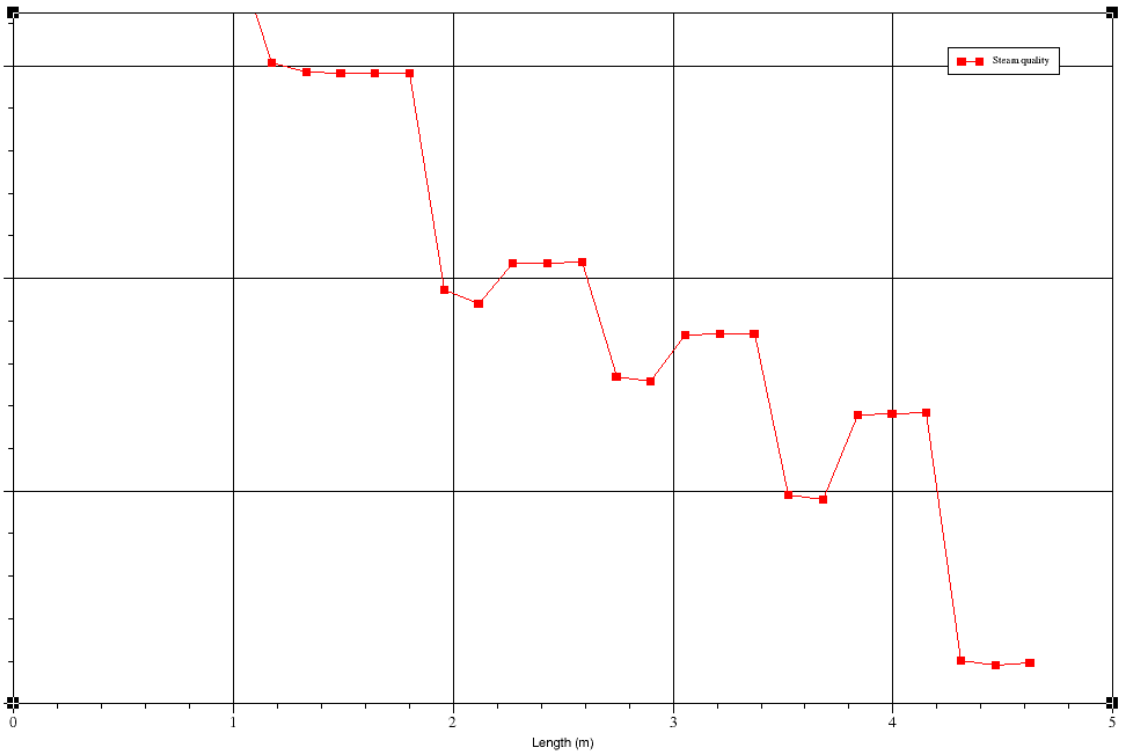

Figure 7.13 Qualitative steam quality evolution across the LP-turbine. Notice the increase after every extraction.

7.4.4 Evaluation of results

The error in the water extractions is of importance as can be seen in Table 7.15, the reason for this is that a small change in the enthalpy of the steam has significant effect on the steam quality; this, together with the large amount of steam flowing across the turbine, causes large variations on the water quantity at the extraction point. As the water extracted depends on the water present at the CV before the extraction point, this water quantity variation is responsible for the errors.

Table 7.16 Compared error in the enthalpy (red) and pressure (blue) results for the different operation points

Stage	Percent error at 80%		Percent error at 60%		Percent error at 40%	
	Enthalpy	Pressure	Enthalpy	Pressure	Enthalpy	Pressure
T3	0,000%	0,198%	0,000%	-0,094%	0,000%	-0,094%
	-0,052%	1,658%	0,012%	2,886%	0,000%	
T4	-0,052%	1,658%	0,012%	2,886%	0,000%	
	0,206%	4,675%	0,192%	3,922%	0,206%	3,727%
T5	0,206%	4,675%	0,192%	3,922%	0,206%	3,727%
	0,933%	-0,411%	0,599%	-1,082%	0,129%	-1,923%
T6	0,857%	-0,411%	0,587%	-1,082%	0,129%	-1,923%
	2,153%	0,464%	9,236%	0,072%	11,736%	-1,923%
T7	2,010%	0,464%	1,693%	0,072%	1,155%	-0,171%
	1,009%	0,157%	1,008%	0,153%	0,897%	0,324%
T8	0,929%	0,157%	0,935%	0,153%	0,850%	0,324%
	-2,790%	0,000%	-2,333%	0,000%	-1,748%	0,000%

