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## Balancing and Data Reconciliation of a steam turbine with auxiliaries

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## Summary

1. The present report describes the modeling and data reconciliation of a steam turbine with auxiliaries (alternator, condensers and high and low pressure heaters).
2. The model is based on a detailed mass and energy balance of main process units and the thermodynamic model based on the 2<sup>nd</sup> Law. This model was complemented by user defined equations defining KPIs.
3. The following KPIs were calculated:
  - Thermodynamic (isentropic) efficiencies of high pressure, intermediate pressure and low pressure turbine sections
  - Fouling of heat transfer areas of all heat exchangers (heaters and condensers) in the form of HTC (Heat Transfer Coefficients)
  - Detailed mass and heat balance of the system including heat duties of all heat transfers
  - Quality (wetness) of steam leaving the low pressure turbine section.
4. The following results of calculation were discussed:
  - Precision (uncertainty) of turbine's section efficiencies and possible improvement in this direction (key measured variables influencing the precision)
  - Precision of other KPIs
  - Possibilities and limitations of gross errors detection.
5. Results of data reconciliation of one data set were complemented by typical results of long term data reconciliation based on industrial data.

# 1. Introduction

Steam turbines are the most common engines which convert thermal energy from pressurized steam to mechanical work. Turbines frequently drive electric generator (alternator). For example, steam turbines provide about 90 % of all electricity generation in the USA (1996).

This report describes modeling, data reconciliation and KPI monitoring of a steam turbine with auxiliaries in the program RECON. There are several important subjects which must be mastered in this direction, for example

- There are a lot of unmeasured streams in the system (uncontrolled extractions, steam parameters, ...) so that a good detailed mass and energy balance model must be set up to observe all important parameters
- The application of the 1<sup>st</sup> Law (mass and energy balancing) is not sufficient for successful optimization of power systems. Essential is the thermodynamic analysis of the system based on modeling in the sense of the 2<sup>nd</sup> Law (entropy, exergy).
- The existence of a wet steam in low pressure parts of the turbine which makes the problem more difficult
- Monitoring heat transfer in a number of heat exchangers
- Modeling the condenser function, which is essential for efficiency of the whole system.

Further will be modeled a typical medium size condensation turbine which is a part of a co-generation system with steam extractions and auxiliaries (feed water heaters, condensate heaters, condenser, gland steam condenser, steam ejectors condenser and alternator).

## 2. A steam turbine with auxiliaries

### 2.1. System description

A steam turbine transforms thermal energy into a mechanical work used to rotate the output shaft. A flowsheet which will be further modeled is shown in the Fig. 2.1.

There is the turbine proper, 2 High Pressure Heaters of the feedwater (HPH), 2 Low Pressure Heaters (LPH) of condensate, condenser of the gland steam and condenser of steam ejectors which heat the turbine condensate. There is also a condenser and an alternator transforming the shaft work in electric power.

The turbine consists of 3 sections:

- High Pressure Turbine (HPT) with pressures 9 MPa – 2.4 MPa
- Intermediate Pressure Turbine (IPT) with pressures 2.4 – 0.6 MPa
- Low Pressure Turbine (LPT) with pressures 0.6 MPa – condenser pressure

IPT and LPT consists of 2 turbine segments each (segments are parts of a section with just 1 extraction stream). IPT has extraction streams 1.4 MPa and 0.6 MPa. LPT has extraction stream 80 kPa. Turbine sections are mounted on 1 shaft which is directly connected with the alternator.

The admission steam enters HPT from the 9 MPa steam collector. Steam leaving HPT (2.4 MPa) splits into 3 streams: (1) the controlled extraction to the 2.4 MPa steam collector STC2.4, (2) the uncontrolled extraction to the High Pressure Heater 1 (HPH1), (3) output going to IPT.

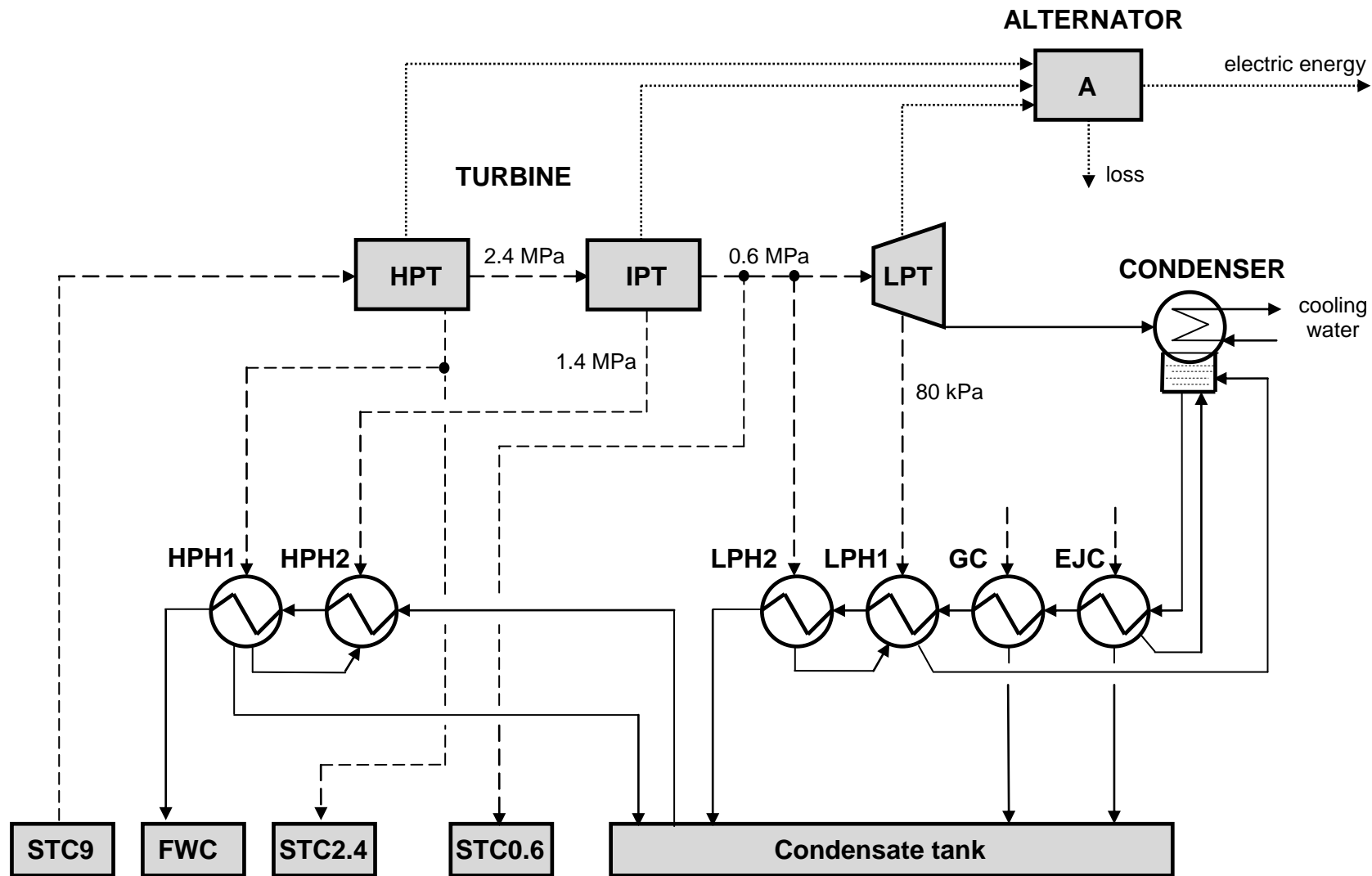
The IPT has 2 segments. The extraction steam 1.4 MPa goes from the first segment to HPH2. The output from the second segment splits into 3 streams: (1) the controlled extraction to the 0.6 MPa steam collector STC0.6, (2) the uncontrolled extraction to the High Pressure Heater 2 (HPH2), (3) output going to LPT.

