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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 2nd 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 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 1st 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 2nd 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 1extraction 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.

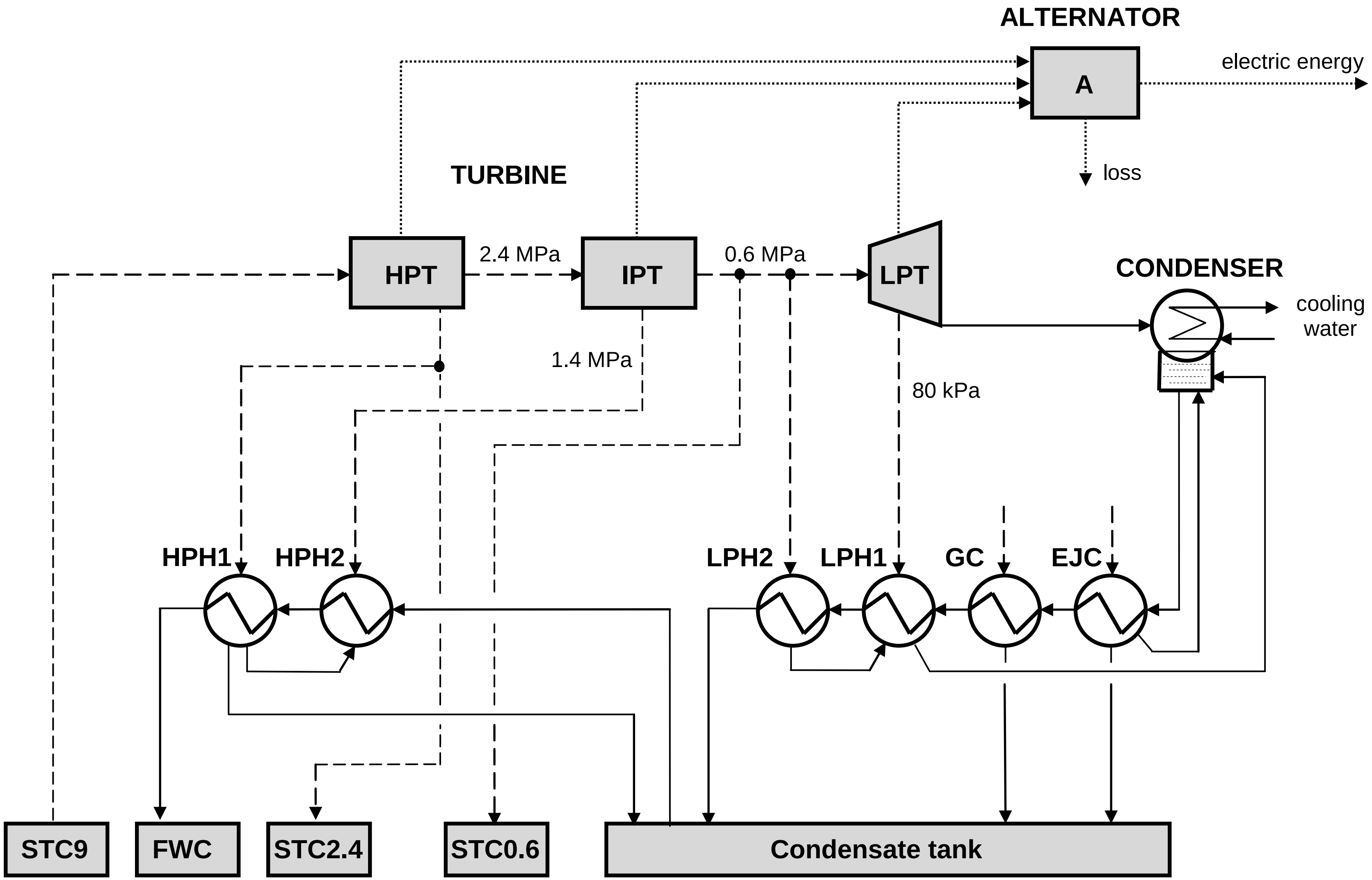

Fig. 2.1: Layout of a turbine with 3 parts and with auxiliaries. STC9 – admission steam collector, FWC – Feed Water Collector, STC2.4 – steam 2.4 MPa collector, STC0.6 – steam 0.6 MPa collector,

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.

