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Numerical Simulation of a Three Stage Natural Circulation Heat Recovery Steam Generator

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Abstract: - The paper presents the short description of a computer program, which calculates the dynamic behavior of steam generators, especially natural circulation boilers. The so called "tube-header-model" is used to describe the connections between a header and a higher number of tubes. For solving the partial differential equations of the conservation laws for the working fluid the two finite volume algorithms SIMPLER and PISO are used. The computer code is suitable for the calculation of a boiler with a different number of pressure stages. To show the suitability of the program for the dynamic simulation of a boiler the computational results of a hot start-up and shut down will be presented for a vertical type natural circulation heat recovery steam generator with 3 pressure stages.

Key-Words: Numerical simulation, natural circulation, heat recovery steam generator, hot start-up, shut down

1 Introduction

The calculation of the static behavior of steam generators is a standard procedure at the design of new boilers. Many computer codes, e. g. NOWA, even developed at the Institute for Thermodynamics and Energy Conversion (ITE), have been written in the past to calculate the steady state behavior of the boilers. As result of such calculations the mass flow distribution of the working medium in the tube network as well as other detailed informations e. g. density, pressure and temperature are available. But a steady state analysis can only find a solution for the mass flow distribution in the tube network of the boiler under steady state conditions. So it is not possible to predict the mass flow distribution under e. g. start up conditions. The definition of the operating conditions should be verified by a dynamic analysis.

At the Institute for Thermodynamic and Energy Conversion a computer program DBS (Dynamic Boiler Simulation) was developed for the calculation of the dynamic behavior of steam generators, especially natural circulation HRSG's. In the following sections a short description of the theoretical background of the program and the results derived from the application of this program on a three stage natural circulation HRSG will be presented.

2 Description of the simulation model

2.1 Generals about the simulation program

At the design of steam generators the header is used for the connections between the internal or external connecting conduits with large tube diameter and a large number of heating surface tubes with a small tube diameter. To get a good representation of the boiler structure in the model a so called "tube-header-model" was developed at ITE to describe the transient mass flow distribution in the tube network of the steam generator [1]. The discretization of the partial differential equations of the conservation laws was done in the past with the aid of the finite volume method SIMPLER [2]. The simulation of different boilers has shown that the SIMPLER algorithm has a robust but a slow convergence rate with the increasing number of control volumes. To decrease the simulation time four different finite volume methods are analysed and the results are presented in [4]. Based on the tube-header-model this investigation has shown that PISO is the algorithm with the fastest convergence rate. But PISO is not as stable as the SIMPLER algorithm. As a result of the different convergence rate and stability PISO is additionally included in the newest release of DBS as second algorithm. Dependent on the convergence rate of the simulation the program switches during the calculation between the two algorithms.

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Modern HRSG's are designed as more pressure systems. Under consideration of this fact, DBS was extended in such a way that there is no pressure stage restriction implemented.
A further detailed analysis of the convergence rate has also shown that the convergence rate of the DBS can be improved if the whole discretized boiler with his different pressure levels will not be represented in a single matrix. The calculation area of the boiler is subdivided into the different pressure stages. Every pressure stage is also subdivided into the four calculation areas feed water, evaporator, superheater and drum. This results in a superior iteration for every pressure stage. With this concept every part of the boiler needs a minimum of iteration steps for convergence. Only the flue gas side must be solved for the whole boiler. This concept of splitting of the calculation area is also implemented in the new release of DBS. A further advantage of this concept is the possibility to use different models for the fluid flow during the calculation e.g. homogeneous equilibrium model for the feed water or superheater and a drift flux model for the evaporator. Such a drift flux model will be implemented in the future.
To get a small band structure for the system of equations, the band width of the coefficient matrix will be optimized at the generation of the calculation data. The administration of all data is done by the use of the graph theory. This allows a simple and detailed description of the tube network.