The LPT has 2 segments. The extraction steam 80 kPa goes from the first segment to LPH1. The output from the second segment goes to the condenser.

Condensate leaving the condenser is heated by the condenser of steam ejectors (EJC), by the condenser of gland steam (GC) and by the above mentioned Low Pressure Heaters. The condenser is cooled by cooling water.

In the upper part of the Fig. 2.1 (dotted lines) there is illustrated the electric energy generation. Streams which represent a mechanical power exerted by the individual turbine parts on the shaft connect the individual turbine sections with the alternator.

There is a certain loss of the mechanical energy in the alternator which is represented by a loss stream. This loss can be either evaluated on the basis of an energy balance of the alternator cooling system or can be taken over from a design documentation.



**Fig. 2.1:** Layout of a turbine with 3 parts and with auxiliaries. STC9 – admission steam collector, FWC – Feed Water Collector, STC2.4 – steam 2.4 MPa collector, STC0.6 – steam 0.6 MPa collector,

There are several problems which can be observed in utilization of turbogenerators. Heat transfer surfaces in the system (high pressure and low pressure heaters and especially condenser) can be fouled. This is typical for condensers due to impurities in cooling waters while the pressure inside condenser influences significantly the efficiency of the electricity generation. Proper monitoring of the condenser efficiency can be used in optimum planning of condenser's cleaning. The other problem connected with condensers generally is their tightness because an air leak into the condenser diminishes the vacuum and thus the condenser's efficiency. This problem can be analyzed by a proper thermodynamic model of the condenser as will be shown later. The efficiency of a turbine proper can be influenced by the quality of turbine's control or by some damage of turbine's blades. The way how to detect such problems is a detailed monitoring of thermodynamic (isentropic) efficiency of all turbine's sections. All such problems should be monitored and solved.

## ***2.2. Targets of monitoring***

The following information is required:

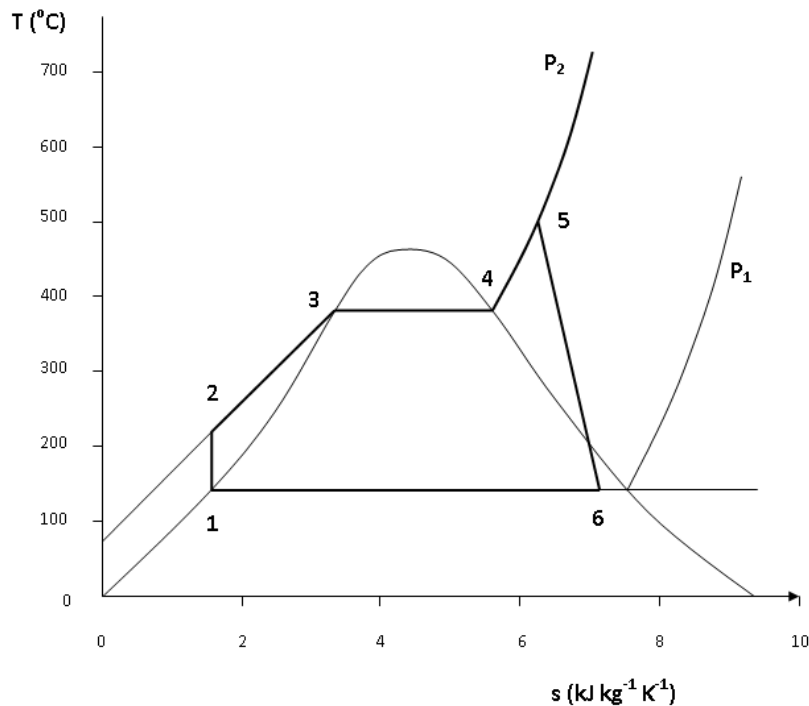
- Detailed mass and energy balance of the system
- Thermodynamic (isentropic) efficiency of all turbine's section (HPT, IPT, and LPT)
- Fouling of heat transfer areas of all main heat exchangers (low pressure and high pressure heaters, and the main condenser) in the form of HTC (Heat Transfer Coefficients)
- Analysis of condenser's function (leak detection)
- Overall efficiency of the whole system.
- Data validation as concerns possible instrumentation errors and malfunction.

## ***2.3. Method of solution***

### **2.3.1 Thermodynamics of steam turbines**

The function of a simple steam turbine is shown in the next Fig. 2.2.





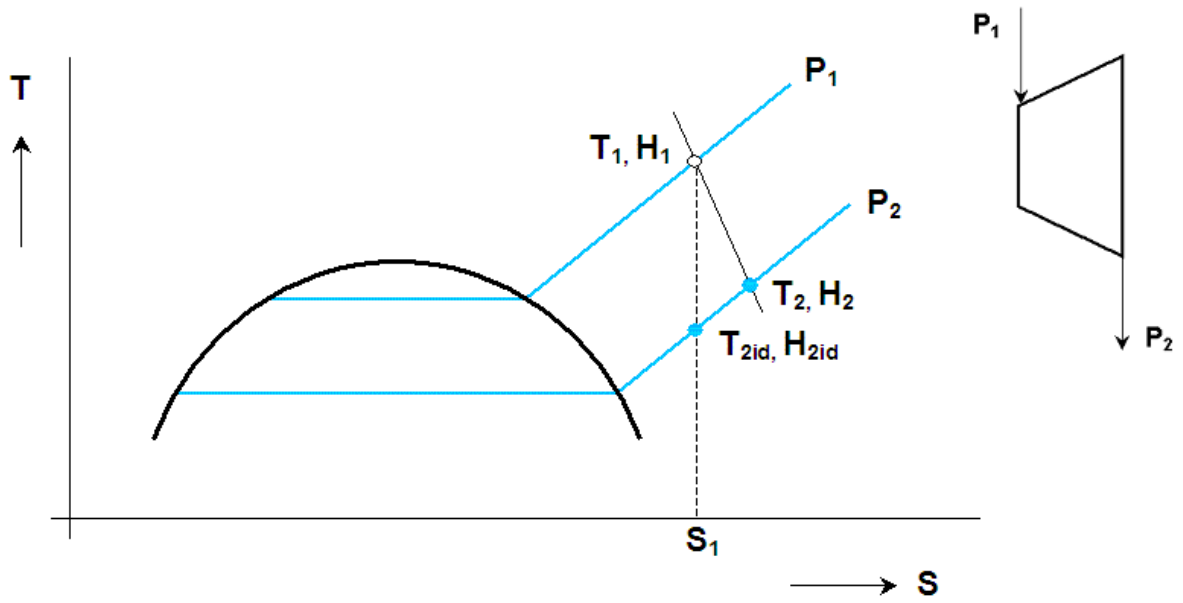
**Fig. 2.2:** A steam cycle in a T-s diagram

There are the following steps in a steam cycle:

- Step 1 – 2: Feed water is pumped from low to high pressure
- Step 2 – 3: Feed water is heated in HP heaters and then in the boiler
- Step 3 – 4: Water is vaporized in the boiler
- Step 4 – 5: Steam is superheated
- Step 5 – 6: Steam expands in a turbine
- Step 6 – 1: Steam condenses in a condenser.

There is a rise of entropy in the step 5 – 6 due to irreversibility of this process under real circumstances.

To measure efficiency of a turbine we compare the real steam expansion in a turbine (irreversible process) with an ideal process with constant entropy (reversible process). The ratio of a real work done in a turbine and the work done in the reversible process is called the *isentropic efficiency*. The calculation of the isentropic efficiency is illustrated in the following *i* – *s* diagram with isobars  $P_1$  and  $P_2$ . The points on isobars are characterized by their temperature *T* and specific enthalpy *H*.