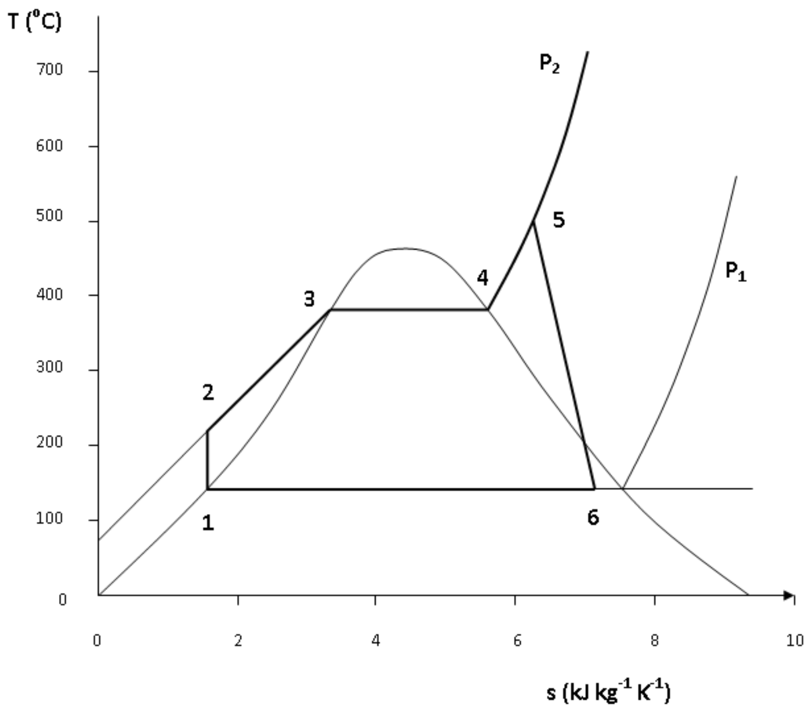

Fig. 2.2: A steam cycle in a T-s diagram

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.

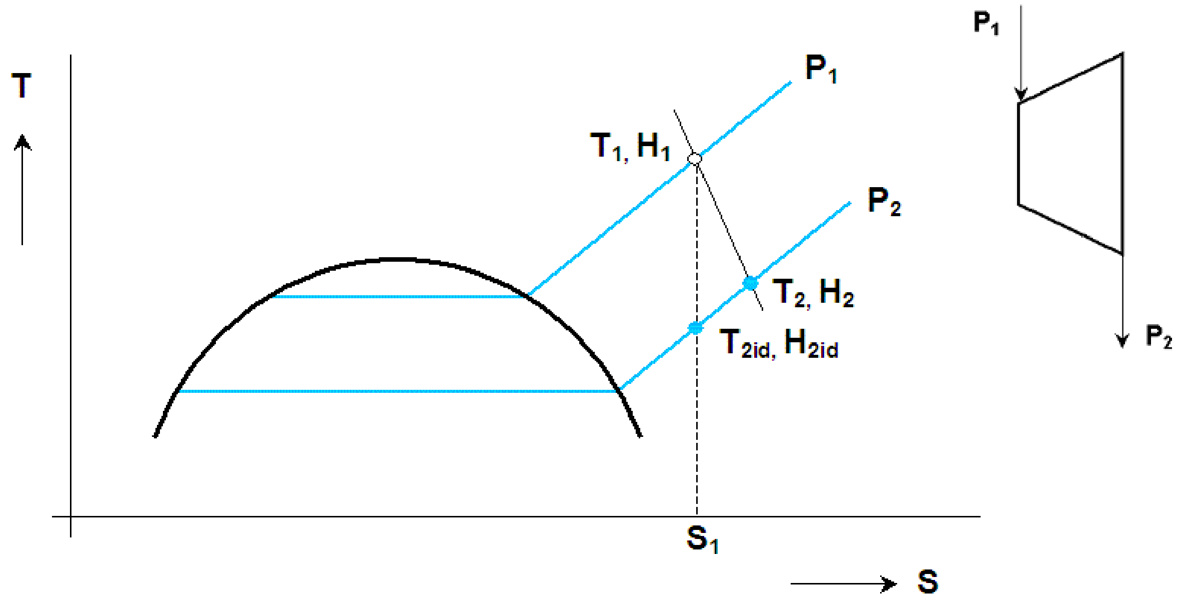

Fig. 2.3: Isentropic efficiency of 1 turbine segment in the *T – s* diagram, superheated steam region

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:

$$A = H_1 - H_2 \qquad (2\text{-}1)$$

and

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

where A is the mechanical work done by the turbine.