2.2 Model for the working medium

The mathematical model for the working medium in the tubes is one-dimensional in flow direction and uses the homogeneous equilibrium model for the two phase flow. For the pressure loss of the two phase flow in a single tube the correlation according to Friedel [5] is used.
The heat exchange between fluid and wall is governed by Newton's law of cooling and the heat transfer through the wall is assumed to be in radial direction only. The heat transfer model used in DBS for the single and two phase flow of the working medium includes correlations for horizontal as well vertical tubes and is described in detail in [1].

2.3 Model of the header

At the design of steam generators the header is used for the connections between the internal or external connecting conduits with large tube diameter and a large number of heating surface tubes with a small tube diameter (see Fig. 1).

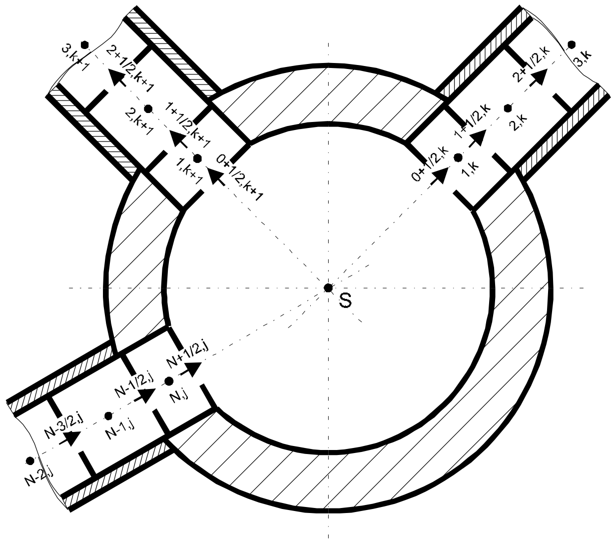

Fig. 1: Model of the header and the connected tubes

Figure 1 shows a header and the connected tubes. The three connected tubes are representative for all the other tubes which are connected to the header. Index *j* denotes the tubes where the mass flow enters the header, while the index *k* indicates the tubes where the mass flow leaves the header. The index *S* denotes the thermodynamic state variables of the header.
Assuming that the distribution of the thermodynamic state of the header is homogeneous, the header can be assumed to be one single point for the calculation. This assumption is justified due to the small vertical dimension of the header, compared to the height of the remaining tube system. Therefore the gravity distribution of density and pressure can be neglected. The huge difference of the cross section area between the header and the connected tubes causes strong turbulence, avoiding stratification of the fluid in the header.
Because momentum is a vector with the direction of the tube axis, the momentum fluxes have to be added using the laws of vector analysis. This fact has to be considered especially at locations where tubes are connected under different angles.
The velocity of the fluid in the header is rather small compared to that inside the tubes. So it can be assumed, that the momentum of the fluid is dissipated at the entrance of the header and has to be regenerated at the outlet of it. Based on this assumption, the momentum balance of the header reduces to a pressure balance. The changes of the momentum at the inlet and the outlet can be taken into account using a pressure loss coefficient ζ.

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2.3.1 Balance equations for the header

The proposed algorithm PISO from Issa [3] is a time-marching procedure in which during each time step one predictor step followed by one or more corrector steps. The algorithm will not be discussed here in detail. For further information about the PISO algorithm see [3] and [4].

Taking into consideration the above described tube-header-model the dissipation and regeneration of the momentum leads to a separation of the momentum balance for the header and the connected tubes. At the formulation of the momentum equations for the single control volumes of the tube network the control volume of the header can be neglected. So, the calculation of the velocities w^*, w^{***} and also of the pseudo-velocities $\tilde{w}$ and $\tilde{\tilde{w}}$ can be neglected. The calculation of the header pressure is not independent from the connected tubes and therefore the first correction equation for the pressure must be written in the following form:

$$a_{mPS}\,p_S^* = \sum_j a_{mWS,j}\,p_{N,j}^* + \sum_k a_{mES,k}\,p_{1,k}^* + b_{mS} \quad (1)$$

