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DESCRIPTION OF THE POWER PLANT MODEL BWR-PLASIM OUTLINED FOR THE BARSEBÄCK 2 PLANT

by

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Abstract. A description is given of a BWR power plant model outlined for the Barsebäck 2 plant with data placed at our disposal by the Swedish Power Company Sydkraft A/B.

The basic equations are derived and simplifications discussed.

The model is implemented with a simulation system DYSYS which assures reliable solutions and easy programming. Emphasis has been placed on the models versatility and flexibility so new features are easy to incorporate. The model may be used for transient calculations for both normal plant conditions and for abnormal occurrences as well as for control system studies.

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CONTENTS

1. INTRODUCTION

A general model for a BWR power plant, called BWR-PLASIM, has been adopted to the Barsebäck 2 plant as it has been possible to get a complete set of data for that plant. This gives an opportunity to make comparisons between calculations and measurements in cooperation with the Swedish Power Company Sydkraft A/B.

The work has been carried out as a link in the department's efforts to maintain a fundamental knowledge of nuclear power plants and in particular the safety evaluation of transients for many abnormal situations.

The model derivation is based upon several years experience with simulation of nuclear plants and the individual components (Ref. 1 and 2). But until now we have not had the possibility to check any model against real measurements. The experience has been used in the selection of the model structure, in the mathematical description of the single components and in the simulation technique in order to obtain a flexible model that reflects the essential characteristics without superfluous details of specific components. Special attention has been given to the simulation carried out with a general simulation system DYSYS (Ref. 5), that assures reliable solutions, and to the structure of the program which makes modifications easy to perform. The model may not be regarded as a final work but rather as a frame for simulations as new features specific for individual transients are easy to implement.

Some main features of the model are given here and a more detailed description is given in Chapter 2-5.

A diagram of the model is shown in Fig. 1, page 4 where the plant's components may be identified. They are the reactor, the turbine connected via the steam line, the feedwater system and three control systems.

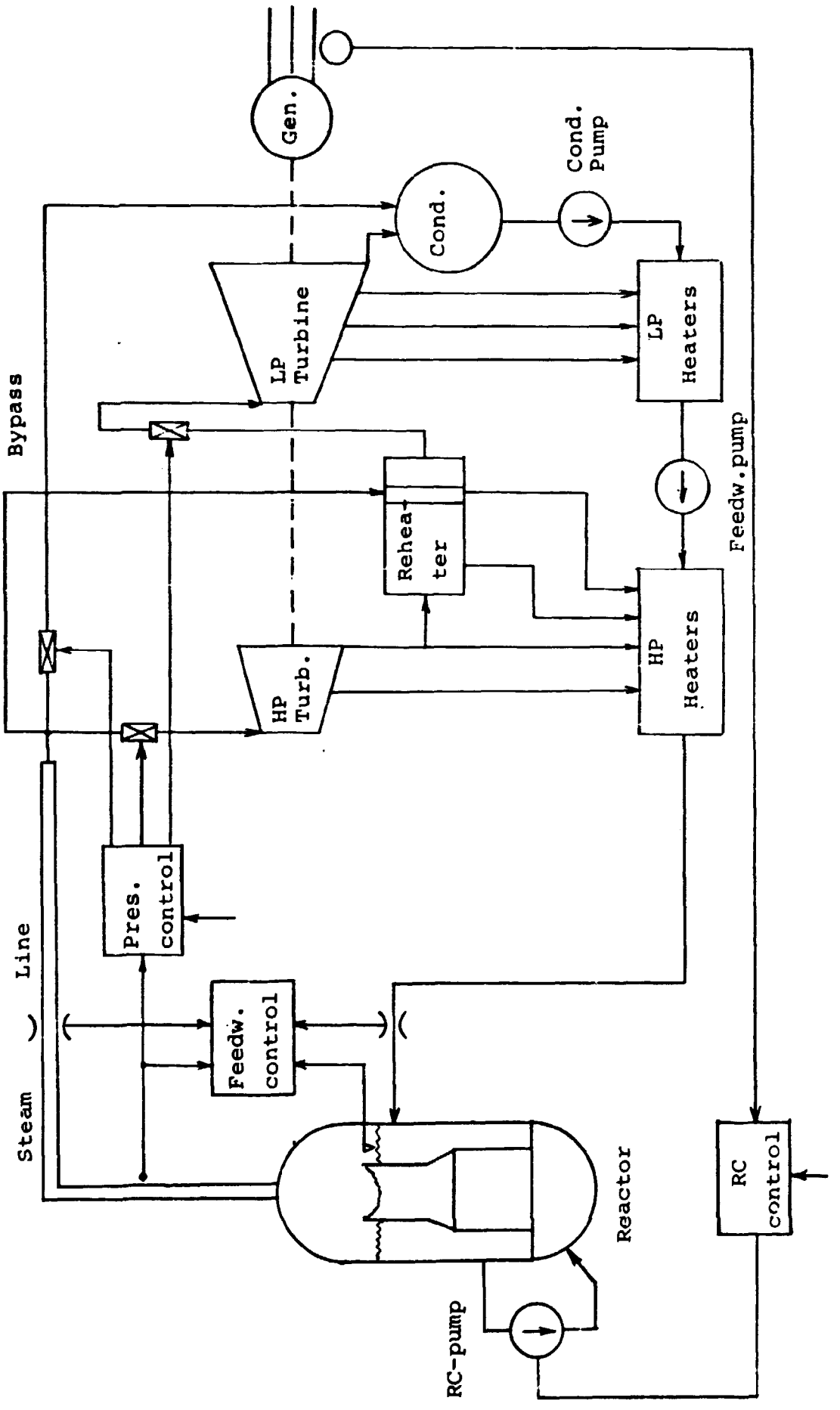


Fig. 1. Scematic diagram of the power plant model.

The reactor model is one-dimensional with 14 active fuel sections along the vertical axis plus two reflector sections for the neutron flux calculation; this is carried out with the prompt-jump approximation and difference technique in a straight forward manner. That approach is selected because it gives a fast calculation with a reasonable time step relative to the thermal time constants, and because a one-dimensional approach can never give a correct power distribution.

The division of the reactor in subsections (nodes) is shown in Fig. 2, page 7. The space in the tank outside the core is divided in 10 volumes and the recirculation loop with the pumps is treated as a pure time delay.

No spatial pressure gradients in the core are taken into account. The pressure is calculated in the riser below the steam separators and in the steam dome. The recirculation flow is calculated by the momentum balance equation integrated from the steam dome to the riser along the flow direction.

The structure of the turbine model is shown in Fig. 4, page 31. The turbine is divided in one HP-section and three parallel LP-sections. The inlet pressures and enthalpies are calculated by lumped parameters referring the turbine steam volumes to the HP- and LP-entrances. The outlet enthalpies are calculated by the Mollier diagram with known inlet and outlet pressures and thermal efficiencies.

The reheater is treated by a point model with one node for the pressure, but two for the enthalpies: the outlet enthalpy from the moisture separator and from the reheater section.

A diagram of the feedwater system is shown in Fig. 5, page 32. Pressure and enthalpy conditions at the steam extraction points on the turbine are calculated by linear combinations of the inlet and outlet values for the turbine section in question. The steam and drain coolers are treated as point models with heat capacity connected to the tubes. The outlet temperature from each steam cooler is delayed with the transport time before it is

used as input to the following drain cooler.

The feedwater flow is calculated by an integrated momentum balance equation for the feedwater line. A diagram showing pressure and flow calculations is given in Fig. 6, page 44.

The steam line is divided in 6 sections described by mass- and momentum balance equations. The main purpose is to simulate the dynamic pressure wave at suddenly varying steam loads as the pressure and its time derivative greatly influences the void in the core. The enthalpy variations are of minor importance as it has no significant feedback to the reactor; so the steam line outlet value is set equal to the reactor outlet value.

The three control systems contain information of confidential nature and will be treated by means of "black boxes" giving only the main characteristics.

The model has been tested by several transient calculations such as turbine trip, reactor trip, loss of load and pump run down, but specific verifications has not yet been done, as some few significant parameters in the control circuit still are uncertain.

2. THE REACTOR MODEL

2.1. Neutron kinetics

The neutron kinetics is described by the one-dimensional, one-group diffusion equation:

$$\frac{1}{v}\frac{\partial\phi}{\partial t} = D\frac{\partial^2\phi}{\partial x^2} - \Sigma_a\phi + (1-\beta)\nu\Sigma_f + \sum_i \gamma_i C_i \qquad (2.1)$$

The parameters D, Σ_a and $\nu\Sigma_f$ are functions of the core composition and the temperature of the different components. Static reactor physics calculations have been used to find polynomials