**Fig. 2.3:** Isentropic efficiency of 1 turbine segment in the  $T-s$  diagram, superheated steam region

The real steam expansion in the turbine segment is represented by the abscissa  $T_1, H_1 - T_2, H_2$ . The ideal isentropic process is represented by the abscissa  $T_1, H_1 - T_{2id}, H_{2id}$ .

As the process is adiabatic, there holds a simple energy balance around the turbine segment:

$$A = H_1 - H_2 \quad (2-1)$$

and

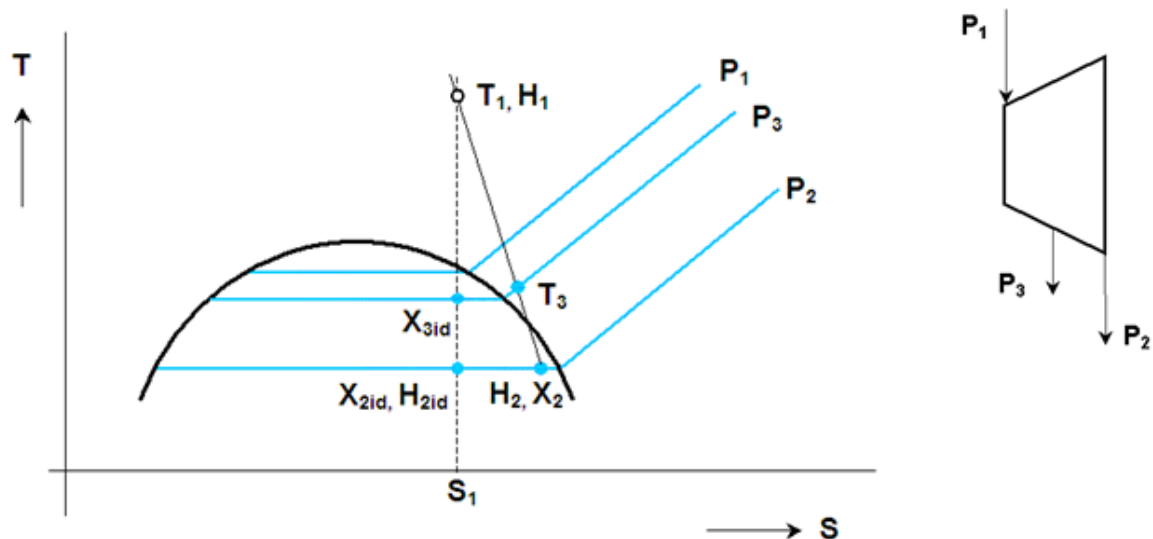
$$A_{id} = H_1 - H_{2id} \quad (2-2)$$

where  $A$  is the mechanical work done by the turbine.

The isentropic efficiency  $\eta$  of a turbine segment is then defined as

$$\eta = \frac{A}{A_{id}} = \frac{H_1 - H_2}{H_1 - H_{2id}} \quad (2-3)$$

Similarly it can be defined efficiency of a turbine section consisting of 2 segments as shown in the next Fig 2.4:



**Fig. 2.4:** Isentropic efficiency of 1 turbine section with 2 segments (1 extraction). Superheated and wet region.

Here can be determined 3 efficiencies: 2 efficiencies for every segment (pressures  $P_1 - P_3$  and  $P_3 - P_2$ ) and one efficiency for the whole section between pressures  $P_1$  and  $P_2$ .

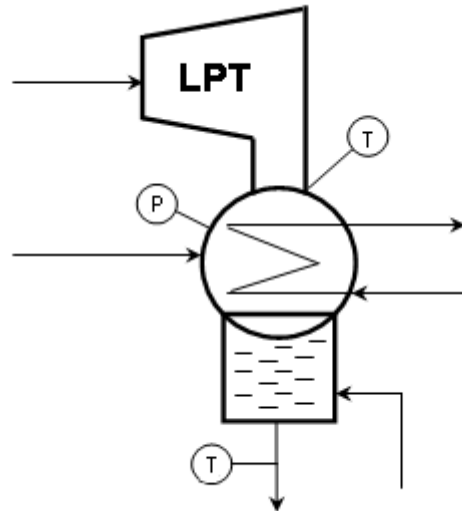
It should be noted that the turbine efficiency determination requires knowledge of the state of the steam in a turbine only, it does not depend on a mass or energy balance. Another important point is the determination of the state of the steam to calculate its enthalpy. In the case of a superheated steam the knowledge of temperature and pressure is sufficient. In a (quite frequent) case of a wet steam the enthalpy is defined by pressure or temperature and also by the steam quality (knowledge of steam content in a mixture with water). As an on-line measurement of steam quality is not common in practice, a detailed mass and energy balance of a whole system can help to observe steam quality.

### 2.3.2 Thermodynamics of condensers

Condensers are inevitable parts of classic power stations. A main target is maintaining high vacuum in a condenser as it has the direct impact on the overall efficiency of the system and on the production of electricity.

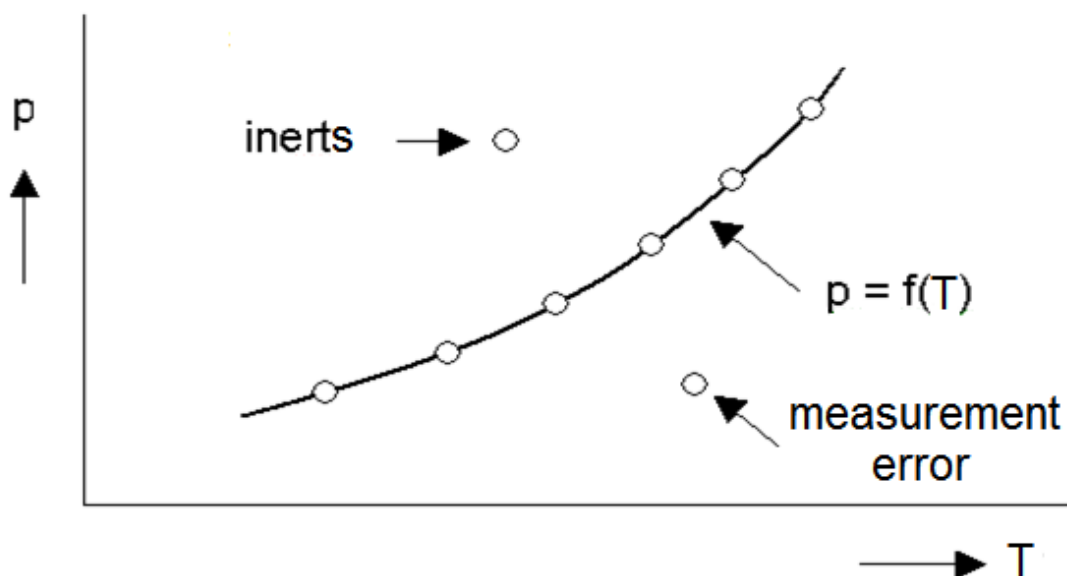
Typical condenser in a power station is a heat exchanger where the exhaust steam coming from the low pressure turbine condenses in the shell side of the exchanger. The cooling water flows in tubes of the exchanger. Large condensers can be separated into several compartments making possible to clean individual compartments during the turbine's operation. On the bottom of the condenser is the condensate collector.

There is usually measurement of pressure in the shell space of a condenser. Also usually measured are temperatures of steam in the inlet of condensers and also the condensate's temperature in the condenser's bottom (see the next Fig. 2.5).