Table 7.17 Error in the water mass flow extracted in the third fourth and fifth extraction of the LP-turbine (extractions A3, A5 and A6)

Extraction	Mass flow error at 80%	Mass flow error at 60%	Mass flow error at 40%
A3	-12,700%	-11,715%	0,000%
A5	-11,469%	-12,018%	-9,205%
A6	-6,194%	-7,429%	-11,241%

Despite the errors in the water mass flow extractions, there is no qualitative and little quantitative difference from the errors committed in the pressures and the enthalpies in the fixed extraction simulations (see Table 7.12), the main error being in the enthalpy after stage T6.

8 Assessment

8.1 Geometry data

The two conic models developed have their geometrical data set in order to avoid a critical flow of the steam and are not based on real data. The knowledge of the geometry and the size of the turbine are necessary for the transient calculation, as the speed with which changes in the inlet steam conditions reflect on the conditions across the turbine depends on the quantity of steam stored in the turbine at a given point, i.e. the turbine internal volume.

The lack of geometry data and the unavailability of transient data made it impossible to assess the adequacy of the model for transient calculations.

8.2 Application of the cone law

The representation of the pressure behaviour has been seen in chapter 7, the error always being below 5%. Considering the absolute deviations, the accuracy of the model seems to improve as the largest deviation takes place between the second and the third stages (T4 and T5) and it is below 0,075 bar which could be in the area of the measurement tolerances (Grote, 2009). It can therefore be considered that the cone law in the form it has been used in the present model (see equation (6.5)) as well as the model as a whole depict the steady state pressure behaviour through the turbine with great accuracy.

8.3 Enthalpy calculation

The representation of the enthalpy behavior presents the main problems and errors. Despite the simplified model used for its calculation (see equation (6.18)), the problems seem to arise due to an incomplete knowledge of some relevant data (see Chapter 4) and due to some of the assumptions made. However, should accurate data be availa-

ble, it is to be expected that the model would depict the enthalpies steady state behavior with an even more satisfactory accuracy.

8.4 Models to be developed

A model for the extractions still needs to be developed. Some empirical formula should be able to describe the amount of fluid extracted and the improvement of the steam quality in the turbine resulting from this extraction. The modifications made in the water extractions (via the fill junction, see Figure 6.8) allow the implementation of this relationship once it is developed.

In the present simulation, given the lack of reliable data (see subchapter 6.3.1.1), a constant efficiency has been used and the variation of this efficiency depending on the power output of the turbine has been neglected. The real internal enthalpy drop at every stage at nominal power is not known, and therefore, given the available data, it has not been possible to adjust the internal efficiency of every stage according to equation (3.7).

Should the extraction model be developed and the nominal stage efficiencies be known, it would be possible to introduce in the subroutine ktutr.f (see Figure 6.4) a calculation for the real internal efficiency of a given stage after equation (3.7).

In order to display the total power extracted from the fluid or, accordingly, the total power output of the turbine, a variable has to be developed which adds the energy extracted from the fluid at every stage (equation (6.18)) and makes it possible to directly see the power output of the turbine (see equation (8.1)).

$$P = \sum \dot{Q} \tag{8.1}$$

8.5 Extension

Although not part of the turbine itself, some models need to be developed in order to successfully couple the turbine model with a plant simulation.

In order to model the whole turbine, i.e. the HP and the LP-turbine in a same simulation, a water separator as well as a re heater have to be included in the simulation. The re heater can be modelled as a heat exchanger and for the moisture separator the moisture separator of the steam generator can be modified. An alternative option is to implement it as a modified water extraction junction.