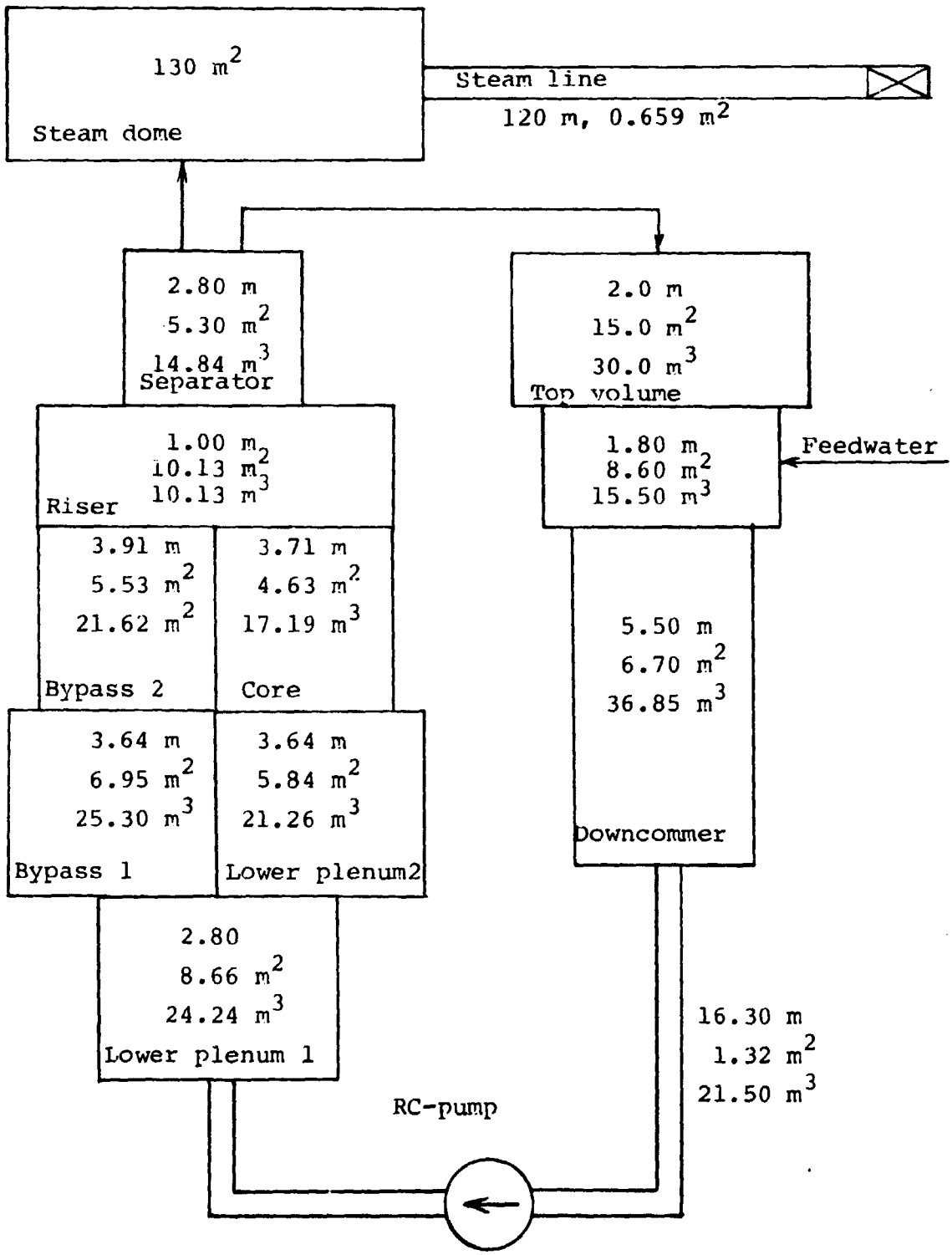


Fig. 2. Reactor model subsections.

in three variables for the three parameters. The three variables are: fuel temperature, water density, and control rod density. Σ_a and $\nu\Sigma_f$ depend on all three variables, while D is independent of the fuel temperature.

In the general form the polynomials have three terms, and each term contains a second order function of the coolant density. In the first term that function stands alone, in the second one it is multiplied by the control rod density (a quantity between 0 and 1) and in the third one it is multiplied by the deviation of the mean fuel temperature from a reference value. So the formulae for the parameters look like:

$$\Sigma = a_o+a_1\rho+a_2\rho^2+(b_o+b_1\rho+b_2\rho^2)C + (c_o+c_1\rho+c_2\rho^2)(T_u-T_{uo}) \quad (2.2)$$

Furthermore, two correction terms are used for the absorption cross section: a correction for neutron absorption by xenon, and a small constant term adjusted to obtain a critical core at prescribed conditions at full load.

As the prompt neutrons' lifetime is small compared with the integration step (10-50 mS) any transient in the flux will vanish within one step when the delayed neutron source is constant. It means that the derivative term in (2.1) can be neglected (prompt-jump approximation).

Integration in space is carried out by division of the space axis in segments Δx and transformation to a difference equation by first-order differences:

$$\frac{d^2\phi}{dx^2} = \frac{1}{\Delta x^2}(\phi_{j-1}-2\phi_j+\phi_{j+1}) \quad (2.3)$$

This corresponds to the assumption that the flux values for three neighbouring sections are placed on a second-order curve when the flux ϕ_j refers to the middle of section number j.

The boundary value problem is solved by means of reflector sections at each end of the fuel sections. It is assumed that the neutron current J back to the reflector from the surroundings

is equal to zero. For the lower section at x = 0 it means that:

$$J_+ = \frac{1}{4}(\phi_o - 2D(\frac{d\phi}{dx})_o) = 0 \qquad (2.4)$$

giving

$$(\frac{d\phi}{dx})_o = \frac{\phi_o}{2D} \qquad (2.5)$$

We can now use an approximation corresponding to the linear difference approximation for the fuel sections and assume that the fluxes ϕ_o, ϕ_1 and ϕ_2 belong to a second-order curve. ϕ_1 and ϕ_2 are the fluxes in the middle of the reflector section and the first fuel section, and ϕ_o the flux at the reflector boundary. This gives an expression for the flux slope at x = 0:

$$(\frac{d\phi}{dx})_o = \frac{9\phi_1 - \phi_2}{3\Delta x + 16D} \qquad (2.6)$$

where D is the diffusion constant for the reflector section.

Similarly we obtain an expression for $\frac{d\phi}{dx}$ at the other reflector boundary:

$$(\frac{d\phi}{dx})_{m+1} = - \frac{9\phi_m - \phi_{m-1}}{3\Delta x + 16D} \qquad (2.7)$$

where m is the total number of sections including the two reflector sections.

Insertion of (2.3), (2.6) and (2.7) into the diffusion equation (2.1) gives:

$$C_{j,j-1}\phi_{j-1} + C_{j,j}\phi_j + C_{j,j+1}\phi_{j+1} = (-\Sigma_i \gamma_i C_i)_j \qquad (2.8)$$

which can be written as a matrix equation

$$\{C\}\phi = S \qquad (2.9)$$

where S is a source vector and {C} is a band matrix with

$$C_{j,j} = (1-\beta)\nu\Sigma_f - \Sigma_a - \frac{2D}{\Delta x^2}$$

$$C_{j,j-1} = C_{j,j+1} = \frac{D}{\Delta x^2}$$

for $2 \leq j \leq m-1$.

$$C_{1,1} = C_{m,m} = -(\Sigma_a + \frac{D}{\Delta x}(\frac{1}{\Delta x} + \frac{9}{3\Delta x + 16D}))$$

$$C_{1,2} = C_{m,m-1} = \frac{D}{\Delta x}(\frac{1}{\Delta x} + \frac{1}{3\Delta x + 16D})$$

for the first and last row in (2.9).

The solution of the diffusion equation is thus reduced to the solution of n linear equations with coupling coefficients different from zero only for neighbouring elements. The coefficients in the matrix C have to be calculated for each time step as functions of three variables, as mentioned earlier.

The neutron flux is converted to power according to the equation:

$$N_j = \frac{A}{\nu} V_f \nu \Sigma_f \phi_j \qquad (2.10)$$

where N_j is the neutron power in core section j, A is the energy release per fission, and V_f is the core volume per section.

The source term in equation (2.7) is the delayed neutrons for which we use three groups. The concentration of the emitters is calculated according to the equation:

$$\dot{C}_i = -\lambda_i C_i + \beta_i \nu \Sigma_f \phi \qquad (2.11a)$$

for each group in each core section. It means that (2.11) introduces 3 x 14 integration variables; that is the reason for using only three groups.

The constants for the delayed neutrons are given in table 2.1.

The xenon concentration is given by the balance equations for iodine and xenon:

$$\dot{C}_{io} = \gamma_{io}\Sigma_f\phi - \lambda_{io}C_{io}$$

$$\dot{C}_{xe} = \lambda_{io}C_{io} + \gamma_{xe}\Sigma_f\phi - \lambda_{xe}C_{xe} - \sigma_{xe}C_{xe}\phi \qquad (2.11b)$$

The correction term for the absorption cross section mentioned in the discussion of eq. (2.2) is then given by $\sigma_{xe}C_{xe}$.

The control rod density for eq. (2.2) is determined as the number of rods inserted in a core section divided by the total number of rods. The weight of partly inserted rods is taken as a linear function of the insertion depth. The rods are divided in 12 groups with rod numbers and positions given as input data. The positions are fixed until a scram signal starts the movement to full inservion with a fixed delay time and fixed velocity.

The thermal power is released promptly for the most part, but a small fraction γ is released as a sum of several exponential decay functions. Six functions are used here with decay constants and power fractions as given in table 2.2. The decay heat calculation is performed for the total thermal power in the core, and the space independent decay heat fraction thus obtained is then used for the individual sections. This simplification greatly reduces the number of integration variables and gives only a small error in the spatial distribution of the decay heat during transients, that involve flux distribution variations.

The equations are:

$$\dot{X}_{qi} = (Q_n - X_{qi})\sigma_i, \qquad Q_n = \sum_{j=1}^{14} N_j$$

$$Q_{nd} = (1-\gamma)Q_n + \sum_{i=1}^{6} \gamma_i X_{qi}, \qquad \gamma = \sum_{i=1}^{6} \gamma_i \qquad (2.12)$$

$$N_{dj} = N_j \, Q_{nd}/Q_n$$

where index j stands for the core section number and i for the decay group number.

The power released in each section, N_{dj}, is further divided in three parts:

Y_{qwc} : A fraction released directly in the coolant inside the fuel element.

Y_{qbp} : A fraction released directly in the coolant in the bypass flow.

$(1-Y_{qwc}-Y_{qbp})$: The main fraction released in the fuel pellets.