The isentropic efficiency η of a turbine segment is then defined as

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

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

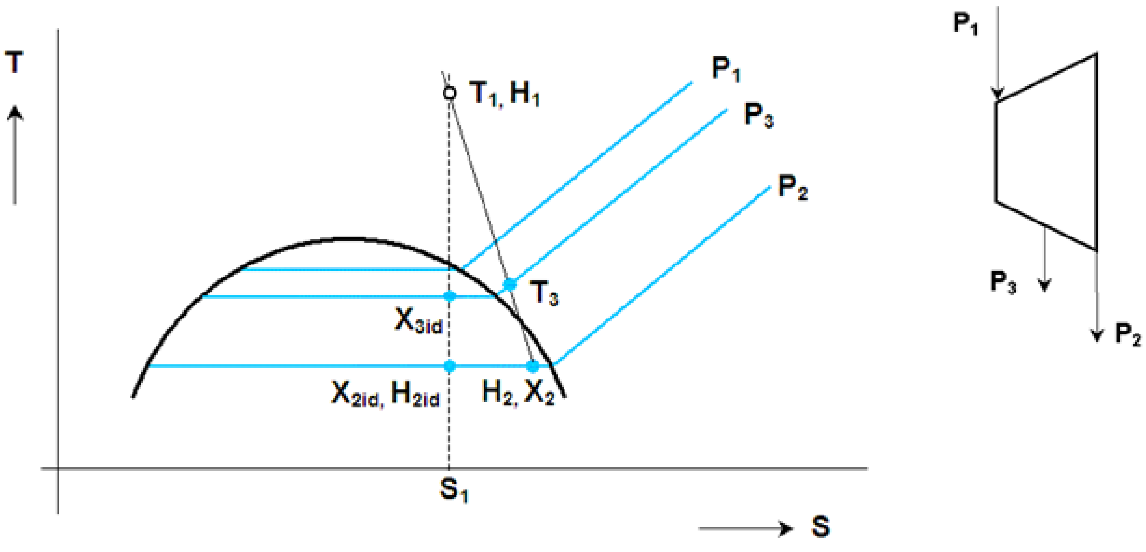

Fig. 2.4: Isentropic efficiency of 1 turbine section with 2 segments (1 extraction). Superheated and wet region.

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).

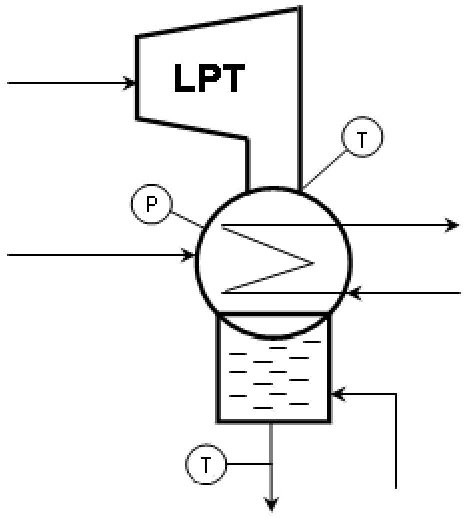

Fig. 2.5: A condenser

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.

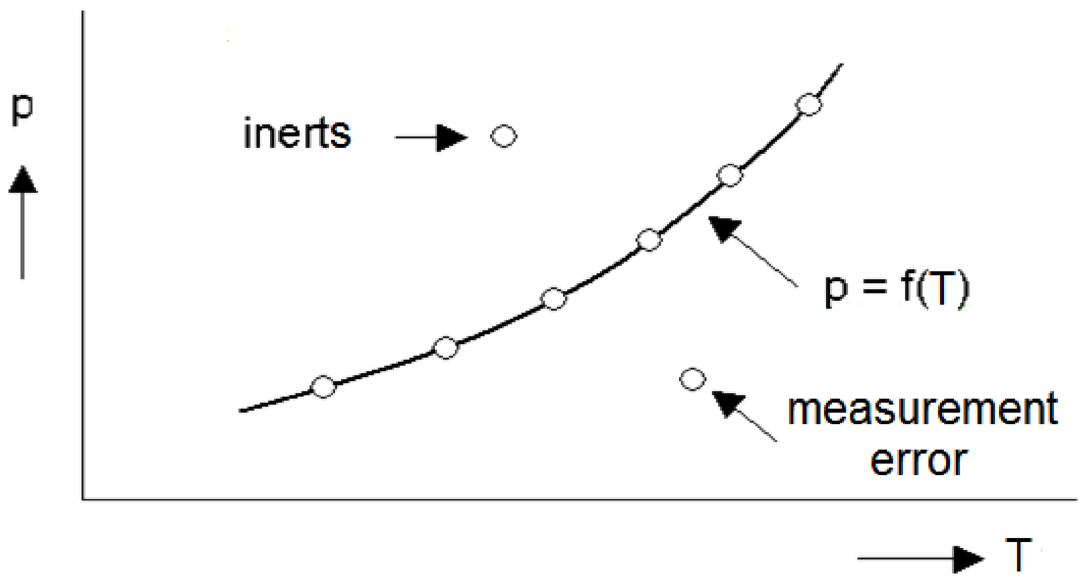

Fig. 2.6: Equilibrium T - p relations in a condenser

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.