**Fig. 2.5:** A condenser

Experience shows that in a well functioning condenser there is established almost a phase equilibrium between steam and condensate in the shell space of the condenser. This means that the steam temperature at the condenser's inlet is in equilibrium with the pressure inside the condenser (there can be a small systematic difference caused by a kinetic energy of steam which can be easily filtered off). For monitoring of condenser's proper function it is important to monitor difference between the real temperature of steam and the equilibrium temperature belonging to the condenser's pressure (or the real pressure and the equilibrium pressure belonging to the steam temperature). The problem is illustrated in the next Fig. 2.6.



**Fig. 2.6:** Equilibrium T - p relations in a condenser

Normally the measured pressure and temperature lie close to the theoretical equilibrium curve (line of saturation). The higher pressure signals the presence of inert gases which are normally withdrawn by ejectors. A significant leak of ambient air can cause this problem. A pressure value below the equilibrium line signals a measurement error of temperature or pressure.

## 2.4. Modeling the turbogenerator in Recon

### 2.4.1 Basic considerations

The flowsheet of the turbogenerator with accessories in RECON is shown in Fig. 2.7 (compare with Fig. 2.1.). The system of units used is shown in Fig. 2.8.

**Physical units**

<input checked="" type="radio"/> Mass balance		<input type="radio"/> Mole balance		<input type="radio"/> Count balance	
Mass	T	Time	H		
Amount of substance	KMOL	Energy (work, heat)	GJ		
Volume	M^3	Flow-rate	T/H	Energy transfer rate	GJ/H
Count	1				
Concentration	%	First period: Start	08.02.2010 10:00		
Temperature	C	Balancing period	60 min (1 h)		
Pressure	MPaG	Starting hour of the day	0		
Atm.pressure (MPa)	1	Specific energy	J/KG		
Steam wetness	%	Density	KG/M^3		
Length	M	Viscosity	PAS		
		HT Coef.	W/M^2/K		

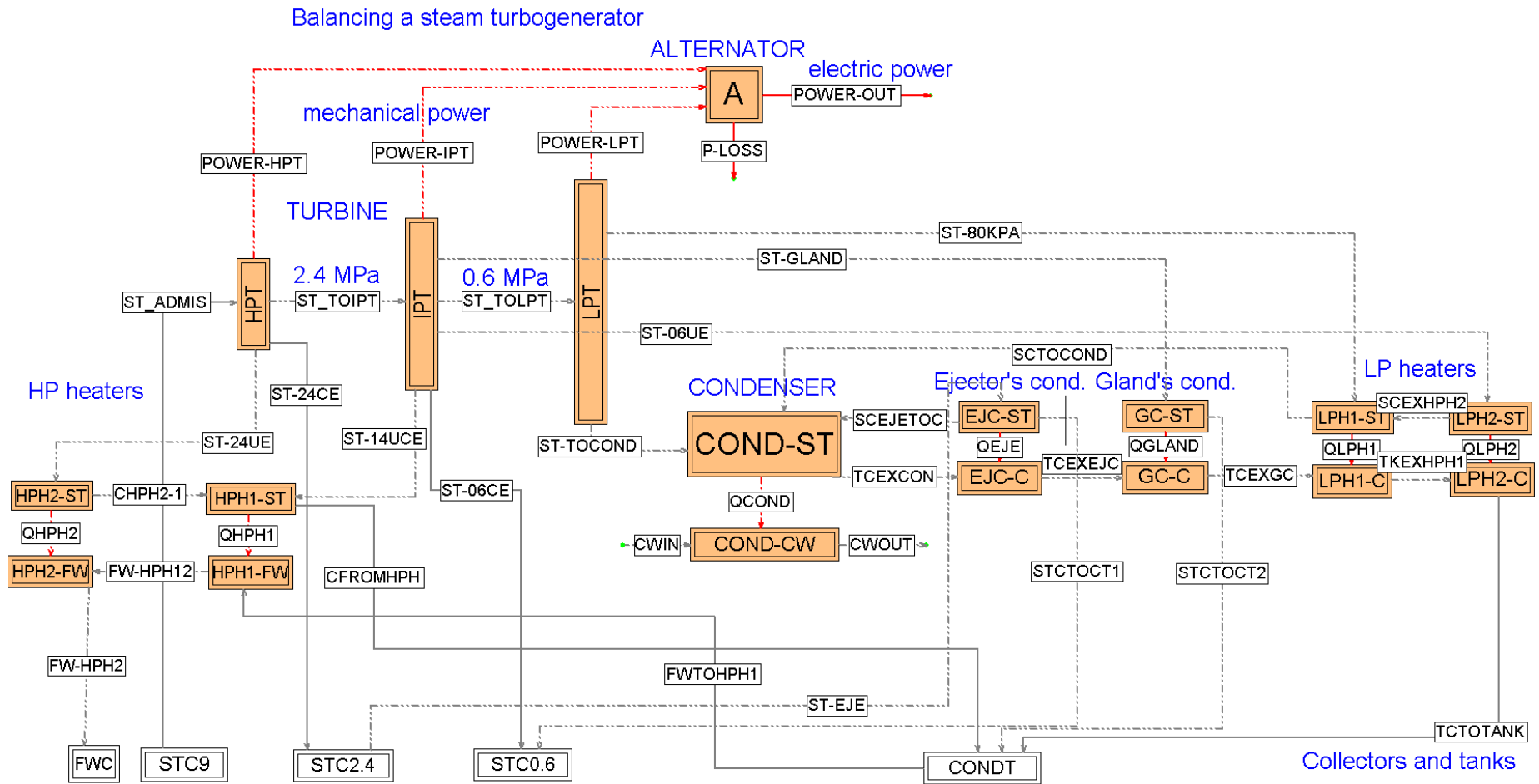
**Fig. 2.8:** Definition of physical units

The base of the balance is mass (t), energy in GJ and time in hours. As in practice the power is measured in MW, measured power must be transformed to GJ/h in the phase of data import. Pressure unit is MPaG. To calculate the absolute pressure, the atmospheric pressure is set to 0.1 MPa. This means that pressure in some parts of the system can be negative.

The only component in the task is H<sub>2</sub>O. All physical properties of water and steam will be modeled by IAPWS IF-97.

### 2.4.2 Modeling turbine sections

The configuration of mass and energy balance for HP section is shown in the next Fig. 2.9.



**Fig. 2.7:** A turbogenerator flowsheet in Recon. Abbreviations: HPT, IPT, LPT – turbine sections, HPH – High Pressure heater, LPH – Low pressure heater, ST – Steam, FW – Feed Water, CW – Cooling Water, C, COND – Condenser, EJC – Ejector's Condenser, GC – Gland steam Condenser, CE – Controlled Extraction, UE – Uncontrolled Extraction

**Node: HPT**

ID	Description
HPT	high pressure turbine

Geodesic height [M] Node pres.

☐ Reaction heat - from database of properties ☐ Invariant balance

Sort of calculations:

- ☒ Balancing
- ☐ Hydraulic node
- ☒ Heat node
- ☐ Reaction node

Non-energy streams incident with node

Stream	Function	Temperature	Pressure	Wetness
IST_ADMIS	H2OV(T,P)	TIAZLH002	PIAHH002	
oST-24CE	H2OV(T,P)	TI005	PI006	
oST-24UE	H2OV(T,P)	TI005	PI006	
oST_TOIPT	H2OV(T,P)	TI005	PI006	

Reactions in node

Reaction

**Fig. 2.9:** Configuration panel of the HP turbine section

Enthalpies of all mass streams are defined by measured temperatures and pressures. It is supposed that all output steam streams have the same temperature and pressure.

A different situation is in the case of the LP section.

**Node: LPT**

ID	Description
LPT	low pressure turbine

Geodesic height [M] Node pres.