Neutron kinetic symbols

ϕ :	Neutron flux	$1/cm^2s$
D :	Diffusion constant	cm
Σ_a:	Absorption cross section	1/cm
Σ_f:	Fission cross section	1/cm
v :	Thermal neutron mean velocity	cm/s
ν :	Yield of prompt neutrons per fission	
C_i:	Concentration of delayed neutron emitter, group i	$1/cm^3$
λ_i:	Decay constant for delayed neutron emitter, group i	1/s
β_i:	Yield of delayed neutrons, group i	
A :	Energy release per fission	MJ
V_f:	Core volume per core section	m^3
Δx:	Length of core section	m
N_j:	Nuclear power in core section j before decay heat correction	MW
N_{dj}:	Nuclear power in core section j after decay heat correction	MW
Q_n:	Total nuclear power before decay heat correction	MW
Q_{nd}:	Total nuclear power after decay heat correction	MW
X_{qi}:	Normalized concentration of decay heat source, group i	
σ_i:	Decay constant for decay heat source, group i	1/s
γ_i:	Energy fraction for decay heat, group i	
C_{io}:	Concentration of iodine	
C_{xe}:	" " " xenon	
γ_{io}:	Fractional yield of iodine per fission	
γ_{xe}:	" " " " xenon " "	

λ_{io}:Decay constant for iodine 1/s
λ_{xe}: " " " xenon 1/s
σ_{xe}:Microscopic cross section for xenon

Table 2.1.

β_i	λ_i s^{-1}	
992 E-6	1.82	
3840 E-6	0.249	β = 6448 E-6
1616 E-6	0.0288	

Table 2.2.

γ_i	σ_i s^{-1}	
2.99 E-3	1.772	
8.25 E-3	5.774 E-1	
15.50 E-3	6.6743 E-2	γ = 0.07336
19.35 E-3	6.214 E-3	
13.49 E-3	4.75 E-4	
13.78 E-3	1.00 E-5	

2.2. Fuel dynamics

The fuel elements are divided in 14 active sections along the vertical axis corresponding to the kinetic nodes. In the horizontal plane only two nodes are used: one for the fuel pellets and one for the canning. Energy balance equations give the fuel mean temperature and the canning temperature. The equations are:

$$C_u \dot{T}_u = Q_u - k_f (T_u - T_{ca})$$

$$C_{ca} \dot{T}_{ca} = k_f (T_u - T_{ca}) - Q_c \qquad (2.13)$$

$$\frac{1}{k_f} = Z_{ugca} = f(T_u)$$

The heat resistance Z_{ugca} is a second order function of the fuel temperature representing both the fuel pellet, the air gap and the canning.

As one node for the pellets may be considered insufficient some remarks will be appropriate: the reactivity feedback from the fuel is weak, and only the mean temperature is used for that calculation. For the heat flow calculations only a minor dynamic error arises as shown in Ref. 1, Chapter 3, and the error is not essential compared with the uncertainty in the air gap resistance. Furthermore, if the center temperature of the fuel should be an important output variable it may be found using a form factor to be a function of the mean temperature. That function may be found with the model mentioned below.

A separate fuel heat transfer model with 10 shell sections for the pellets has been described earlier in Ref. 1, Chapter 3.1. For the present work it has been used for calculation of the function for Z_{ugca} with some modifications. The model has been programmed for both transient and quick static calculations. The equations are given in detail in Ref. 1, but two points should be mentioned: The thermal conductivity λ_u for the pellets is taken as a function of the temperature,

$$\lambda_u = 38.24/(T+129.4) + 4.788\ E\text{-}13 \times T^3 \qquad W/cm\ ^oC,$$

where T is in deg. Kelvin; also, the thermal conductivity for air gap plus canning is used as a tabulated function of the fuel mean temperature with data from the reactor manufacturer.

Fuel dynamics symbols

All symbols are for one axial section; the section index j has been omitted for simplicity:

Q_u : Nuclear power released in the fuel
Q_c : Heat flow to the coolant
T_u : Fuel mean temperature
T_{ca}: Canning temperature
C_u : Fuel heat capacity
C_{ca}: Canning heat capacity
k_f : heat transfer coefficient from fuel to canning.

2.3. Hydraulic dynamics in the core

2.3.1. Heat transfer from fuel to coolant

The heat flow from the fuel to the coolant is calculated according to the well known correlation of Dittus-Boelter or Thom depending on which of them gives the greatest flow. The calculations are done in parallel and the highest value is chosen. Besides, the heat released directly in the coolant is added. The equations are:

$$Q_{c1} = \frac{0.023\ E\text{-}3}{De_c^{0.2}} O_{ca} \left(\frac{W_c}{A_c}\right)^{0.8} H_c(T_c)\,(T_{ca}-T_c)$$

$$Q_{c2} = 1.973\ E\text{-}3\ O_{ca} \exp\left(\frac{p_r}{43.4}\right)(T_{ca}-T_{ps})^2 \qquad (2.14)$$

$$Q_c = \mathrm{Max}(Q_{c1}, Q_{c2})$$

$$Q_{cc} = Q_c + Y_{qwc}\,\dot{N}_d$$

where

$$H_c(T) = c_p^{0.3}\ \lambda^{0.7}\ \eta^{-0.5}$$

For the normal power range the heat flow will be governed by Q_{c2}, only for the lowest core section will Q_{c1} be dominant.

The heat transfer parameter H_c is discussed in Ref. 1, Chapter 3. The approximation

$$H_c = 0.924 + 0.001(T_c-300)$$

is valid with an accuracy of 1 % in the working range 250 - 300 °C.

2.3.2. Heat transport by the coolant

The heat transferred to the coolant is used for heating the water to saturation and for steam production. The problem in calculating subcooled boiling has been discussed by many specialists in thermodynamics. One method known as the RAMONA model for the boiling mechanism is proposed in Ref. 3. The main idea is to use two evaporation terms, one for surface boiling and one for bulk boiling. The two terms look like:

$$W_{sf} = Q^*/(h_{fg}+c_p(T_g-T_f)\frac{\rho_f}{\rho_g} + (T_{ca}-T_g)(\frac{\rho_f}{\rho_g}-1)\frac{c_p}{2}) \qquad (2.15a)$$

$$W_{bb} = (R_o+R_i\alpha(1-\alpha))((T_f-T_g)+\kappa|T_f-T_g|)/h_{fg} \qquad (2.15b)$$

where index f stands for fluid, g for steam, and Q^* is the heat flow to the coolant determined by Q_{c2} when it dominates Q_{c1} in eqs. (2.14); α is the void fraction and R_o, R_1 and κ are parameters.

Eq. (2.15a) utilizes the total heat for steam production but reduces the amount of steam using an artificially increased value for the evaporation heat. Eq. (2.15b) gives a positive evaporation term when $T > T_{cs}$, transferring power from superheated water to steam, and a negative term when $T < T_{cs}$, transferring power from steam to subcooled water.

Eqs. (2.15) have been tried in the reactor model for both static and dynamic calculations. It appears that they give very poor numerical stability and for certain conditions even oscillations. That is not surprising as eq. (2.15b) contains a condensation term that does not go towards zero together with the void fraction resulting in oscillations between no void and a small void

content in the first section with a possibility for steam production. The stability can be improved by setting $R_o = 0$ for $T_c < T_{cs}$, but a poor dynamic stability during integration is still obtained. Experiments have shown that the stability can be improved further and the integration step increased using a very small condensation term and the normal value for the evaporation heat, but an otherwise decreased value of the heat used for evaporation. That leads to another boiling model defined by the following equations:

$$Q_{cb} = Q_{c2}((T_c-T_c^*)/(T_{cs}-T_c^*))^2 \quad \text{for } Q_{c2} > Q_{c1}$$

$$\text{otherwise } Q_{cb} = 0 \tag{2.16a}$$

$$W_{sf} = Q_{cb}/(h_{fg}+\kappa_2 c_p(T_{cs}-T_c)) \tag{2.16b}$$

$$W_{bb} = V_c\kappa_o(R_o+R_1\alpha(1-\alpha))(T_c-T_{cs}) \quad \text{for } T_c > T_{cs} \tag{2.16c}$$

$$W_{bb} = V_c\kappa_1 R_1\alpha(1-\alpha)(T_c-T_{cs}) \quad \text{for } T_c < T_{cs} \tag{2.16d}$$

$$W_e = W_{sf} + W_{bb} \tag{2.16e}$$

$$Q_{ce} = W_e(h_{fg}+c_p(T_{cs}-T_c)) \tag{2.16f}$$

$$Q_{ct} = Q_{cc} - Q_{ce} \tag{2.16g}$$

The parameter T_c^* is an essential temperature parameter some degrees below the saturation value. Together with the other parameters it must be adjusted to fit the void profile for other more detailed calculations. Here it is set equal to the coolant core inlet temperature. κ_2 is set equal to 1, κ_1 to 0, which exclude steam condensation completely, and κ_o is set to 0.25 in order to make the superheating of the water suitable low (0.2 - 0.3 ^oC).

The values for R_0 and R_1 are taken from the ANDYCAP model, Ref. 4, $R_0 = 15$ and $R_1 = 40$ MW/m^3 ^oC.

Besides the parameter set here, the neutron cross section coef-

ficients, mentioned in Chapter 2.1, and the steam slip factor S discussed below, are essential for the power and the void pr - files and must be considered together when adapted to a given reactor core. The static profiles for 100 % power as obtained with the present approach is shown in Fig. 3 and compared with the result from the power company obtained by the program "POLCA-ON-LINE".