with the coefficients

$$b_{mS} = \frac{(\rho_S^0 - \rho_S)V_S}{\Delta t} + \sum_j (\rho\tilde{w}A)_{N+1/2,j} - \sum_k (\rho\tilde{w}A)_{0+1/2,k}, \quad (2)$$

$$a_{mWS,j} = (\rho A)_{N+1/2,j}\,d_{wN,j}, \quad (3)$$

$$a_{mES,k} = (\rho A)_{0+1/2,k}\,d_{e1,k} \text{ and} \quad (4)$$

$$a_{mPS,k} = \sum_j a_{mWS,j} + \sum_k a_{mES,k} \quad (5)$$

$$d_{wN,j} = A_{N+1/2,j} / a_{mPS}. \quad (6)$$

The conditional equation for the second pressure correction can be written in the following form:

$$a_{mPS}\,p_S^{**} = \sum_j a_{mWS,j}\,p_{N,j}^{**} + \sum_k a_{mES,k}\,p_{1,k}^{**} + b_{mS} \quad (7)$$

The coefficients for the second pressure correction equation (7) of the header are identical to the equations (2) – (6) by replacing the velocity $\tilde{w}$ through $\tilde{\tilde{w}}$.

The energy balance for calculating the specific enthalpy h_S of the header has the following form:

$$a_{hPS}h_S = \sum_j a_{hWS,j}h_{N,j} + \sum_k a_{hES,k}h_{1,k} + b_{hS} \quad (8)$$

with the coefficients

$$a_{hPS}^0 = \frac{\rho_S^0 V_S}{\Delta t}, \quad (9)$$

$$b_{hS} = S_{hcS}V_S + a_{hPS}^0 h_{PS}^0, \quad (10)$$

$$a_{hWS,j} = [[(\rho w)_{N+1/2,j}, 0]]A_{N+1/2,j}, \quad (11)$$

$$a_{hES,k} = [[-(\rho w)_{0+1/2,k}, 0]]A_{0+1/2,k} \text{ and} \quad (12)$$

$$a_{hPS} = \sum_j a_{hWS,j} + \sum_k a_{hES,k} + a_{hPS}^0 - S_{hPS}V_S. \quad (13)$$

The operator [[A,B]] is equivalent to DMAX1(A,B) in the computer language FORTRAN. The FORTRAN-instruction denotes the greater value of A and B.

2.3.2 Balance equations for the header connected control volumes

In this section, the discretized balance equations for the header connected control volumes will be presented. The control volumes which are connected to the header can be seen in Fig. 1. All equations are independent of the flow direction.

The following equations describes the first control volume at the tube inlet. The momentum equation for the predictor step of the velocity $w_{0+1/2,k}^*$ of the tubes *k*, *k+1*, etc. leads to:

$$a_{e0,k}\,w_{0+1/2,k}^* = a_{ee0,k}\,w_{1+1/2,k}^* + b_{e0,k} + (p_S^n - p_{1,k}^n)A_{0+1/2,k} \quad (14)$$

with the coefficients

$$a_{e0,k}^0 = \frac{\rho_{0+1/2,k}^0 A_{0+1/2,k}\Delta x_{0+1/2,k}}{\Delta t}, \quad (15)$$

$$b_{e0,k} = S_{ee0,k}A_{0+1/2,k}\Delta x_{0+1/2,k} + a_{e0,k}^0 w_{0+1/2,k}^0, \quad (16)$$

$$a_{ee0,k} = [[-(\rho w)_{1+1/2,k}, 0]]A_{1+1/2,k} \text{ and} \quad (17)$$

$$a_{e0,k} = a_{ee0,k} + a_{e0,k}^0 - S_{ep0,k}A_{0+1/2,k}\Delta x_{0+1/2,k}. \quad (18)$$