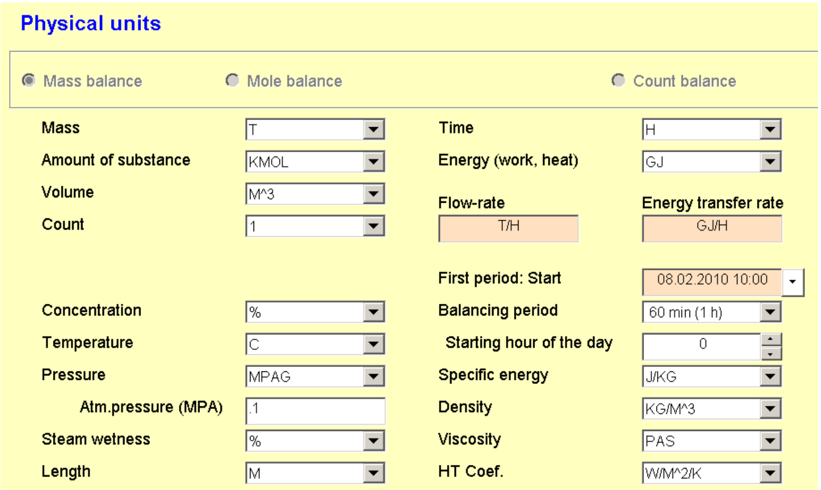

Fig. 2.8: Definition of physical units

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_2O. 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.