Eqs. (2.16) are static equations only. Dynamic equations cannot be based upon the energy balance directly due to the empirical procedure in eqs. (2.16); but energy conservation considerations can be made for the steam and water phase separately. This leads to a correction term to eq. (2.16e) and to an equation for the coolant temperature:

$$W_e = W_{sf} + W_{bb} - V_c \alpha \rho_{gs} h'_{gs} \dot{P}/h_{fg} \tag{2.16e}$$

$$T_{c,j} = (Q_{ct} - W_{f,j-1}(T_{c,j} - T_{c,j-1}))/V_c(1-\alpha)\rho_{fs}c_p \tag{2.17}$$

where the core section index j is used only where it is needed. The mark "'" indicates a derivation with respect to the pressure. The mass balance is used to determine the mass flows out from the core sections:

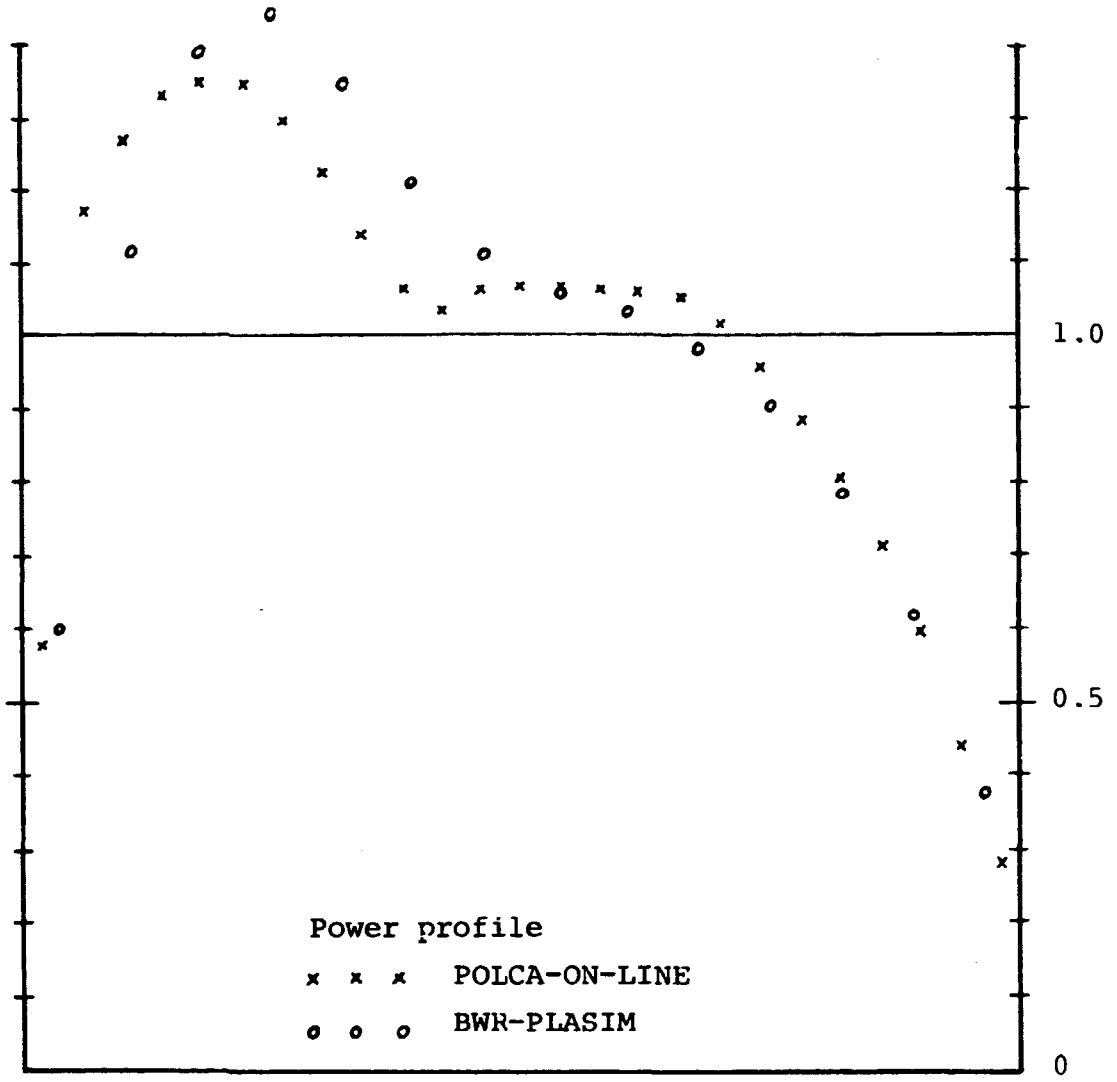
$$W_{f,j} = W_{f,j-1} - W_e + V_c(\rho_{fs}\dot{\alpha} - (1-\alpha)\dot{\rho}_{fs}) \tag{2.18a}$$

$$W_{g,j} = W_{g,j-1} + W_e - V_c(\rho_{gs}\dot{\alpha} + \alpha\dot{\rho}_{gs}) \tag{2.18b}$$

In these equations the saturation density ρ_{fs} for the water is used thrcughout the core giving a small error (2-3 %) in the lower part.

A correlation for the steam velocity through the cooling channel is necessary for determination of the void content. The commonly used slip ratio S defined as the ratio between steam and water velocity is used. It gives a relation between the mass flows:

$$W_g = W_f \frac{\alpha}{1-\alpha} \frac{\rho_{gs}}{\rho_{fs}} S \tag{2.19}$$



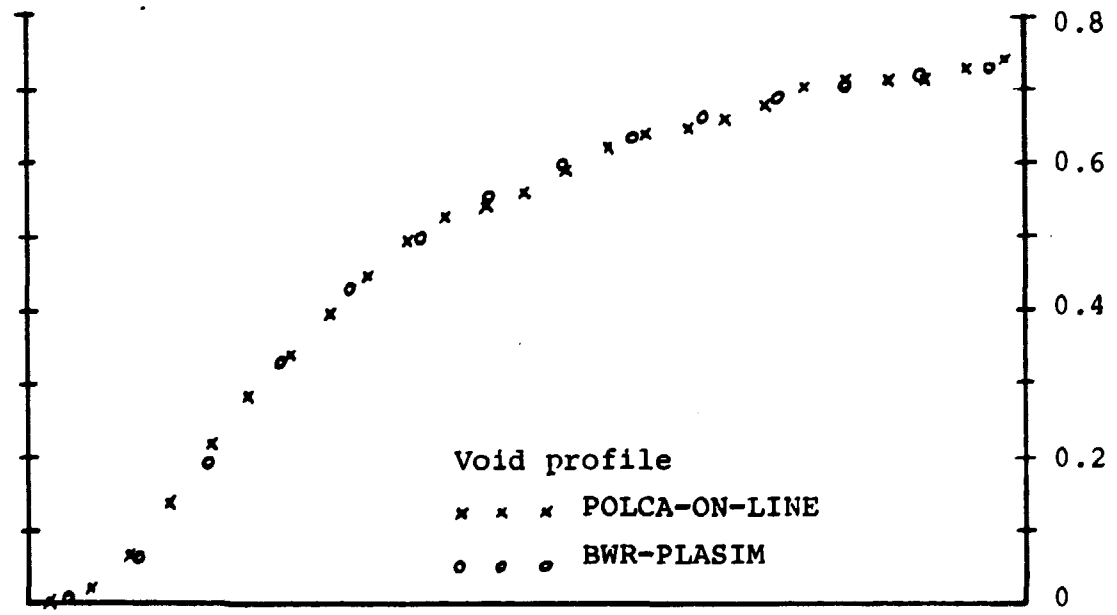


Fig. 3. Power and void profiles at 100% power.

Eqs. (2.18) and (2.19) are combined and give a solution for the water mass flow from a core section when the inlet flow and the evaporation rate are known:

$$W_{f,j}\left(1 + \frac{\alpha S}{1+\alpha}\right)_j = W_{f,j-1}\left(1 + \frac{\alpha S}{1+\alpha}\right)_{j-1} + W_e(\rho_{fs}-\rho_{gs})/\rho_{gs} - V_c\dot{P}((1-\alpha)\rho'_{fs} + \alpha\rho'_{gs}\rho_{fs}/\rho_{gs}) \quad (2.20)$$

Eq. (2.18a) is now used for calculation of the void fraction:

$$\dot{\alpha} = (W_{f,j}-W_{f,j-1}+W_e+V_c(1-\alpha)\rho'_{fs}\dot{p})/V_c\rho_{fs} \quad (2.21)$$

The slip ratio S is used as a function of α as in Ref. 4.

$$S = S_o + a\alpha^b \quad (2.22)$$

However, the data used for the ANYCAP model does not give the correct outlet void so the values have been modified to give agreement with other static calculations. The following values are used:

$$S_o = 1.13 , \qquad a = 0.85 , \qquad b = 1.75$$

In order to save computing time (2.22) has been approximated by a third-order function:

$$S = a_0 + a_a\alpha + a_2\alpha^2 + a_3\alpha^3$$

2.3.3. Pressure drop by friction

The friction pressure drop is needed for calculation of the coolant flow in Chapter 2.6.

From the water mass flow in each core section the steam mass flow and the volume flows per square meter can be found using the slip ratio S:

$$U_{f,j} = W_{f,j}/A_c\rho_{fs}$$

$$U_{g,j} = U_{f,j} S_j \alpha_j / (1-\alpha_j) \qquad (2.23)$$

$$W_{g,j} = U_{g,j} \; A_c \; \rho_{gs}$$

The pressure drop is calculated as in Ref. 1 page 30 using the Becker correlation for one-phase friction and the two-phase friction multiplier. The equation is:

$$\frac{\Delta p_c}{\rho_{fs}} = 0.092 \; \frac{\Delta x}{D_{ec}^{1.2}} \; F_f(T) \; \sum v^{1.8} \left(1+2400 \; \frac{x}{p}\right) \qquad (2.24)$$

where X is the steam quality $W_g/(W_f+W_g)$,

v is the velocity $(W_f+W_g)/A_c\rho_{fs}$

and F is a friction parameter $(\eta/\rho)^{0.2}$

The pressure drop has been normalized by ρ_{fs} in order to obtain the present form of the friction parameter F, which appears to be fairly constant in a large pressure and temperature range.