The proportional part of the source term leads to:

$$S_{ep0,k}\Delta x_{0+1/2,k} = -\frac{|\Delta p_{Fric0+1/2,k}|}{|w_{0+1/2,k}|} - (1+\zeta_{in})\frac{\rho_{0+1/2,k}w_{0+1/2,k}[[w_{0+1/2,k}, 0]]}{2w_{0+1/2,k}}. \quad (19)$$

The calculation of the pseudo-velocity $\tilde{w}$ must be done with:

$$\tilde{w}_{0+1/2,k} = w_{0+1/2,k}^* - d_{e1.k}\left(p_S^n - p_{1,k}^n\right). \quad (20)$$

The first pressure correction equation leads to:

$$a_{mP1,k}\,p_{1,k}^* = a_{mW1,k}\,p_S^* + a_{mE1,k}\,p_{2,k}^* + b_{m1,k} \quad (21)$$

with the coefficients

$$b_{m1,k} = \frac{(\rho_{1,k}^0 - \rho_{1,k})A_{1,k}\Delta x_{1,k}}{\Delta t} + (\rho\tilde{w}A)_{0+1/2,k} - (\rho\tilde{w}A)_{1+1/2,k}, \quad (22)$$

$$a_{mW1,k} = (\rho A)_{0+1/2,k}\,d_{w1,k}, \quad (23)$$

$$a_{mE1,k} = (\rho A)_{1+1/2,k} d_{e1,k} \text{ and} \quad (24)$$

$$a_{mP1,k} = a_{mW1,k} + a_{mE1,k} . \quad (25)$$

With the first explicit step, the correction of the velocity $w^*_{0+1/2,k}$ can be calculated with

$$w^{**}_{0+1/2,k} = \tilde{w}_{0+1/2,k} + d_{e1,k}(p^*_S - p^*_{1,k}) \quad (26)$$

The conditional equation for the second pressure correction can be written in the following form:

$$a_{mP1,k} p^{**}_{1,k} = a_{mW1,k} p^{**}_S + a_{mE1,k} p^{**}_{2,k} + b_{m1,k} \quad (27)$$

The coefficients for the second pressure correction equation (27) of the header connected tube are identical to the equations (22) – (25) by replacing the velocity $\tilde{w}$ through $\tilde{\tilde{w}}$. $\tilde{\tilde{w}}$ is calculated by:

$$\tilde{\tilde{w}}_{0+1/2,k} = w^{**}_{0+1/2,k} - d_{e1,k}(p^*_S - p^*_{1,k}) + \frac{a_{ee1,k}(w^{**}_{1+1/2,k} - w^*_{1+1/2,k})}{a_{e1,k}} . \quad (28)$$

The new velocity is calculated from the second explicit correction equation with the following form:

$$w^{***}_{0+1/2,k} = \tilde{\tilde{w}}_{0+1/2,k} + d_{e1,k}(p^{**}_S - p^{**}_{1,k}) \quad (29)$$

The equation for the specific enthalpy for the header connected control volume leads to:

$$a_{hP1,k} h_{1,k} = a_{hW1,k} h_S + a_{hE1,k} h_{2,k} + b_{h1,k} \quad (30)$$

with the coefficients

$$a^0_{hP1,k} = \frac{\rho^0_{1,k} A_{1,k} \Delta x_{1,k}}{\Delta t}, \quad (31)$$

$$b_{h1,k} = (S_{hc} A \Delta x)_{1,k} + a^0_{hP1,k} h^0_{1,k}, \quad (32)$$

$$a_{hW1,k} = \left\| (\rho w)_{0+1/2,k}, 0 \right\| A_{0+1/2,k}, \quad (33)$$

$$a_{hE1,k} = \left\| -(\rho w)_{1+1/2,k}, 0 \right\| A_{1+1/2,k} \text{ and} \quad (34)$$

$$a_{hP1,k} = a_{hW1,k} + a_{hE1,k} + a^0_{hP1,k} - (S_{hP} A \Delta x)_{1,k} . \quad (35)$$

The equations for a control volume at the tube outlet (subscript N) are not described here in detail. This set of equations can be derived from the equations (14) to (35).