☐ Reaction heat - from database of properties ☐ Invariant balance

Sort of calculations:

- ☒ Balancing
- ☒ Hydraulic node
- ☒ Heat node
- ☐ Reaction node

Non-energy streams incident with node

Stream	Function	Temperature	Pressure	Wetness
IST_TOLPT	H2OV(T,P)	TI006	PI009	
oST-80KPA	H2OV(T,P)	ST80KPA	PI010	
oST-TOCOND	H2O(T,X)	TI038		XTOCOND

Reactions in node

Reaction

**Fig. 2.10:** Configuration panel of the LP turbine section

The steam leaving the LP section to the condenser is wet and its quality is not measured. The enthalpy of the stream ST-TOCOND is therefore calculated from the temperature in the condenser and the wetness XTOCOND (fraction of the liquid water in the mixture with steam). The wetness is the unmeasured variable which is observable on the basis of energy balance of the condenser.

## 2.4.3 Modeling heat exchangers

The example of a steam heater configuration is in the Fig. 2.11:

**Node: HPH2-ST**

ID	Description
HPH2-ST	high pressure heater 2 - steam side

Geodesic height [M] Node pres.

☐ Reaction heat - from database of properties ☐ Invariant balance

Sort of calculations:

- ☒ Balancing
- ☐ Hydraulic node
- ☒ Heat node
- ☐ Reaction node

Non-energy streams incident with node

Stream	Function	Temperature	Pressure	Wetness
IST-24UE	H2OV(T,P)	TI005	PI006	
oCHPH2-1	H2O(P,X)		PI006	Xwater

Reactions in node

Reaction

**Fig. 2.11:** Configuration panel of the High Pressure Heater2 – the steam part of the heater

The entering Uncontrolled Extraction steam 2.4 MPA (ST-24UE) is defined by pressure and temperature. In contact with a cold feed water this steam condenses and is withdrawn from the heater. It is supposed that the condensate is saturated water at pressure of the extraction steam.

#### 2.4.4 Turbine efficiencies

Turbine's isentropic efficiencies were discussed in Section 2.3.1, Figures 2.3 and 2.4. The Equation (2-3) requires the knowledge of enthalpy of the steam at the end of the reversible (isentropic) steam expansion in the ideal turbine ( $H_{2id}$  in Fig. 2.3.). This calculation is configured as a *user defined equation* which is a part of the model (*Model supplement*). Possible functions available in Recon are shown in the next Fig. 2.12.

Independent variables		Calculated variables		
Mass flow	Temperature	Stream: Enthalpy	Heat duty	Strapping table
Concentration	Pressure	Stream: Density	Heat transfer	Density table
Compound: Molar weight	Steam wetness	Stream: Molar weight	Sat.steam: Temperature	Viscosity table
Heat flow	Pipe roughness	Stream: Entropy	Sat.steam: Pressure	Auxiliary properties
	Auxiliary		Reaction extent	
User equation				

**Fig. 2.12:** Definition of user defined equations in Recon

The only problem is the temperature of the ideal steam expansion  $T_{2id}$ . This temperature can be found simply by solving the equation (see also Fig. 2.3).

$$S(T_1, P_1) = S(T_{2id}, P_2) \quad (2-4)$$

where  $S(T, P)$  is the specific entropy of steam at temperature  $T$  and pressure  $P$ . In Recon it is written in the implicit form which is created in the Recon's GUI:

$$[WE5<T1:P1>]-[WE5<T2ID:P2>] = 0 \quad (2-5)$$

Here WE5 is the entropy function name in Recon. The unknown temperature  $T_{2id}$  is calculated automatically during the solution of the whole task. The calculation of the efficiency according to Eq. (2-3) with the aid of enthalpy functions (Fig. 2.12) is then straightforward. The calculation of efficiency of the LPT is similar, only the entropy is a function of temperature and steam wetness.

#### 2.4.5 Heat Transfer Coefficients (HTC)

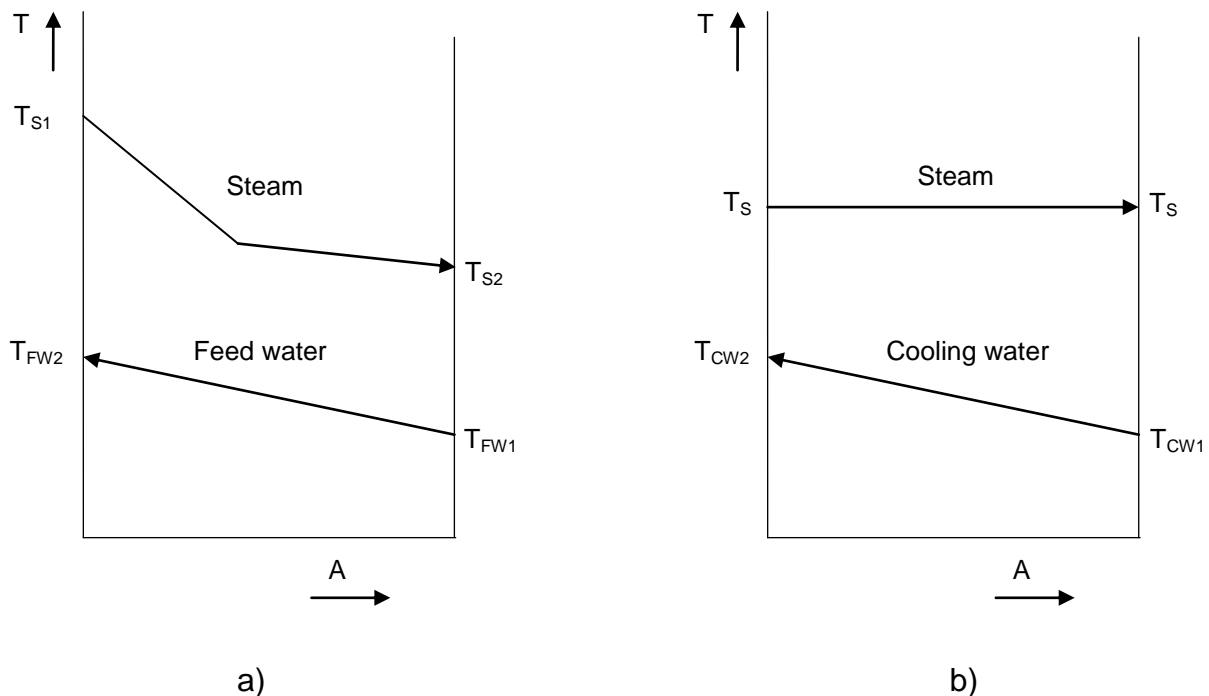
Heat transfer in heat exchangers is usually defined in the form



$$Q = K A dT \quad (2-6)$$

Where       $Q$  = heat flux  
                $K$     HTC  
                $A$     heat transfer area  
                $dT$     mean temperature difference.

One problem is a definition of  $dT$ . An analytical solution for a single phase flow in simple heat exchangers is available but the situation in practice is frequently more difficult. Typical problems can be two phase heat transfer or multi-pass exchangers. Temperature profiles of two types of exchangers are shown in the next Fig. 2.13.



**Fig. 2.13:** Temperature profiles in heat exchangers.  $A$  = heat transfer area.

a) High Pressure Heater    b) Turbine's Condenser

In the case of HPH the steam is superheated. It is cooled to its saturation temperature and then condenses. Some condensate sub-cooling is possible. Situation in the case of a condenser is a little bit simpler. Steam is wet and the temperature on the steam side of exchanger is constant.

The temperature difference  $dT$  can be defined in different ways. In this report is  $dT$  calculated as follows:

- The countercurrent flow is supposed
- The temperature difference is calculated as the arithmetic average of temperature differences at exchanger's end.