Symbols for core hydraulics

Q_{c1}: Convective heat flow from fuel surface
Q_{c2}: Boiling " " " " " "
Q_c: Actual " " " " " "
Q_{cc}: Total heat flow to coolant
Q_{ce}: Heat used for steam production
Q_{ct}: " " " water heating
V_c: Core section volume
A_c: " " cross section
D_{ec}: " " hydraulic diameter
W_c: Water mass flow to the cooling channel
W_f: " " " in " " "
W_g: Steam mass flow in " " "
W_e: Evaporation rate in the cooling channel
U_f: Water volume flow in the cooling channel
U_g: Steam colume flow in " " "
α: Void fraction

X: Steam quality
S: Steam slip ratio
T_c: Coolant temperature
T_{cs}: Saturation temperature

Steam and water parameters:
ρ: Density
h: Enthalpy
c_p: Specific heat
η: Dynamic viscosity
index gs stands for saturated steam
" fs " " " " water
"'" indicates derivation with respect to pressure.

2.4. The core bypass

The bypass is divided in two volumes, one below the core without heat transfer from the fuel and one around the cooling channels with a fraction, Y_{qbp}, of the nuclear power going directly to the water. The equations for the temperatures are:

$$\dot{T}_{bp1} = W_{bp}(T_{\ell p1} - T_{bp1})/V_{bp1}\rho_{fs} \tag{2.25}$$

$$\dot{T}_{bp2} = (Y_{qbp}Q_{nd}/c_p + W_{bp}(T_{bp1} - T_{bp2}))/V_{bp2}\rho_{fs} \tag{2.26}$$

The water flow W_{bp} is a fixed fraction, X_{bp}, of the total coolant flow W_{rc}. The saturation value ρ_{fs} is used for the water density for simplicity.

Symbols for the core bypass

V_{bp1}: Volume of core bypass lower part
V_{bp2}: " " " " upper part
W_{bp}: Water flow through core bypass
T_{bp1}: Temperature of core bypass lower part
T_{bp2}: " " " " " upper part.

2.5. The reactor top

The part of the reactor above the core is divided in four volumes: the riser, the steam separators, the steam dome and the top volume around the separators above the feedwater inlet.

The pressure is calculated for the riser and the steam dome. The volumes have no heat sources, but mass exchange between steam and water phases takes place due to pressure variation in space and time and the cold water flow from the core bypass.

2.5.1. The riser

It is assumed that the riser contains sufficiant steam to keep the water at saturation.

The mass exchange between the phases due to cold water from the core bypass is:

$$\Delta W_r = W_{bp} c_p (T_{cs} - T_{bp2}) / h_{fg} \tag{2.27}$$

resulting in net steam and water flows to the riser as:

$$W_g = W_{go} - \Delta W_r \quad , \quad W_f = W_{fo} + \Delta W_r \tag{2.28}$$

where index "o" indicates outlet from the core.

The mass balance gives an equation for the pressure in the riser:

$$\dot{p}_r V_r (\alpha_r (\rho'_{gs} - \rho'_{fs}) + \rho'_{fs}) = W_g + W_f - W_r + V_r (\rho_{fs} - \rho_{gs}) \dot{\alpha}_r \tag{2.29}$$

The energy and mass balance are combined to give an equation for the void fraction:

$$\dot{\alpha}_r V_r \rho_{fs} = W_{fr} - W_f - \dot{p}_r \frac{V_r}{h_{fg}} (\alpha_r \rho_{gs} h'_{gs} + (1-\alpha_r)(\rho_{fs} h'_{fs} - h_{fg} \rho'_{fs}) - 0.1)$$

(2.30

Eqs. (2.29) and (2.30) can be combined and give a solution for $\dot{p}$ without $\dot{\alpha}_r$:

$$\dot{p}_r V_r \Big[\alpha_r \Big(\rho'_{gs} + (\rho_{fs}-\rho_{gs}) \frac{h'_{gs}}{h_{fg}} \frac{\rho_{gs}}{\rho_{fs}} \Big) + (1-\alpha_r) \Big(\dot{\rho}_{fs} + (\rho_{fs}-\rho_{gs}) \Big(\frac{\dot{h}_{fs}}{h_{fg}} - \frac{\dot{\rho}_{fs}}{\rho_{fs}} \Big) \Big)$$

$$- \frac{0.1}{\rho_{fs} h_{fg}} (\rho_{fs}-\rho_{gs}) \Big] = W_g - W_{gr} + \frac{\rho_{gs}}{\rho_{fs}} (W_f - W_{fr}) \qquad (2.31)$$

The steam and water flows out from the riser are given by:

$$W_{gr} = X_r W_r \quad , \quad W_{fr} = W_r - W_{gr} \qquad (2.32)$$

$$X_r = \frac{\alpha_r S_r \rho_{gs}}{\alpha_r S_r \rho_{gs} + (1-\alpha_r) \rho_{fs}}$$

2.5.2. The steam separators

The steam separators are described by the momentum balance alone for determination of the total mass flow. It is assumed that the steam quality throughout the separators is the same as in the riser, so the friction force can be calculated as in Chapter 2.3.3 for a single core section. The equations will then be:

$$\frac{\Delta\rho_s}{\rho_{sf}} = k_{sf} (1+2400 \frac{X_r}{\rho_r}) \left(\frac{W_r}{A_s \rho_{fs}} \right)^{1.8} \qquad (2.33)$$

$$\dot{W}_r = \frac{A_s}{L_s} ((p_r - p_e) 1.E5 - \Delta p_s) - g A_s (\alpha_r \rho_{gs} + (1-\alpha_r) \rho_{fs}) \qquad (2.34)$$

The friction coefficient k_{sf} is determined at full load steady state so the pressure difference $p_r - p_e = 0.3$ bar.

Neglect of the energy balance means that no heat capacity is connected to the coolant in the separators and no mass exchange between phases takes place. These phenomena are referred to the riser and the top volume. Furthermore it is assumed that all the steam goes to the steam dome; this means that the separators are ideal.

2.5.3. The top volume around the separators

The water leaving the separators is assumed to change state from saturation at the pressure p_r to saturation at p_e by steam flashing. The amount of steam is:

$$\Delta W_{sp} = (p_r - p_e) W_{fr} h'_{gs} / h_{fg} \tag{2.35}$$

The water flow to the top volume is then reduced to:

$$W_t = W_{fr} - \Delta W_{sp} \tag{2.36}$$

The temperature of the bulk water in the top volume follows the saturation temperature at the pressure p_e very quickly by flashing at decreasing pressure, and slowly at increasing pressure. Using the same procedure as for bulk boiling in the core the equation will be:

$$\Delta W_{gt} = V_t \kappa_o R_o (T_t - T_{es}) \qquad \text{for } T_t > T_{es} \tag{2.37}$$

$$\text{otherwise } \Delta W_{gt} = 0$$

$$\dot{T}_t = (W_t (T_{es} - T_t) - \Delta W_{gt} h_{fg} / c_p) / V_t \rho_{fs} \tag{2.38}$$

The steam production to the steam dome is then:

$$W_{ge} = W_{gr} + \Delta W_{sp} + \Delta W_{gt} \tag{2.39}$$

The water level in the tank is determined for the top volume as:

$$\dot{W}_{lev} = (W_t + W_{fe} - W_{rc} - \Delta W_{gt}) / A_t \rho_{fs} \tag{2.40}$$

where W_{fe} is the feedwater flow.

2.5.4. The steam dome

The mass balance is used to determine the pressure p_e. The steam load W_l is given by the steam load circuit, here the steam line

model. The energy balance equation is ignored assuming saturation conditions at all the times; this means that steam enthalpy variations due to pressure variations is neglected as they are small compared with enthalpy variations in the water mass in the top volume.

The equation for the pressure is:

$$\dot{p}_e = (W_{ge}-W_\ell)/V_e\rho'_{gs} \qquad (2.41)$$

Symbols for the reactor top

The riser:

W_r: Total mass flow out from the riser
W_{gr}: Steam " " " " " "
W_{fr}: Water mass flow out from the riser
W_g: Steam " " to the riser
W_f: Water " " " " "
V_r: Volume
α_r: Void fraction
p_r: Pressure
X_r: Steam quality
S_r: Slip ratio

The steam separators:

A_s: Flow cross section
L_s: Flow length
Δp_s: Friction pressure drop
g: Gravitation constant

The top volume:

V_t: Volume
T_t: Temperature
W_t: Water flow from separators
ΔW_{sp}: Steam released by the pressure drop p_r-p_e
ΔW_{gt}: Steam released by flashing at pressure decrease.
$W_{\ell ev}$: Reactor water level relative to core outlet

The steam dome:

V_e: Volume

W_{ge}: Total steam flow from reactor

W_ℓ: Steam load.

2.6. The recirculation path

The recirculation path contains four volumes inside the tank: the feedwater chamber, the downcommer and the lower plenum divided in two parts. The downcommer and the lower plenum is connected by the pump circuit characterized by a long narrow tube.

The feedwater chamber has two inlet flows at quite different temperatures (about 285 and 175°C) with slightly different values for the specific heat. This is taken into account by the ratio c_{pr} between the low and the high value.

The energy balance for the four volumes then gives equations for the temperatures:

$$\dot{T}_b = ((W_{rc}-W_{fe})(T_t-T_b) + c_{pr}W_{fe}(T_{fe}-T_b))/V_b\rho_{fs} \tag{2.42}$$

$$\dot{T}_{dc} = W_{rc}(T_b-T_{dc})/V_{dc}\rho_{fs} \tag{2.43}$$

$$\dot{T}_{\ell pl} = W_{rc}(T_{rci}-T_{\ell pl})/V_{\ell pl}\rho_{fs} \tag{2.44}$$

$$\dot{T}_{\ell p2} = W_c(T_{\ell pl}-T_{\ell p2})/V_{\ell p2}\rho_{fs} \tag{2.45}$$

The dynamics of the pump path is described by a pure time delay:

$$T_{rci}(t) = T_{dc}(t-\tau_{hc}) \tag{2.46}$$

where τ_{hc} is the time a water particle used to travel from tube inlet to outlet with varying speed and arrive at the outlet at time t.