The number of tubes which are connected to a header is not restricted by the model.

2.4 Model for the flue gas

The structure of the flue gas side is - corresponding to the working medium - also very general. The idea was to build up the flue gas pass as a network. It includes mixing and distribution points, inlets and outlets as well as recirculation paths. An individual flue gas pass can be subdivided into a number of parallel passes. This parallel passes are used for the simulation of flue gas passes with different flue gas streams. There is no energy and mass exchange between the flue gas fluxes of the individual passes.

For the description of the flue gas the one-dimensional partial differential equation of the conservation law for the energy is used. The momentum balance for the flue gas is neglected. The flue gas mass flows are calculated quasi-stationary, while the energy balance is calculated unsteady. The discretization of the energy balance is done corresponding to the finite volume method.

The forced convective heat transfer coefficient between the flue gas and the tubes can be calculated under the use of a different number of correlations for plain or finned inline or staggered tube banks.

3 Simulated heat recovery boiler

Figure 2 shows the overall design of the simulated three pressure stage HRSG. The drum pressure at steady state for the high pressure system (HP) is approximately 125 bars, for the intermediate pressure system (IP) approximately 40 bars and for the low pressure system (LP) 13 bars. The HP system consists of four economiser (HPECON1-4), an evaporator (HPEVAP) with 6 parallel tube paths and an superheater (HPSH), the IP system consists of an economiser (IPECON), an evaporator (IPEVAP) with 3 parallel tube paths and a superheater (IPSH). The LP system consists of an economiser (LPECON) and an evaporator with 3 parallel tube paths (LPEVAP). The saturated steam of the LP system is used by another process and will not be superheated. In the simulation model all parallel tube paths are included.

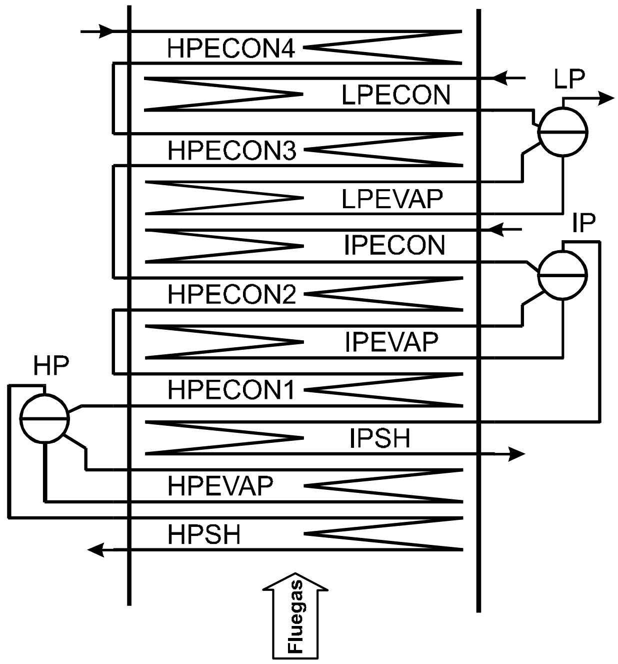

Fig. 2: Design of the HRSG

The exhaust flue gas mass flow and the temperature of the gas turbine, which are input boundary conditions for the simulations, are given as a function of time and can be shown in Fig. 3. The total simulation time was 9000 s.