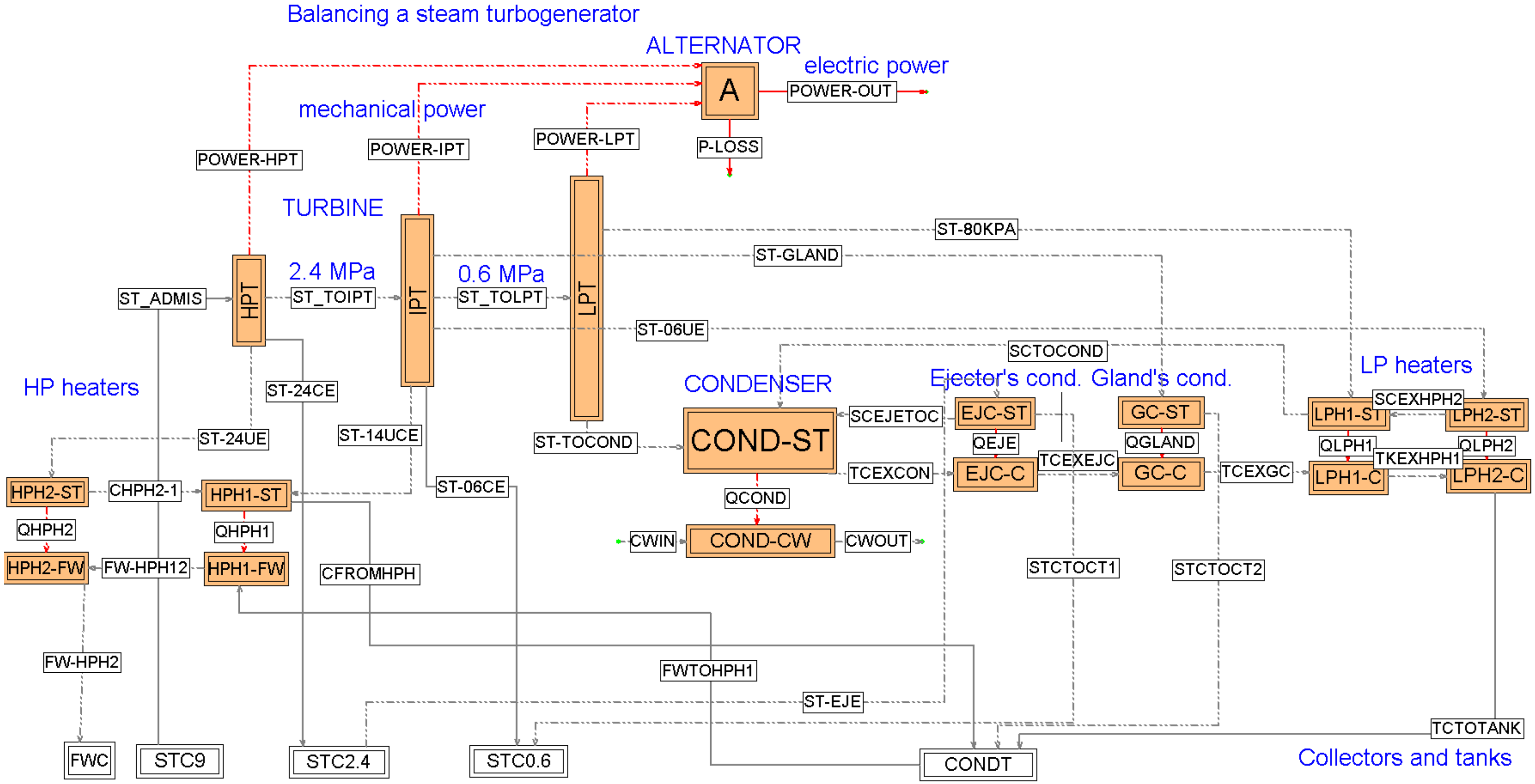

Fig. 2.7: A turbogenerator flowsheet in Recon. Abbreviations: HPT, IPT, LPT – turbine sections, HPH – High Pressure heater, LPH – Low pressure heater, ST – Steam, FW – Feed Water, CW – Cooling Water, C, COND – Condenser, EJC – Ejector's Condenser, GC – Gland steam Condenser, CE – Controlled Extraction,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

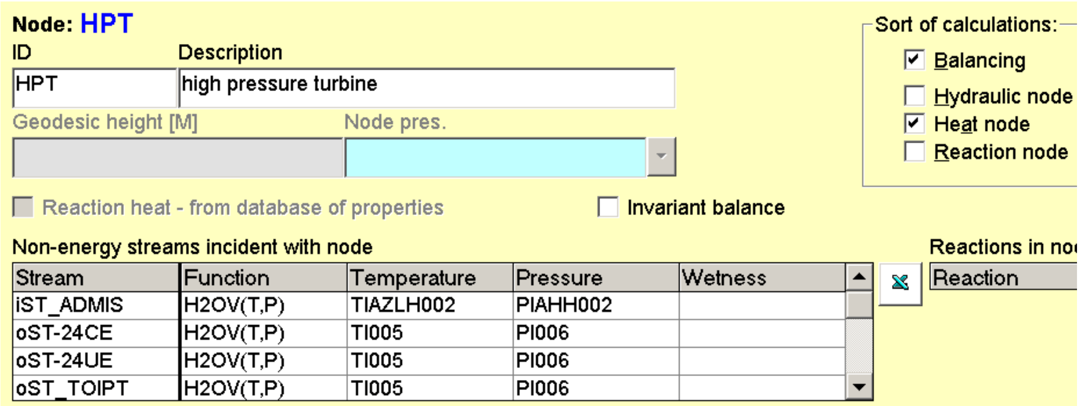

Fig. 2.9: Configuration panel of the HP turbine section

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.

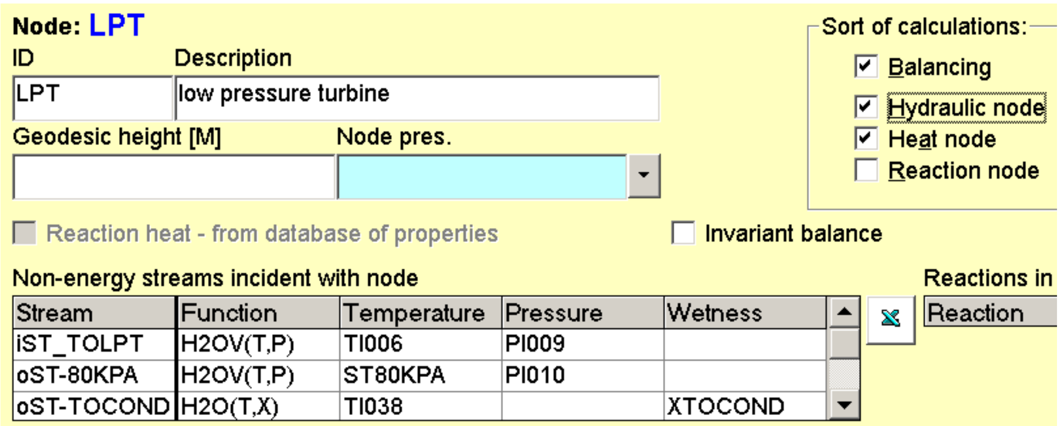

Fig. 2.10: Configuration panel of the LP turbine section

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:

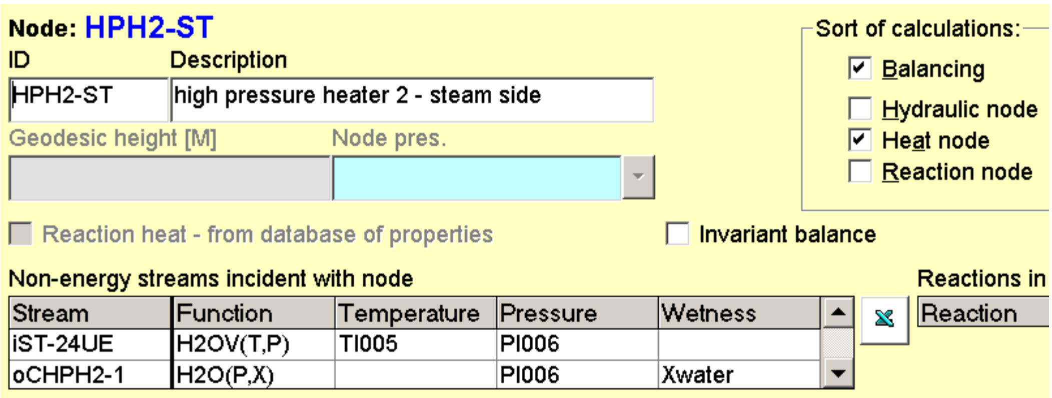

Fig. 2.11: Configuration panel of the High Pressure Heater2 – the steam part of the heater

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.

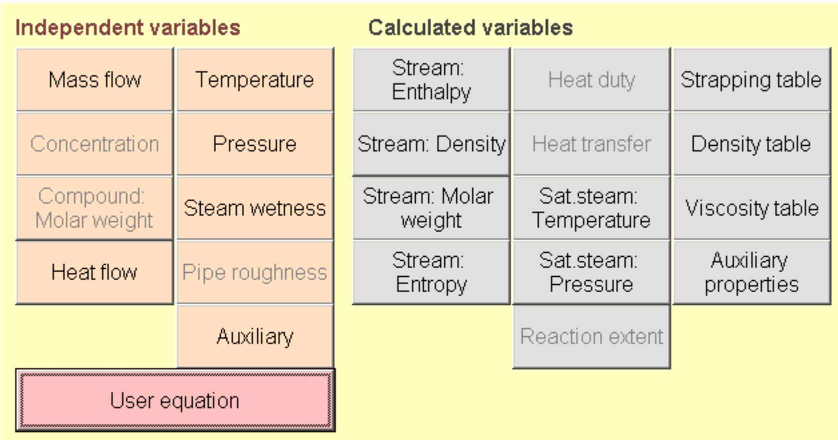

Fig. 2.12: Definition of user defined equations in Recon

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).

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

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 (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

$$Q = K A dT \tag{2-6}$$

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.