Given that the steam quality of the steam at the outlet of the moisture is known and constant for all the operation points and that both the inlet mass flow as well as the inlet steam quality are known (x_{out}, $\dot{m}_{in}$ and x_{in} respectively in Figure 8.1), the modelling of the moisture separator is quite simple.

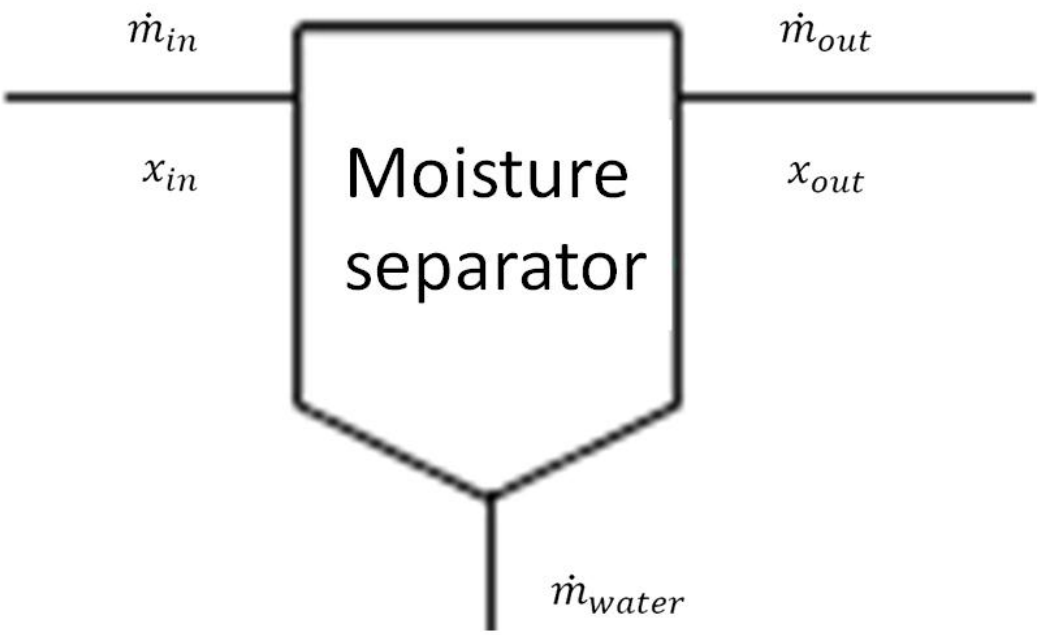

Figure 8.1 Moisture separator (detail of Figure 4.)

Making a mass balance:

$$\dot{m}_{in} = \dot{m}_{out} + \dot{m}_{water} \tag{8.2}$$

And knowing that:

$$\dot{m}_{out}\, x_{out} = \dot{m}_{in}\, x_{in} \tag{8.3}$$

The water mass flow to be extracted is:

$$\dot{m}_{water} = \dot{m}_{in}\left(1 - \frac{x_{in}}{x_{out}}\right) \tag{8.4}$$

All the variables on the right side of equation (8.5) are known. So that the moisture separator can be easily modelled using the modified fill junctions used for the water extractions in the turbine (see subsection 6.3.2)

For the transient calculations that involve a variation of the angular speed of the turbine, such as load rejection and rump up, a momentum balance determines the variation of angular speed.

The power in a rotational system, i.e. a turbine is defined by the product of the torque M and the angular velocity ω.

$$P = \omega\, M = 2\pi f\, M \;\rightarrow M_t = \frac{P}{2\pi f} \tag{8.5}$$

The momentum balance equation of the whole system equation (8.6) results from a breaking torque, the moment of inertia Θ of the whole rotating machine and the torque provided by the turbine.

$$\frac{P}{\omega} = \Theta\frac{d\omega}{dt} + M_{br} \;\rightarrow\; \frac{P}{f} = 2\pi\Theta\frac{df}{dt} + M_{br} \tag{8.6}$$

$$\frac{df}{dt} = \frac{1}{2\pi\Theta}\left(\frac{P}{2\pi f} - M_{br}\right) \tag{8.7}$$

Equation (8.7) describes the angular velocity variation. The breaking torque term M_{br} includes all the components of the equation different from the turbine torque and have to be modelled.