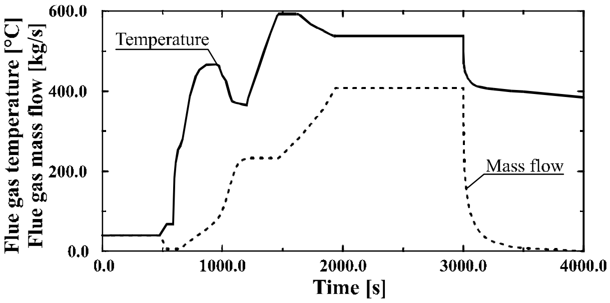

Fig. 3: Mass flow and temperature ramps of the flue gas

During the first period of the simulation (480 s) the gas turbine purges the boiler. After synchronization of the gas turbine (up to 540 s) the gas turbine starts and reaches full load 1950 s after simulation start. During the time period between 1950 s and 3000 s the gas turbine works with the full load operating condition. Following the stationary gas turbine operation the shut down of the gas turbine is simulated between 3000 s and 9000 s. In this period the exhaust mass flow of the gas turbine decreases rapidly to zero, while the exhaust temperature decreases approximately linearly with a small gradient.

3.1 Initial conditions for the simulation

For the simulation of the hot start-up and shut down of the gas turbine the following initial conditions for the boiler are used:

- The steam generator is filled with water near boiling condition.
- The pressure distribution of the working medium in the tube network of the boiler is affected by gravity.
- The velocity of the fluid at the start of the calculation process is equal to zero.
- The initial fluid temperature in the evaporator of the boiler is identical to the boiling temperature at drum pressure.
- The initial drum pressure for the HP-system was 100 bars, for the IP-system 32 bars and for the LP-system 12 bars.
- The water level in the different drums starts at low water level.

4 Results of the numerical simulation

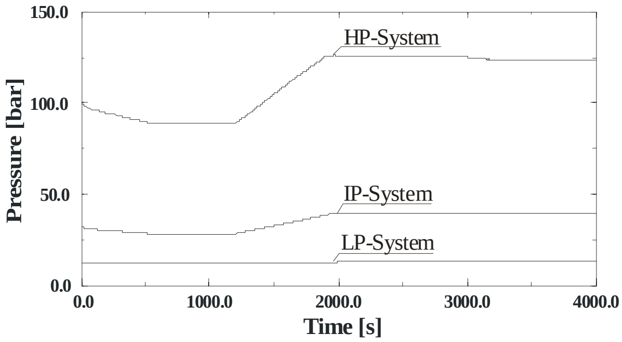

Fig. 4: Drum pressure of the different pressure stages

Figure 4 shows the development of the drum pressure for the different pressure stages of the HRSG during the first 4000 s of the simulation. During the purging process of the HRSG the drum pressure of the HP-system decreases approximately linear to 89 bars, the IP-system to 28.3 bars and the drum pressure of the LP-system is during this process approximately constant. During the first phase of the hot start-up (time between 540 s and 1200 s after simulation start) these drum pressure values are constant followed by an increase of the HP drum pressure with 3 bars/min, the IP drum pressure with 0.88 bars/min and the LP drum pressure with 0.05 bars/min up to the different full load pressures. After achieving the system pressure at full load the drum pressures are approximately constant.

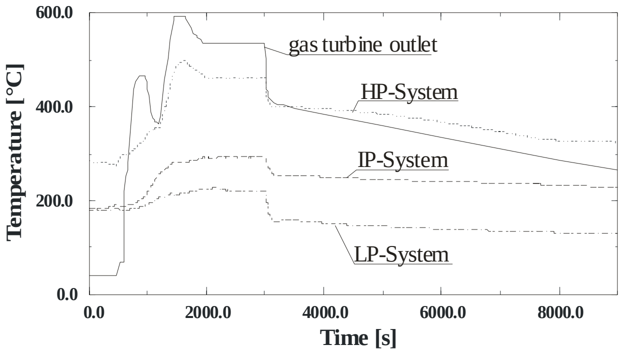

Fig. 5: Flue gas temperature

Figure 5 shows the development of the flue gas temperature at the gas turbine outlet as well as in front of the evaporator of the different pressure stages. It can be seen, that during the purging process and the first period of the hot start-up the flue gas temperature before the evaporators is higher than the gas turbine outlet temperature. Therefore the flue gas is heated up and the tubes are cooled down. This can be seen also at the gas turbine shut down at the HP-system.