For example, in the case of the high pressure heater in Fig. 2.13 the temperature difference is calculated as follows:

$$dT = \frac{(T_{S1} - T_{FW2}) + (T_{S2} - T_{FW1})}{2} \quad (2-7)$$

### 3. Results of data reconciliation

Complete results of calculation are in the Appendix 1 (RECON's output file). It is supposed that the reader is familiar with basics of Data Reconciliation in the extent of report [4]. Here are some excerpts of main results:

The iterative calculation required 3 iterations. The calculation lasted 0.7 second on a standard desktop PC. The main global characteristics are

#### GLOBAL DATA

Quadratic programming	Yes
Number of nodes	23
Number of heat nodes	18
Number of streams	41
Number of energy streams	12
Number of components	1
Number of temperatures	26
Number of pressures	15
Number of wetnesses	4
Number of auxiliaries	24
Number of measured variables	33
Number of adjusted variables	16
Number of non-measured variables	66
Number of observed variables	66
Number of non-observed variables	0
Number of free variables	0
Number of equations (incl. UDE)	68
Number of independent equations	68
Number of user-defined equations (UDE)	33
Degree of redundancy	2
Mean residue of equations	1,8776E-09
Qmin	2,1548E+00
Qcrit	5,9900E+00
Status (Qmin/Qcrit)	0,359738

The redundancy is not very high (only 2) which means that the efficiency of possible gross errors detection will not be high. The Status of data quality (see the last row) is well below the critical value 1, so that no gross error was detected.

There are several important KPIs which belong to auxiliary variables. Selected KPIs are presented in the next table:

#### AUXILIARIES

Name	Type	Inp.value	Rec.value	Abs.error
EFFIC_HP	NO	71,078	70,904	2,059
EFFIC_IP	NO	67,053	67,040	1,830
EFFIC_LP	NO	66,902	67,884	7,440
HTCCOND	NO	1234,712	1229,755	153,633
HTCHPH1W	NO	1815,418	1814,834	208,658
HTCLPH1W	NO	1444,467	1443,837	66,783
HTCNT02W	NO	416,930	416,785	23,068
HTCVT02W	NO	1866,474	1866,815	223,025

(NO = Nonmeasured Observable variable, Inp.value = first guess, Abs.error = uncertainty)

The efficiencies of the individual turbine's sections differ in their precision (column *Abs.error* which represents their uncertainties). HP and IP sections have uncertainties close to 2 per cents (absolutely) while the LP efficiency is over 7 abs. per cents (about 11 relative per cents). This problem is probably related to problems with determination of wetness of the steam leaving the LP turbine section which is based on relatively complex balance of the LP turbine, condenser and LP heaters. For comparison, here is the wetness of the steam leaving the LP turbine section:

W E T N E S S E S

Name	Type	Inp.value	Rec.value	Abs.error
XTOCOND	NO	6,000	6,429	2,464 %

The wetness of steam entering the condenser is observable, but its uncertainty is relatively high (38 % of the value observed). The heat transfer coefficients (HTC) are estimated with tolerance about  $\pm 5 - 12$  % (column *Abs.error*.)

We can ask a question, which variables are influencing the uncertainty of turbine efficiencies. The next table presents shares of important variables on the variance of the HP turbine section (the vector of shares):

Type Variable  
-----  
V EFFIC\_HP

THE VARIANCE OF GIVEN VARIABLE IS CAUSED MAINLY BY:

Type	Variable	Share
P	PI006	7 %
P	PIAHH002	10 %
T	TI005	30 %
T	TIAZLH002	54 %
Sum		100 %

Legend:  
P Pressure  
T Temperature

The vector of Shares informs how much the individual primary variables influence the Efficiency's variance. It is clear that there are 2 variables which together represent 84 % of the Efficiency's variance:

Temperature TIAZLH002 (admission steam)  
Temperature TI005 (steam at the HP turbine outlet)

These variables represent the bottleneck of Efficiency's precision. These instruments should be therefore carefully maintained. It should be noted that the uncertainties of temperatures were set as 3 °C for TIAZLH002 and 2 °C for TI005.

Another question is "How well are results protected against gross measurement errors?". This depends on redundancy of measurements. The best measure of redundancy is the adjustability of redundant variables. Adjustabilities greater than 0.1 are shown in the next table (for variables with adjustabilities lesser than 0.1 gross errors will probably remain undetected). The column *Threshold value* shows how large must a gross error be to be detected with probability 90 %. For example, the error of the admission steam (variable MF ST\_ADMIS) must be at least 6.808 T/HR to be detected with probability 90 %.

# REDUNDANT MEASUREMENTS

Type	Variable	Adjustability	Threshold value	Unit
MF	ST_ADMIS	0,518215	6,808	T/H
T	TI063	0,203804	6,000	C
T	TI072	0,211627	5,901	C

Legend: MF - mass flow; T - temperature.

In the following part of this Chapter will be presented typical long term trends of some variables gathered with the aid of the turbogenerator model described above. On the next Fig. 3.1 is a trend of the Heat Transfer Coefficient of the condenser.

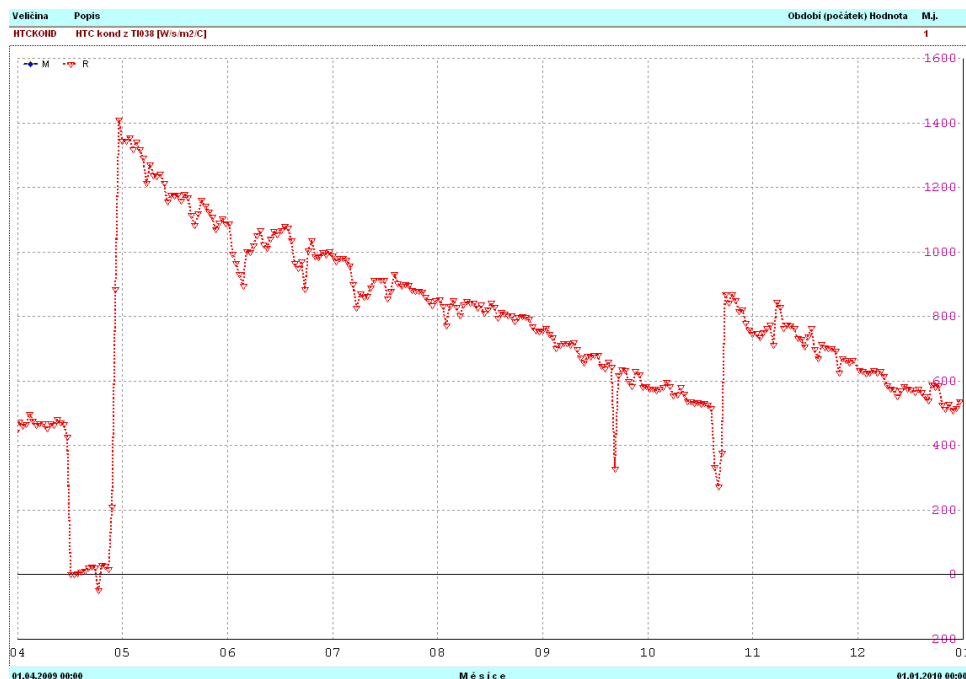


Fig. 3.1: Trend of the condenser's HTC

The time interval shown in the figure is 9 months (the horizontal axis). After the cleaning of the condenser at the end of April the HTC value was about 1400 W/s/m<sup>2</sup>/°C. After 5 months of operation in October the HTC fell down to a value of about 500 W/s/m<sup>2</sup>/°C due to fouling on the cooling water side. After that one half of the condenser's area was cleaned again. After that the HTC value increased to about 850 W/s/m<sup>2</sup>/°C and the fouling continued. The monitoring of condenser's fouling can be used for optimization of cleaning cycles of condenser's tubes.