9 Summary and Outlook

The object of this paper was to develop and to implement a turbine model in ATHLET. Although some aspects of the turbine have not been considered yet, the basic equations and the general layout of the model have been successfully implemented.

The thermo-hydraulic model developed needs only the thermodynamic properties of the steam at some points of the plant (basically extractions, turbine inlet as well as turbine outlet) operating at nominal power. The performed steady-state simulations predict the behaviour of the steam pressure as well as the enthalpy with very good precision, the error being below 5% at all the simulations. The fact that almost all the necessary input data is to be found at the plant heat and mass balances easies the use of the model as it is data to which the potential user is expected to have access.

So far steady state calculations for a KONVOI LP-turbine have been successfully carried out. In principle the model should describe the transient conditions with the similar accuracy as it describes the steady state conditions. However, the fact that the water extraction model needs to be improved is expected to diminish the accuracy of the results. In order to perform such calculations, variables such as the geometry of the turbine (internal volume), the real isentropic efficiency of every stage, the moment of inertia of the turbine-generator complex as well as the nominal pressure at the HP-turbine inlet are necessary.

In order to couple the turbine model with the secondary loop, the models for the water extractions still need to be improved. The final result however is more than satisfactory as the main aspects of the turbine modelling have been successfully addressed thus laying the basis for the implementation of the whole turbine once the necessary reference and validation data is available.

The Steam turbine being one of the most complex components of the secondary loop, the simulation capabilities of ATHLET regarding the secondary loop have been significantly extended by the provision of a turbine model.

10 Bibliography

Ainsworth, A. (2007, August 21). *English Wikipedia.* Retrieved May 21, 2010, from http://en.wikipedia.org/wiki/File:Rankine_cycle_Ts.png.

E.ON. (n.d.). *http://www.eon.com/en/businessareas/35214.jsp*. Retrieved July 2, 2011

Grote, W. (2009). *Ein Beitrag zur modellbasierten Regelung von Entahmedampfturbinen.* Ruhr-Universitäat Bochum.

GRS. (2009). *AHTLET Code Manuals Mod 2.2 Cycle A.* Gesellschaft für Anlagen- und Reaktorsicherheit mbH.

Kleinedler, M. (2002). Heft 28 Aufbau von Kernkraftwerken (DWR). *KWS - Fachhefte für den Kraftwerksbetrieb, zweite Auflage*. Verlag technisch-wissenschaftlicher Schriften.

Lehrstuhl für Energiesysteme . (2010, Oktober 20). Skript der Vorlesung Dampfturbinen.

Plavšić, D. (2008). *Methodolical Investigation into Turbine Modelling for LWR System Codes.* ISaR GmbH.

Ray, A. (1980, April). Dynamic modelling of power plant turbines for controller design. *Applied Mathematical Modelling, Vol. 4*, pp. 109-112.

Siemens AG Bereich Energieerzeugung. (n.d.). *Kundenschulung.*

Stodola, A. (1922). *Dampf- und Gasturbinen, 5. Auflage.* Berlin: Springer.

Strauß, K. (2006). *Kraftwerkstechnik: zur Nutzung fossiler, nuklearer und regenerativer Energiequellen, 5. Auflage.* Berlin: Springer-Verlag.

Traupel, W. (2001). *Thermische Turbomaschinen, 4. Auflage.* Berlin/Heidelberg/New York: Springer-Verlag.

Zimmer, G. (2008, Dezember). Modelling and simulation of steam turbine processes: individual models for individual tasks. *Mathematical and Computer Modelling of Dynamical Systems, Vol. 14, No. 6*, pp. 469-493.