Figure 6 to 8 shows the mass flow distribution in selected evaporator tubes of the different pressure stages. In the following only the development in the HPEVAP will be analysed in detail.

During the purging period the tube temperature of the HPSH and the HPEVAP tubes will be cooled down and therefore the drum pressure of the HP-system decreases (see Fig. 4 and 5). As a result of this process the working fluid starts to circulate and a small steam mass flow leaves the drum. This produced steam is condensate in the HPSH. Between the time period of 480 s and 540 s after simulation start the gas turbine outlet mass flow decreases and the flue gas outlet temperature increases and therefore the heat flow from the tubes to the flue gas decreases. With increasing flue gas mass flow and temperature the heat flow direction between flue gas and boiler tubes changes and therefore the mass flow circulation in the evaporator tubes as well as the steam production increases and achieves the steady state at full load.

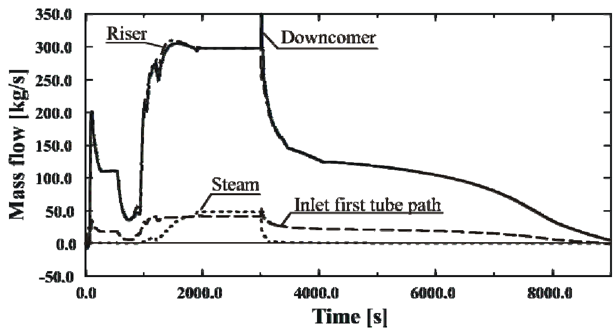

Fig. 6: Mass flow in the tubes of the HPEVAP

With the beginning of the gas turbine shut down the flue gas mass flow decreases rapidly (see Fig. 3). During this period the fluid mass from the drum is stored into the tube network of the evaporator. This can be seen in the increase of the fluid mass flow in the downcomer as well as in the decrease of the water level in the drum (Fig. 6 and Fig. 9). After completion the storing process the mass flow in the tube network of the evaporator decreases slowly to zero.

The description of start-up and shut down process for the HP-system is also valid for the IP- and LP-system. Compared to the HP-system the point of time for the start and the end of the fluid circulation of the IP- and LP-system is shifted in time (see Fig. 6 to 8).

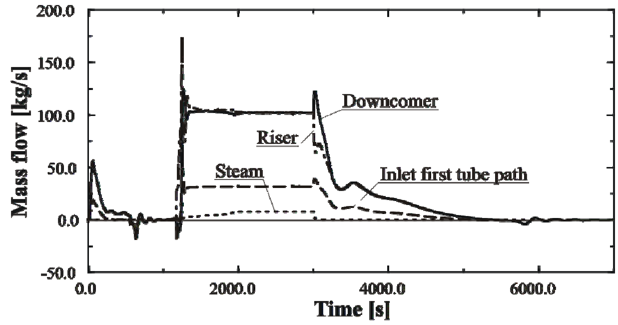

Fig. 7: Mass flow in the tubes of the IPEVAP

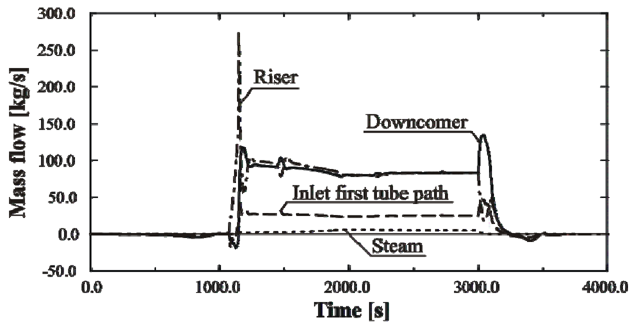

Fig. 8: Mass flow in the tubes of the LPEVAP

The change of the water level for the three pressure stages is shown in Fig. 9. It can be seen that at the beginning of the simulation the water level for all three drums is located at the low water level. Compared to the HP- and IP-system, the LP-system has the highest overshoot of the normal water level. With the beginning of the gas turbine shut down mass will be stored into the tube network of the different systems. This leads to a decrease of the water level. At the end of the shut down process the normal water level at all systems is reached.