In the next Fig. 3.2 is the dependence of measured pressure in the condenser P and the measured temperature T at the steam inlet in the condenser (compare with Fig. 2.6).

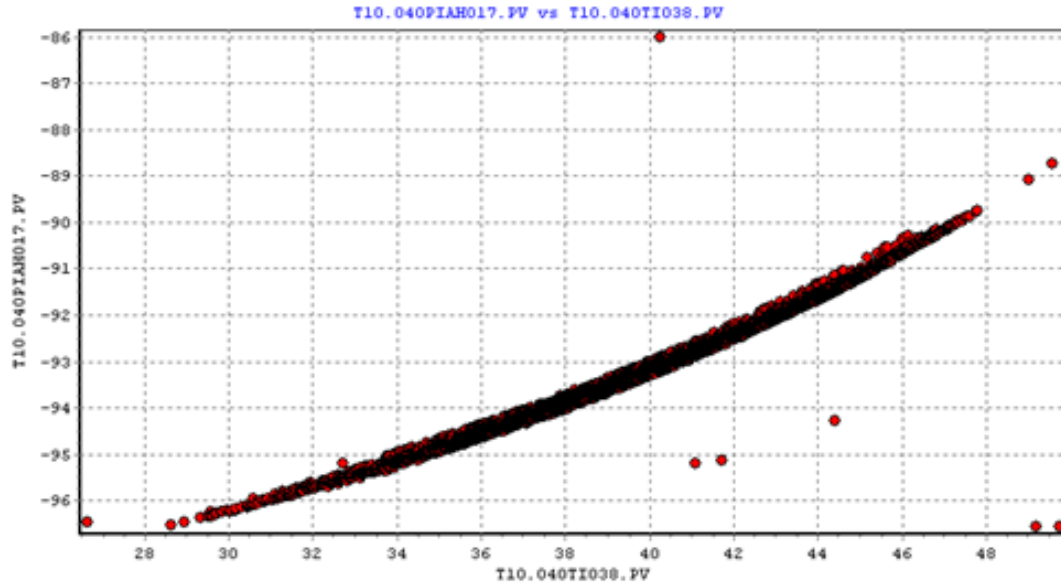


Fig. 3.2: Measured pressure (PIAH017) and temperature (TI038) in the condenser.

Every point in the graph represents 1 hour average, the overall time interval is about 1 year (8345 hours). After deleting 6 outliers the equilibrium pressure on the basis of measured temperature TI038 was calculated. The comparison of measured pressure and the equilibrium pressure is shown in the next Fig. 3.3.

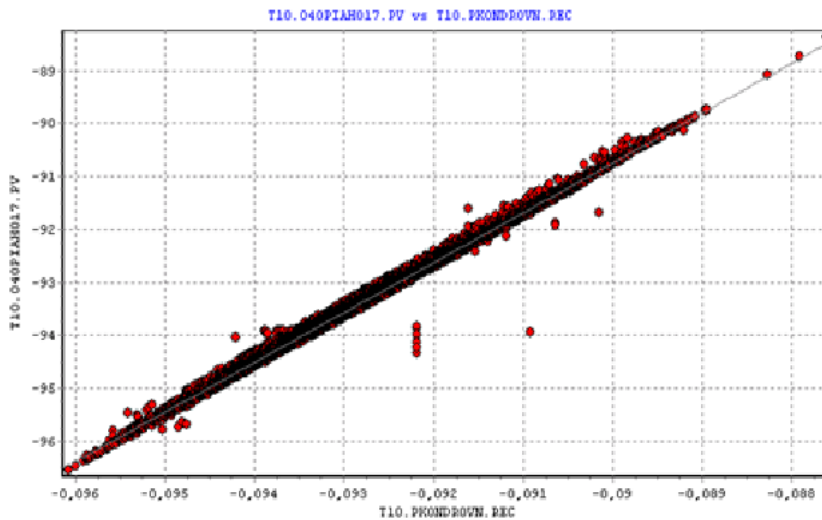


Fig.3.3: Comparison of the equilibrium pressure (horizontal axis) and the measured pressure (vertical axis)

The correlation coefficient between the two pressures is 0.985. It is clear that this condenser is tight. There are only several outlying points which can be probably explained by other reasons than a leak which has usually a systematic character.

In the Fig. 3.4 there are trends of efficiencies of HP sections of 5 identical turbines.

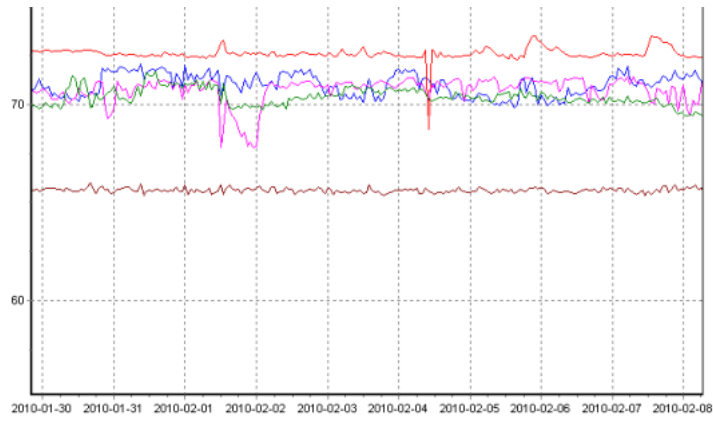


Fig. 3.4: Trends of isentropic efficiencies of HP turbines [%]

This information can serve for diagnostics of turbines and also for optimum exploitation of turbine systems.

## Literature

- [1] EN 60953-2: Rules for steam turbine thermal acceptance tests – Part 2: Method B – Wide range of accuracy for various types and sizes of turbines. idt IEC 953-2:2090.
- [2] IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam. International Association for the Properties of Water and Steam. ASME PRESS, NY 1998
- [3] API: API Data Book. 7<sup>th</sup> edition. API 2005
- [4] Process data validation in practice. Applications from chemical, oil, mineral and power industries. Report CPT 229-07. Usti nad Labem 2007



## Appendix 1: RECON's output file

RECON 11.1.2-Pro [ChemPlant Technology]

Task: TURBINE (Detailed balance TG03)

Balance: [08.02.2010 10:00; 08.02.2010 11:00)

### I T E R A T I O N S

Iter	Qeq	Qx	Qy	Qmin	
START	2,0157E+04				
1	1,0746E+01	9,3210E+00	1,5777E+03	2,1496E+00	
2	4,5740E-04	1,6749E-02	6,5251E+00	2,1549E+00	
3	6,5306E-10	1,1876E-07	1,9192E-05	2,1549E+00	
4	1,7823E-03	6,9711E-02	1,0366E+01	2,1548E+00	SQP
5	1,8776E-09	1,6518E-06	7,8355E-04	2,1548E+00	

Legend:

Qeq mean residual of equations

Qx mean increment of measured variables in iteration

Qy mean increment of non-measured variables in iteration

Qmin least-square function

### G L O B A L D A T A

Quadratic programming Yes

Number of nodes 23

Number of heat nodes 18

Number of streams 41

Number of energy streams 12

Number of components 1

Number of temperatures 26

Number of pressures 15

Number of wetnesses 4

Number of auxiliaries 24

Number of measured variables 33

Number of adjusted variables 16

Number of non-measured variables 66

Number of observed variables 66

Number of non-observed variables 0

Number of free variables 0

Number of equations (incl. UDE) 68

Number of independent equations 68

Number of user-defined equations (UDE) 33

Degree of redundancy 2

Mean residue of equations 1,8776E-09

Qmin 2,1548E+00

Qcrit 5,9900E+00

Status (Qmin/Qcrit) 0,359738

### S T R E A M S

Name	Type	Inp.value	Rec.value	Abs.error	
CFROMHPH	MC	20,802	20,872	0,388	T/H
CHPH2-1	NO	9,266	9,270	0,957	T/H
CWIN	NO	2627,321	2617,290	566,089	T/H
CWOUT	NO	2627,321	2617,290	566,089	T/H
FW-HPH12	NO	186,598	186,615	3,510	T/H
FW-HPH2	NO	186,598	186,615	3,510	T/H
FWTOHPH1	MC	187,436	186,615	3,510	T/H
SCEJETOC	NO	0,193	0,193	0,048	T/H
SCEXHPH2	NO	1,893	1,892	0,101	T/H
SCTOCOND	NO	4,614	4,612	0,127	T/H
ST-06CE	MC	70,470	70,237	1,306	T/H
ST-06UE	NO	1,893	1,892	0,101	T/H
ST-14UCE	NO	11,606	11,602	0,961	T/H
ST-24CE	MC	37,996	37,929	0,744	T/H
ST-24UE	NO	9,266	9,270	0,957	T/H
ST-80KPA	NO	2,721	2,720	0,113	T/H
ST-EJE	NO	0,386	0,386	0,095	T/H
ST-GLAND	NO	0,869	0,869	0,084	T/H
ST-TOCOND	NO	31,099	31,086	0,619	T/H
STCTOCT1	NO	0,193	0,193	0,048	T/H
STCTOCT2	NO	0,869	0,869	0,084	T/H
ST_ADMIS	MC	164,339	165,605	1,584	T/H
ST_TOIPT	NO	118,451	118,406	1,724	T/H
ST_TOLPT	NO	33,820	33,806	0,672	T/H
TCEXCON	NO	35,906	35,891	0,705	T/H
TCEXEJC	NO	35,906	35,891	0,705	T/H
TCEXGC	NO	35,906	35,891	0,705	T/H
TCTOTANK	MC	35,953	35,891	0,705	T/H
TKEXHPH1	NO	35,906	35,891	0,705	T/H

#### E N E R G Y   S T R E A M S

Name	Type	Inp.value	Rec.value	Abs.error	
POWER-HPT	NO	47,405	47,252	1,526	GJ/H
POWER-IPT	NO	24,932	24,916	0,824	GJ/H
POWER-LOSS	MN	2,113	2,113	0,211	GJ/H
POWER-LPT	NO	17,792	17,988	1,807	GJ/H
POWER-OUT	MN	88,043	88,043	0,880	GJ/H
QCOND	NO	73,070	72,791	2,727	GJ/H
QEJE	NO	0,862	0,862	0,213	GJ/H
QGLAND	NO	2,251	2,249	0,216	GJ/H
QHPH1	NO	26,333	26,326	2,002	GJ/H
QHPH2	NO	20,545	20,553	2,121	GJ/H
QLPH1	NO	6,747	6,744	0,251	GJ/H
QLPH2	NO	4,297	4,296	0,230	GJ/H

#### T E M P E R A T U R E S

Name	Type	Inp.value	Rec.value	Abs.error	
CWVIN	MN	11,442	11,442	1,000	C
CWVOUT	MN	18,082	18,082	1,000	C
EQUILCOND	NO	23,774	23,774	5,477	C
EQUILHPH1	NO	197,269	197,269	0,440	C
EQUILHPH2	NO	221,024	221,023	0,503	C
EQUILLPH1	NO	91,833	91,833	0,088	C
EQUILLPH2	NO	159,776	159,776	0,329	C
GLANDC	F	400,000	400,000		C

ST0.6MPAID	NO	194,037	194,037	2,028	C
ST1.4MPA	MC	292,000	292,619	9,950	C
ST2.4MPA	F	374,000	374,000		C
ST2.4MPAID	NO	314,937	314,665	3,016	C
ST80KPA	MN	100,200	100,200	1,000	C
TI005	MC	365,713	365,731	2,000	C
TI006	MN	244,277	244,277	2,000	C
TI038	MN	25,038	25,038	1,000	C
TI043	MC	22,689	22,685	1,000	C
TI044	MC	28,420	28,430	1,000	C
TI045	MC	43,443	43,436	1,000	C
TI046	MN	116,704	116,704	1,000	C
TI047	MN	169,683	169,683	2,000	C
TI062	MC	193,416	193,365	1,998	C
TI063	MC	218,853	218,100	1,592	C
TI064	MN	88,332	88,332	1,000	C
TI072	MC	160,111	160,878	1,577	C
TIAZLH002	MN	525,696	525,696	3,000	C

#### P R E S S U R E S

Name	Type	Inp.value	Rec.value	Abs.error	
ATM	F	-5,55E-18	-5,55E-18		MPAG
CONDEQUIL	NO	-0,097	-0,097	1,89E-4	MPAG
CONDLPH	F	1,000	1,000		MPAG
CW	F	0,180	0,180		MPAG
FW	F	14,200	14,200		MPAG
PI006	MC	2,265	2,265	0,023	MPAG
PI009	MN	0,515	0,515	5,15E-3	MPAG
PI010	MN	-0,025	-0,025	2,48E-4	MPAG
PI011	MN	0,227	0,227	2,27E-3	MPAG
PI012	MC	1,368	1,368	0,014	MPAG
PI013	MN	2,255	2,255	0,023	MPAG
PIAH017	MN	-0,097	-0,097	9,71E-4	MPAG
PIAHH002	MN	8,985	8,985	0,090	MPAG
ST0.6MPA	F	0,540	0,540		MPAG
ST2.4MPA	F	2,300	2,300		MPAG

#### W E T N E S S E S

Name	Type	Inp.value	Rec.value	Abs.error	
XDOKONDID	NO	17,193	17,191	0,218	%
XTOCOND	NO	6,000	6,429	2,464	%
Xsteam	F	0,00E+0	0,00E+0		%
Xwater	F	100,000	100,000		%

#### A U X I L I A R I E S

Name	Type	Inp.value	Rec.value	Abs.error	
DP-COND	NO	-0,231	-0,232	0,989	
DTNTO2	NO	57,258	57,258	0,780	
DTVTO2	NO	15,288	15,291	1,355	
EFFIC_HP	NO	71,078	70,904	2,059	
EFFIC_IP	NO	67,053	67,040	1,830	
EFFIC_LP	NO	66,902	67,884	7,440	
ENT-0.6MPA	NO	2945,024	2945,067	4,230	

ENT-0.6MPAID	NO	2836,697	2836,698	4,385
ENT-2.4MPA	NO	3165,488	3165,489	4,528
ENT-2.4MPAID	NO	3049,040	3048,402	6,840
ENT-9MPA	NO	3451,666	3450,819	7,543
ENT-NT-OUT	NO	2397,636	2389,641	60,123
ENT-NT-OUTID	NO	2126,831	2126,870	7,310
HTCCOND	NO	1234,712	1229,755	153,633
HTCHPH1W	NO	1815,418	1814,834	208,658
HTCLPH1W	NO	1444,467	1443,837	66,783
HTCNT02W	NO	416,930	416,785	23,068
HTCVT02W	NO	1866,474	1866,815	223,025
MARGIN	NO	679,898	679,976	3,414
NIAH052	F	24,456	24,456	
PCONDABS	NO	2,945	2,945	0,971
TDCOND	NO	10,274	10,276	1,225
TDHPH1	NO	20,146	20,147	1,362
TDLPH1	NO	25,948	25,949	0,712

End of results

Calculations lasted	00:00:0.669
Report created	05.09.2013 08:54:46