The approach with complete mixing in the reactor volumes and

flow with no mixing in the pump path is an idealized one, but the dynamic errors will to a certain degree cancel each other at the end.

Symbols in the recirculation path

V_b: Volume of feedwater chamber
V_{dc}: " " downcommer
$V_{\ell pl}$: " " lower plenum part 1
$V_{\ell p2}$: " " " " " 2
W_{rc}: Water mass flow through pumps
W_{fe}: Feedwater flow
W_c: Water mass flow through the core
T_b: Temperature in feedwater chamber
T_{dc}: Temperature in downcommer
$T_{\ell pl}$: " " " lower plenum part 1
$T_{\ell p2}$: " " " " " " 2
T_{rci}: " " at tank inlet from pumps
T_{fe}: " " of feedwater.

2.7. Reactor Coolant Flow

The coolant flow is calculated by the momentum balance equation integrated from the reactor top through the recirculation path and the core to the riser. The equation is simplified by neglecting the gain and losses in momentum due to feedwater inlet and restrictions in the flow area. The main terms are the driving forces and the frictional losses. The equation can be written as:

$$\dot{J} = D - \Delta p_f$$

where J is the total momentum per square meter, D is the driving pressure and Δp_f the friction pressure drop.

The driving force from the void in the core is determined by:

$$D_c = g(\rho_{fs}-\rho_{gs}) \sum_{\Delta x} \alpha_{\%} \Delta x \qquad (2.47)$$

where the summation is carried out for the 14 core sections.

The water column around the steam separators above the core outlet gives a static head acting as a driving force:

$$D_b = g\rho_{fs}W_{\ell ev} \tag{2.48}$$

The pump pressure is found by a functional approximation to the pump characteristic. It can be expressed as a polynomial with the relative pump speed O_{rcp} and the volume flow V_{rc} as independent variables:

$$\Delta p_u = A_{rcp}O_{rcp}^2 + B_{rcp}O_{rcp}V_{rc} + C_{rcp}V_{rc}^2 \tag{2.49}$$

where the three parameters have been optimized to data from the normal working area of the pump characteristic.

The frictional forces in the loop outside the core follow the same law as used for one-phase friction in the core. It means that the pressure drop normalized by ρ_{fs} can be written as:

$$\frac{\Delta p_d}{\rho_{fs}} = D_{pdk}v_d^{1.8} \tag{2.50}$$

where the D_{pdk} is determined at full power steady state so a specified pump speed is obtained.

The water velocity in the downcommer is now found by:

$$v_d = (((p_e - p_r + p_{pu})\ 1.E5 + D_c + D_b)/\rho_{fs} - \frac{\Delta p_c}{\rho_{fs}} - \frac{\Delta p_d}{\rho_{fs}})/S_{\ell y} \tag{2.51}$$

where

$$S_{\ell y} = \sum_y L_y \frac{A_{dc}}{A_y}$$

with the summation carried out for all cross sections in the path from reactor top to core outlet. $S_{\ell y}$ is used as a constant neglecting the small variation of L in the top volume.

The coolant mass and volume flows are determined as:

$$W_{rc} = v_d A_d \rho_{fs} \quad \text{and} \quad V_{rc} = W_{rc}/\rho_{fs} \tag{2.52}$$

The flow is divided in two parts:

$$W_{bp} = X_{by} W_{rc}$$

flowing around the fuel elements and

$$W_c = W_{rc} - W_{bp}$$

flowing through the fuel elements.

Symbols for coolant flow

D_c: Driving pressure from void in the core
D_b: " " " " water column in the recirculation path above the core outlet
Δp_{pu}: Pump pressure
O_{rcp}: Pump speed relative
Δp_c: Friction pressure drop in the core
Δp_d: " " in the recirculation path
A_d: Downcommer cross section.

3. THE TURBINE AND FEEDWATER HEATERS

The model for the turbine and the feedwater heaters is implemented as an independent routine connected to the reactor and steam line model through the steam inlet and feedwater outlet conditions. The condenser pressure is assumed to be constant. The feedwater flow is controlled by the feedwater control system which is described in Chapter 5. The principal layout of the turbine and feedwater heater system is shown in Figs. 4 and 5. All parameters not given by the geometry of the system or as

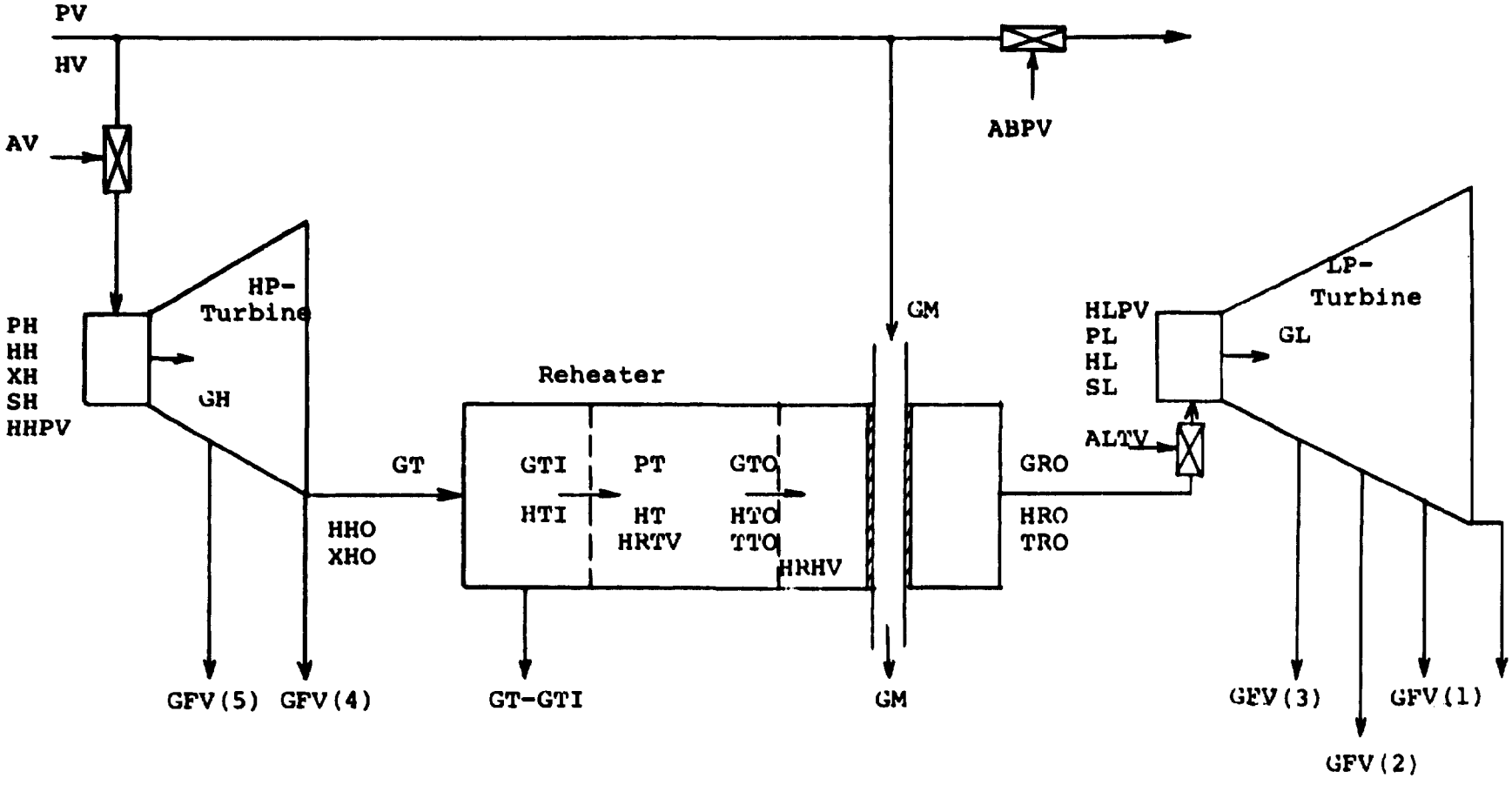


Fig. 4. Structure and variables for turbine and reheater.