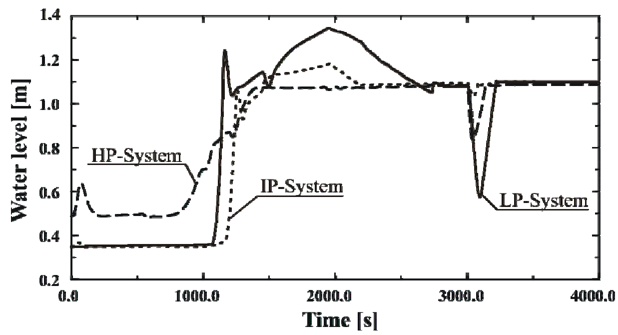

Fig. 9: Water level for the drums of the different pressure stages

5 Conclusion

For the simulation of the fluid flow in a tube network a mathematical model has been developed, which describes the fluid flow in the tubes as well as the changes of state in the connecting elements. The so called tube-header-model presented in this paper is very well suited for the simulation of this problem. The fluid flow in the tube is modelled one-dimensionally by means of the conservation equations for the mass, the momentum and the energy. The headers are modelled as points, whereby the differential equation for the momentum balance is reduced to a pressure balance. The finite volume methods SIMPLER and PISO are used for solving the sparse coefficient matrix for the system. The system of equations representing the momentum balance is decoupled at the headers, so the matrix will remain tridiagonal structured.

Finally results of a warm start-up and shut down of a three stage natural circulation HRSG are presented.

6 Nomenclature

A	Cross sectional area [m^2]
a	Coefficient of PISO [kg/s] or [ms]
b	Constant coefficient of PISO [kg/s]
h	Specific enthalpy [J/kg]
p	Pressure [Pa]
Δp_{Fric}	Pressure loss due to friction [Pa]
S_c	Constant part of the source term
S_p	Prop. part of the source term
Δt	Time step [s]
V	Volume [m^3]
w	Velocity [m/s]
Δx	Length [m]
ρ	Density [kg/m^3]
ζ_{in}	Inlet pressure loss [Pa]

Subscripts

E	East neighbor cell
e	East boundary of a cell
ee	East boundary of a cell at the staggered grid
h	Energy balance
m	Mass balance
S	Header
W	West neighbor cell
w	West boundary of a cell

Superscripts

n	Solution of the last iteration step
0	Value at the old time step
*	Solution of the predictor step
**	Solution of the first corrector step
***	Solution of the second corrector step
~	Pseudo-velocity

References:

[1] Walter, H., *Modeling and Numerical Simulation of Natural Circulation Steam Generators*, Fortschritt-Berichte VDI, Series 6, No.: 457, VDI-Verlag, Düsseldorf, 2001.

[2] Patankar, S. V., *Numerical Heat Transfer and Fluid Flow*, Series in Computational Methods in Mechanics and Thermal Sciences, Hemisphere Publ. Corp., Washington, New York, London, 1980.

[3] Issa, R. I., Solution of the Implicitly Discretised Fluid Flow Equations by Operator-Splitting, *Journal of Computational Physics*, Vol.62, 1985, pp. 40-65

[4] Walter, H. and Weichselbraun, A.: A Comparison of Four Finite-Volume-Algorithm for the Dynamic Simulation of Natural Circulation Steam Generators, Proceedings of the 4th IMACS Symposium on Mathematical Modeling, Vienna, Austria, 5.-7. February 2003, ARGESIM Report No.24, Vol. 2, p. 531-540

[5] Friedel, L.: "Improved Friction Pressure Drop Correlation for Horizontal and Vertical Two-Phase Pipe Flow, *European Two-Phase Group Meeting*, Ispra, Italy, 1979, Paper E 2, pp. 1 - 25.