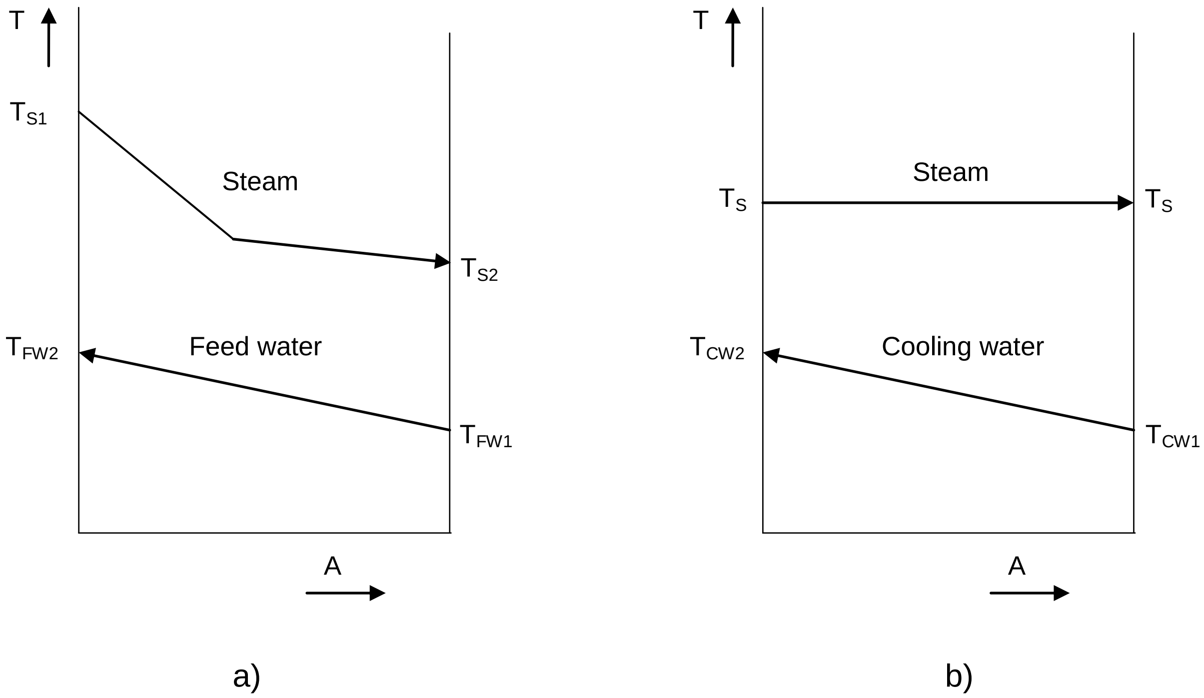

Fig. 2.13: Temperature profiles in heat exchangers. A = heat transfer area.
a) High Pressure Heater b) Turbine's Condenser

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:

$$\mathrm{dT} = \frac{(\mathrm{T}_{\mathrm{S1}} - \mathrm{T}_{\mathrm{FW2}}) + (\mathrm{T}_{\mathrm{S2}} - \mathrm{T}_{\mathrm{FW1}})}{2} \tag{2-7}$$

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

```
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
```

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:

A U X I L I A R I E S

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
HTCNTO2W	NO	416,930	416,785	23,068
HTCVTO2W	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 ± 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 ^{o}C for TIAZLH002 and 2 ^{o}C for TI005.

Another question is the "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 %.

R E D U N D A N T   M E A S U R E M E N T S

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.

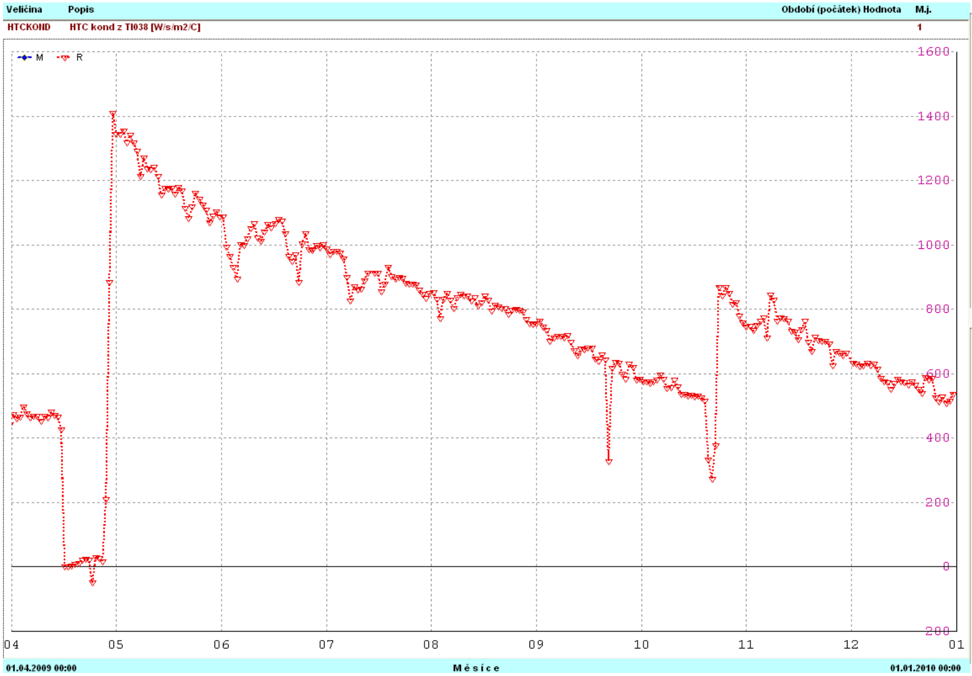

Fig. 3.1: Trend of the condenser's HTC

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^2/^{o}C. After 5 months of operation in October the HTC fell down to a value of about 500 W/s/m^2/^{o}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^2/^{o}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).