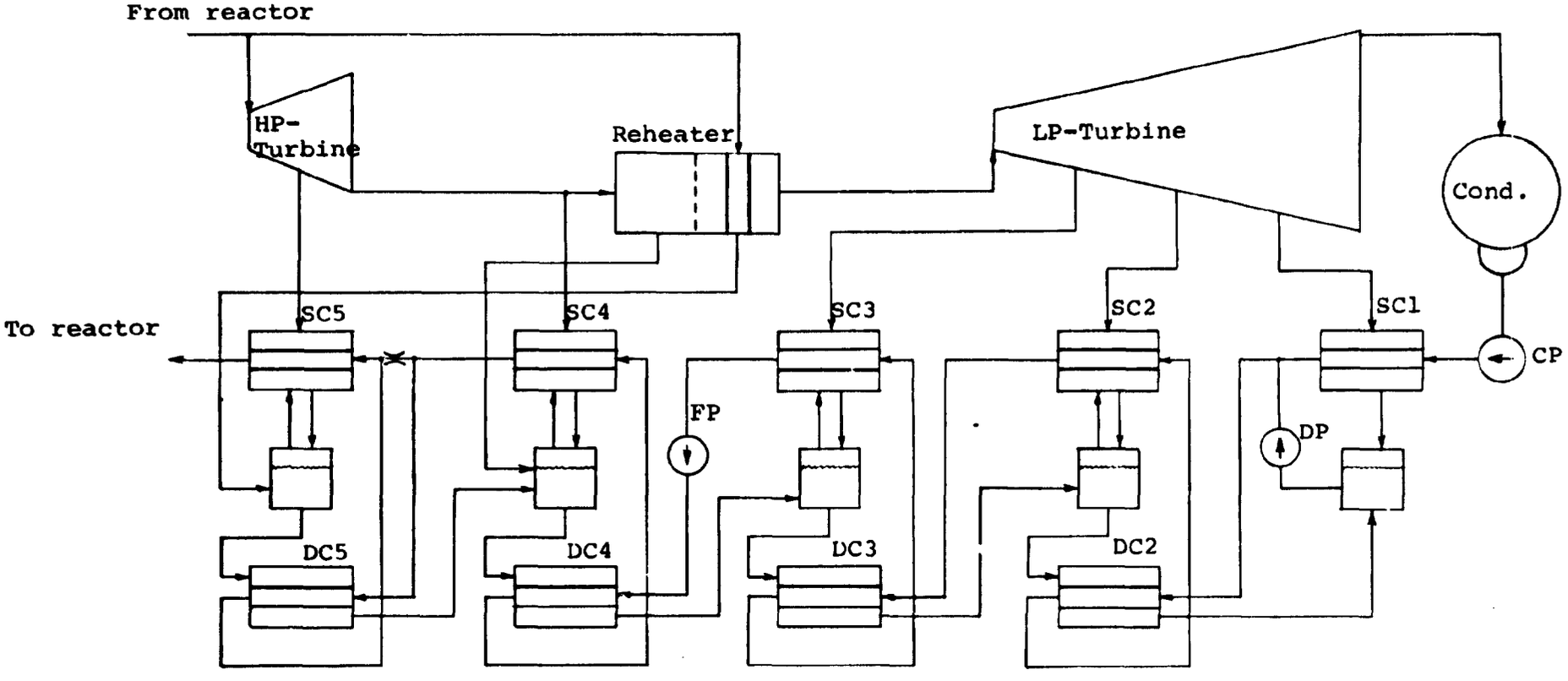


Fig. 5. The feedwater system.

steam data are determined from five sets of static data for the power range 60 - 100 % of full power. As all steam flows inside the system are determined by algebraic equations with the same few state variables in pressure and temperature, the calculations must be performed in a definite order to give coincidence in time.

3.1. Flow and pressure calculations for the turbine

Only three variables are used for the pressure in the system. They are the HP- and LP-turbine inlet pressure and the reheater pressure. For the two turbine stages the steam storage volumes are concentrated in front of the inlets. The mass balance equations give:

$$\dot{p}_h = (G_v - G_h)/V_h \rho'_{gh}$$

$$\dot{p}_\ell = (G_{ro} - G_\ell)/V_\ell \rho'_{g\ell} \qquad (3.1)$$

$$\dot{p}_t = (G_{ti} - G_{ro})/(V_t + V_r)\rho'_{gr}$$

The pressure derivatives $\rho' = \frac{\partial \rho}{\partial p}$ are used as constant values which are slightly different for the three conditions.

The steam flow G_v through the regulating valve is calculated as described in Ref. 1, Chapter 7:

$$G_v = K_v A_v p_v Y_v (1 - A_{trip}) \qquad (3.2a)$$

where the parameter Y_v is a function of the inlet-outlet pressure ratio for the valve. In Ref. 1 it is calculated for idealized conditions and for a constant value of the isentropic exponent. For practical cases it may be approximated by:

$$Y_v = \mathrm{SQRT}(1 - 5.5888(p_h/p_v - 0.577)^2) \quad \text{for } p_h/p_v > 0.577$$

and (3.26)

$$Y_v = 1 \quad \text{for } p_h/p_v \leq 0.577$$

Eq. (3.26) describes one quarter of an ellipse for $P_h/p_v > 0.577$. The steam flow area A_v has been normalized to 1 for full open valve, and the flow constant K_v is determined from static data for the valve. The last term $(1-A_{trip})$ is normally equal to 1; it accounts for the flow cut off at turbine trip, where A_{trip} goes from 0 to 1 as a delayed ramp function with a delay time equal to 0.1 s and a rise time equal to 0.3 s.

The inlet flows to the turbine stages are for normal conditions proportional to the inlet pressures, but at some abnormal conditions with extremely low pressure the flows go toward zero as the outlet-inlet pressure ratio approaches 1. To take care of that situation a variable Y is used as a rough approximation to the function Y in eq. (3.26). The flows to the HP- and LP-turbine stages can then be calculated as:

$$\begin{aligned} G_h &= K_h p_h Y_h \\ G_\ell &= K_\ell p_\ell Y_\ell \\ Y_h &= 1 - \frac{100}{9}\left(\frac{P_t}{P_h}-0.7\right)^2 \\ Y_\ell &= 1 - \frac{100}{9}\left(\frac{P_c}{P_\ell}-0.7\right)^2 \end{aligned} \qquad (3.3)$$

The pressure drop between the reheater and the LP-turbine is small and not essential at all, but the pressure node in the reheater is needed because of the intercept valve in front of the LP-turbine. The steam flow through that valve is simply taken as:

$$G_{ro} = K_{h\ell} A_{\ell tv} (P_t - P_\ell) \qquad (3.4)$$

Finally is the steam flow around the turbine through the bypass valve calculated by:

$$G_b = K_b A_{bpv} P_v \qquad (3.5$$

The valve areas $A_{\ell tv}$ and A_{bpv} are both normalized to 1 for full open valve. The flow constants K_h, K_ℓ, $K_{h\ell}$ and K_b are found from

the static data.

The equations are here given for one LP-turbine. There are actually three with slightly different flows due to deviations in the steam extraction to the feedwater heaters, and the program makes proper allowance for that.

3.2. Enthalpy and power calculation for the turbine

The inlet enthalpy to the regulating valve should actually have been calculated by the steam line model as outlined in Chapter 4, but is here used as the value for the steam dome in the reactor. The approximation is justified by four arguments: the enthalpy variation with pressure is small, the pressure variation is relatively slow, the steam transit time in the steam line is short and finally the influence of small enthalpy variation is not important. So, the inlet enthalpy is calculated as follows assuming 0.5 % humidity in the steam:

$$h_v = 0.995\ h_{gs}(p_e) + 0.005\ h_{fs}(p_e) \tag{3.6}$$

For the turbine itself two state variables for enthalpy calculations are used. These are the total enthalpy at the HP- and the LP-turbine inlets. Furthermore, two state variables are used for steam enthalpy in the reheater; these are discussed in Chapter 3.3. Energy balance equations give:

$$\begin{aligned} \dot{H}_{hp} &= G_v h_v - G_h h_h \quad , \quad h_h = H_{hp}/V_h p_h \rho'_{gh} \\ \dot{H}_{\ell p} &= G_{ro} h_{ro} - G_\ell h_\ell \quad , \quad h_\ell = H_{\ell p}/V_\ell p_\ell \rho'_{g\ell} \end{aligned} \tag{3.7}$$

The steam expansion and variation of steam conditions in the turbines are described by the Mollier diagram in equational form using thermal efficiencies determined from the static data. The calculational procedure is as follows:

First, the inlet steam quality X and entropy S are found from enthalpy h and pressure p. Then, the outlet quality and enthalpy is found for isentropic expansion with known outlet pressure.

Afterwords, the real enthalpy drop Δh is found with a known thermal efficiency γ, and finally the real outlet values for the enthalpy and steam quality.

The equations for the HP-turbine are:

$$X_h = (h_h - h_{fs}(p_h)) / (h_{gs}(p_h) - h_{fs}(p_h))$$

$$S_h = X_h \, S_{gs}(p_h) + (1 - X_h) S_{fs}(p_h)$$

$$X_{ho} = (S_h - S_{fs}(p_t)) / (S_{gs}(p_t) - S_{fs}(p_t))$$

$$h_{ho} = X_{ho} \, h_{gs}(p_t) + (1 - X_{ho}) h_{fs}(p_t) \qquad (3.8)$$

$$\Delta h_h = (h_h - h_{ho}) \gamma_h$$

$$h_{ho} = h_h - \Delta h_h$$

$$X_{ho} = (h_{ho} - h_{fs}(p_t)) / (h_{gs}(p_t) - h_{fs}(p_t))$$

For the LP-turbine inlet the steam is normally superheated, and the entropy is calculated using an extrapolation of the constant pressure line into the superheated steam area of the Mollier diagram, but with a correction of the slope. The constant pressure curve for superheated steam is actually bending upwards against higher enthalpies. However, the straight-line approximation error is small for normal working conditions. The equations are:

$$S_\ell = S_{gs}(p_\ell) + 0.91 (h_\ell - h_{gs}(p_\ell)) (S_{gs}(p_\ell) - S_{fs}(p_\ell)) / (h_{gs}(p_\ell) - h_{fs}(p_\ell))$$

$$X_c = (S_\ell - S_{fs}(p_c)) / (S_{gs}(p_c) - S_{fs}(p_c))$$

$$h_c = X_c \, h_{gs}(p_c) + (1 - X_c) \, h_{fs}(p_c) \qquad (3.9)$$

$$\Delta h_\ell = (h_\ell - h_c) \gamma_\ell$$

$$h_c = h_\ell - \Delta h_\ell$$

The turbine power is calculated for the HP- and LP-turbines separately with corrections for the steam extracted to the feed-water heaters:

$$E_h = G_f \Delta h_f - G_{fv}(5)\,(h_{fv}(5)-h_{ho})$$

$$E_\ell = G_\ell \Delta h_\ell - G_{fv}(3)\,(h_{fv}(3)-h_c) - G_{fv}(2)\,(h_{fv}(2)-h_c) - G_{fv}(1)\,(h_{fv}(1)-h_c) \qquad (3.10)$$

$$E_t = E_h + E_\ell$$

For simplification the equations are given for one LP-turbine, while the program is written for the three LP-turbine layout.