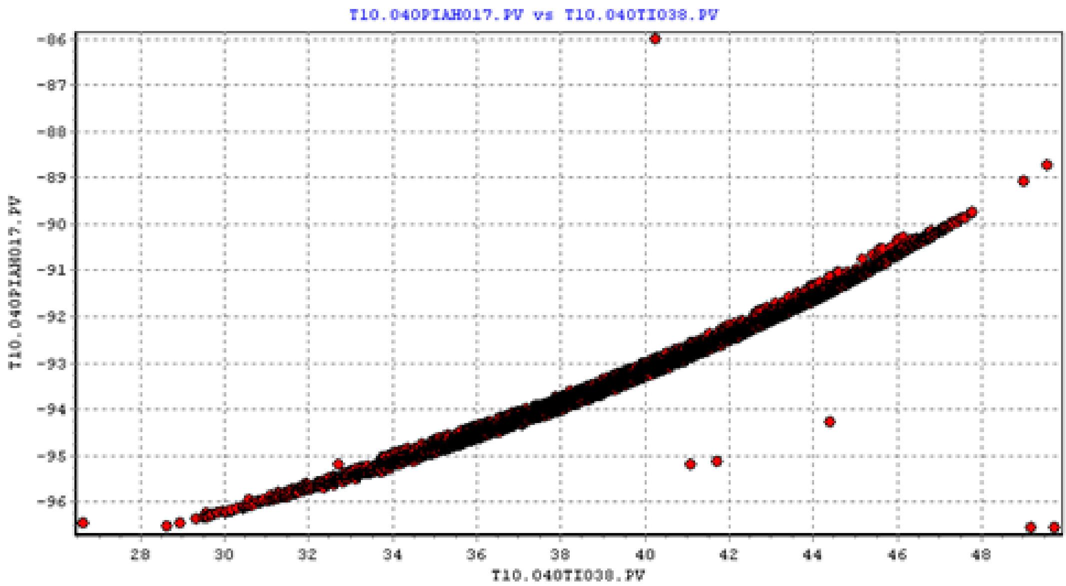

Fig. 3.2: Measured pressure (PIAH017) and temperature (TI038) in the condenser.

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.

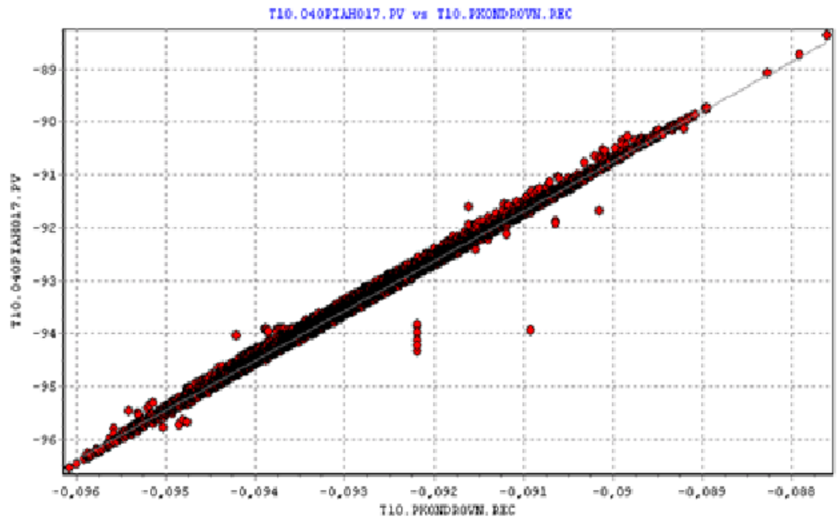

Fig.3.3: Comparison of the equilibrium pressure (horizontal axis) and the measured pressure (vertical axis)

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.

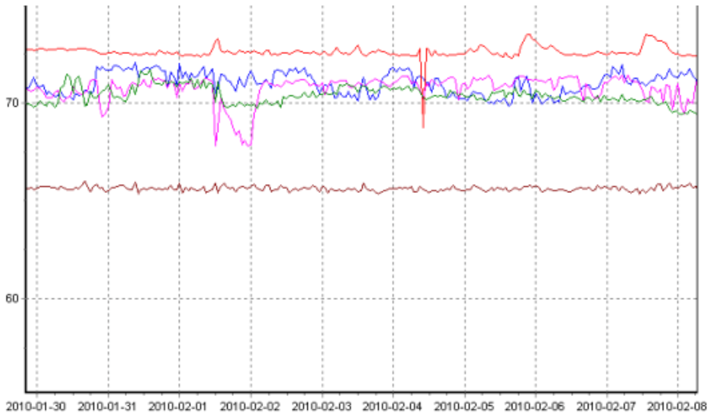

Fig. 3.4: Trends of isentropic efficiencies of HP turbines [%]

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. 7th 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
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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
HTCNTO2W	NO	416,930	416,785	23,068
HTCVTO2W	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