3.3. The reheater model

The principal structure of the model is shown in Fig. 4 together with the turbine. The reheater is divided in two compartments: the moisture separator (MS) and the superheater (SU). The steam inlet conditions are given by eq. (3.8). It is assumed that the moisture is separated near the inlet so the mass and energy balance can be used for saturated steam in the MS.

Four state variables are used to describe the reheater: steam pressure in the whole volume as discussed in Chapter 3.1, total steam enthalpy in the MS, total steam enthalpy and tube temperature in the SU.

The inlet flow with moisture G_t and without moisture G_{ti} is given by:

$$G_t = G_h - G_{fv}(5) - G_{fv}(4)$$

$$G_{ti} = G_t\,(X_{ho}+0.005) \qquad (3.11)$$

It is assumed that after the MS the steam still contain 0.5 % humidity. The enthalpy of the steam flow to the MS compartment is then:

$$h_{ti} = 0.995\ h_{gs}(p_t) + 0.005\ h_{fs}(p_t) \tag{3.12}$$

The steam flow from the MS to the SU section is set equal to the outlet flow to the LP-turbine G_{ro}, as only one pressure node is used. The enthalpy for that steam flow is then given by the total enthalpy state variable H_{rtv}:

$$\begin{aligned} G_{to} &= G_{ro} \\ h_{to} &= h_t = H_{rtv}/V_t p_t \rho'_{gr} \end{aligned} \tag{3.13}$$

The energy balance equation for the MS compartment can now be written as:

$$\dot{H}_{rtv} = G_{ti}\ h_{ti} - G_{to} h_{to} \tag{3.14}$$

And the energy balance for the SU compartment gives:

$$\begin{aligned} \dot{H}_{rhv} &= G_{to}\ h_{to} + Q_{rr} - G_{ro} h_{ro} \\ h_{ro} &= H_{rhv}/V_r p_t \rho'_{gr} \\ Q_{rr} &= K_{qr}(T_{tr} - T_{rm}) \\ T_{rm} &= R_{trm}\ T_{to} + (1-R_{trm}) T_{ro} \\ T_{to} &= T_{sat}(p_t) \\ T_{ro} &= T_{sat}(p_t) + (h_{ro} - h_{gs}(p_t))/c_{pr} \end{aligned} \tag{3.15}$$

Q_{rr} is the heat flow from the tubes to the steam on the secondary side. T_{rm} is a weighted mean value of the inlet and outlet steam temperature. The weighting factor R_{trm} and the heat transfer coefficient K_{qr} are determined from the static data. The outlet temperature T_{to} from the MS is normally equal to the saturation temperature. The specific heat c_{pr} is a mean value for superheated steam in the normal working range.

Eqs. (3.11) - (3.15) describe the secondary side of the reheater for all normal situations. Some abnormal conditions may occur and the program makes provision for three:

The steam from the HP-turbine may be superheated for a very low inlet pressure; for that case

$$G_{ti} = G_t \quad \text{and} \quad h_{ti} = h_{ho} \tag{3.16}$$

For the same situation the steam flow to feedwater heater No. 4 may be greater than the flow through the HP-turbine; for that case $G_t < 0$ and the following relations are used:

$$G_{ti} = G_t \quad \text{and} \quad h_{ti} = h_t \tag{3.17}$$

Finally provision is taken for the possibility that the steam in the MS becomes superheated; for that case

$$T_{to} = T_{sat}(p_t) + (h_{to}-h_{gs}(p_t))/c_{pr} \tag{3.18}$$

The situation with reversed flow out from the SU is not anticipated as the intercept valve position downwards is limited to 3 % open and the flow through the LP-turbine is unlimited in all cases.

The primary side of the SU with live steam inlet flow and condensate outlet flow is described by the following equations:

$$\begin{aligned}
p_m &= p_v - K_{vm}G_m \\
T_m &= T_{sat}(p_m) \\
h_m &= h_{fs}(T_m) \\
Q_{tr} &= K_{qt}(T_m-T_{tr}) \\
\dot{T}_{tr} &= (Q_{tr}-Q_{rr})/C_{tr} \\
G_m &= Q_{tr}/(h_v-h_m)
\end{aligned} \tag{3.19}$$

where C_{tr} is the heat capacity of the tubes. The heat transfer coefficient K_{qt} is determined from the static data. For determination of the steam flow G_m the heat flow to the tubes is used as the "driving force" directly as a measure of the amount of absorbed steam.

3.4. The steam and drain coolers

Feedwater heater No. 1 is a steam cooler without drain cooler while No. 2-5 consist of both a steam and a drain cooler. The drainage is pumped from stage No. 1 into the feedwater line. The layout of the system is shown in Fig. 5 for one line out of two identical lines. The system is simplified by the omission of the tanks that collect the drainage from the steam cooler and the following stage, and for stage No. 4 and 5 also condensate from the reheater.

For each stage it is assumed that the incoming water is brought to the saturation temperature by steam production or comsumption to or from the steam cooler. That is a reasonable approximation for a plant model as the overall influence is very small. However, for a detailed model of the turbine and the feedwater system these tanks ought to be simulated with the calculations of water level and temperature as state variables.

The steam and drain coolers are both described by point models. The calculation of heat transfer between the tube wall and water flows is based upon a static model of the space distributed system as outlined in App. A. The result is that the heat transfer for the distributed system can be calculated in a point model using weighted mean values between inlet and outlet temperatures. The weighting factor is a function of the water flow, specific heat and the heat transfer coefficient.

The energy equation for a steam cooler gives:

$$\dot{T}_{tf} = (k_{tc}(T_{fv}-T_{tf}) - k_{fc}(T_{tf}-T_{fm}))/C_{ts}$$

$$\dot{T}_{fo} = (k_{fc}(T_{tf}-T_{fm}) - G_f c_{pf}(T_{fo}-T_{fi}))/C_{fs}$$

$$T_{fm} = BT_{fi} + (1-B)T_{fo}$$

$$B = 1/Z - E/(1-E) \qquad (3.20)$$

$$E = \mathrm{Exp}(-Z) \quad , \quad Z = \frac{(kL)_f}{G_f c_p} \quad , \quad \frac{1}{(kL)_f} = \frac{1}{k_{tc}} + \frac{1}{k_{fc}}$$

$$T_{fv} = T_{sat}(p_{fv})$$

The total heat transfer coefficient $(kL)_f$ is found from the static data, but the division between k_{tc} and k_{fc} is unknown; here k_{tc} and k_{fc} are chosen to be equal.

The steam flow to the steam cooler is derived from the heat flow to the tubes corrected for the heat exchange with the incoming water needed to keep it at the saturation temperature. It is assumed that all steam is condensed and that the condensate leaves the steam cooler with saturation temperature. The steam flow to the cooler is then:

$$G_{fv} = (k_{tc}(T_{fv}-T_{tf}) - \Sigma G_{di} c_{pd}(T_{ci}-T_{fv}))/(h_{fv}-h_w) \qquad (3.21)$$

$$h_w = h_{fs}(p_{fv})$$

G_{di} and T_{ci} stand for water inlet flow to the stage and temperature. The inlet pressure p_{fv} and enthalpy h_{fv} is given by the turbine. Simple algebraic relations between inlet and outlet values for the turbine stage are determined from static data:

$$(p_{fv}(i) - p_c)/(p_\ell - p_c) = p_{fvr}(i) \qquad \text{for } 1 \leq i \leq 3$$

$$p_{fv}(4) = p_{ho} \qquad (3.22)$$

$$(p_{fv}(5) - p_{ho})/(p_h - p_{ho}) = a_o + a_1 G_h$$

where $p_{fvr}(i)$, a_o and a_1 are constants.

$$(h_{fv}(i)-h_c)/(h_\ell-h_c) = h_{fvr}(i) \qquad \text{for } 1 \leq i \leq 3$$

$$h_{fv}(4) = h_{ho} \tag{3.23}$$

$$(h_{fv}(5) - h_{ho})/(h_h-h_{ho}) = h_{fvr}(5)$$

where $h_{fvr}(i)$ are constants.

For the drain coolers the equations are:

$$\dot{T}_{td} = (k_{td}(T_{dm}-T_{td}) - k_{fd}(T_{td}-T_{fdm}))/C_{td}$$

$$\dot{T}_{fdo} = (k_{fd}(T_{td}-T_{fdm}) - G_f c_{pf}(T_{fdo}-T_{fdi}))/C_{fd}$$

$$\dot{T}_{do} = (G_d c_{pd}(T_{di}-T_{do}) - k_{td}(T_{dm}-T_{td}))/C_d$$

$$T_{dm} = (1-B)\ T_{di} + B\ T_{do} \tag{3.24}$$

$$T_{fdm} = B\ T_{fdi} + (1-B)\ T_{fdo}$$

$$B = 1/Z - E/(1-E)$$

$$E = \mathrm{Exp}(-Z) \quad , \quad Z = \frac{(kL)_d}{c_p}\left(\frac{1}{G_f} - \frac{1}{G_d}\right) \quad , \quad \frac{1}{(kL)_d} = \frac{1}{k_{td}} + \frac{1}{k_{fd}}$$

As for the steam cooler the total heat transfer coefficient $(kL)_d$ is determined from the static data and divided equally between k_{td} and k_{fd}.

Using the point models as described here steady-state solutions that fit perfectly well to the known static data for the power range 60 - 100 % have been found. The deviations for the feed-water temperature are less than $2^{\circ}C$ for all stages.

For the dynamic case the point model is a rough simplification for the individual heaters, but for the complete plant it is regarded as a reasonable compromise.

The connection between the heaters is mainly a time delay, but with some damping caused by mixing at the tube ends and nonuniform flow velocity over the tube cross sections. The combined dynamic effect can be found only by measurements on the actual system and such measurements are not available. The connections are simulated as pure time delays, which overestimate this effect for the connecting tubes alone, but gives at the same time a compensation for time delays lost inside the heaters using the point models. The equations for one connection are:

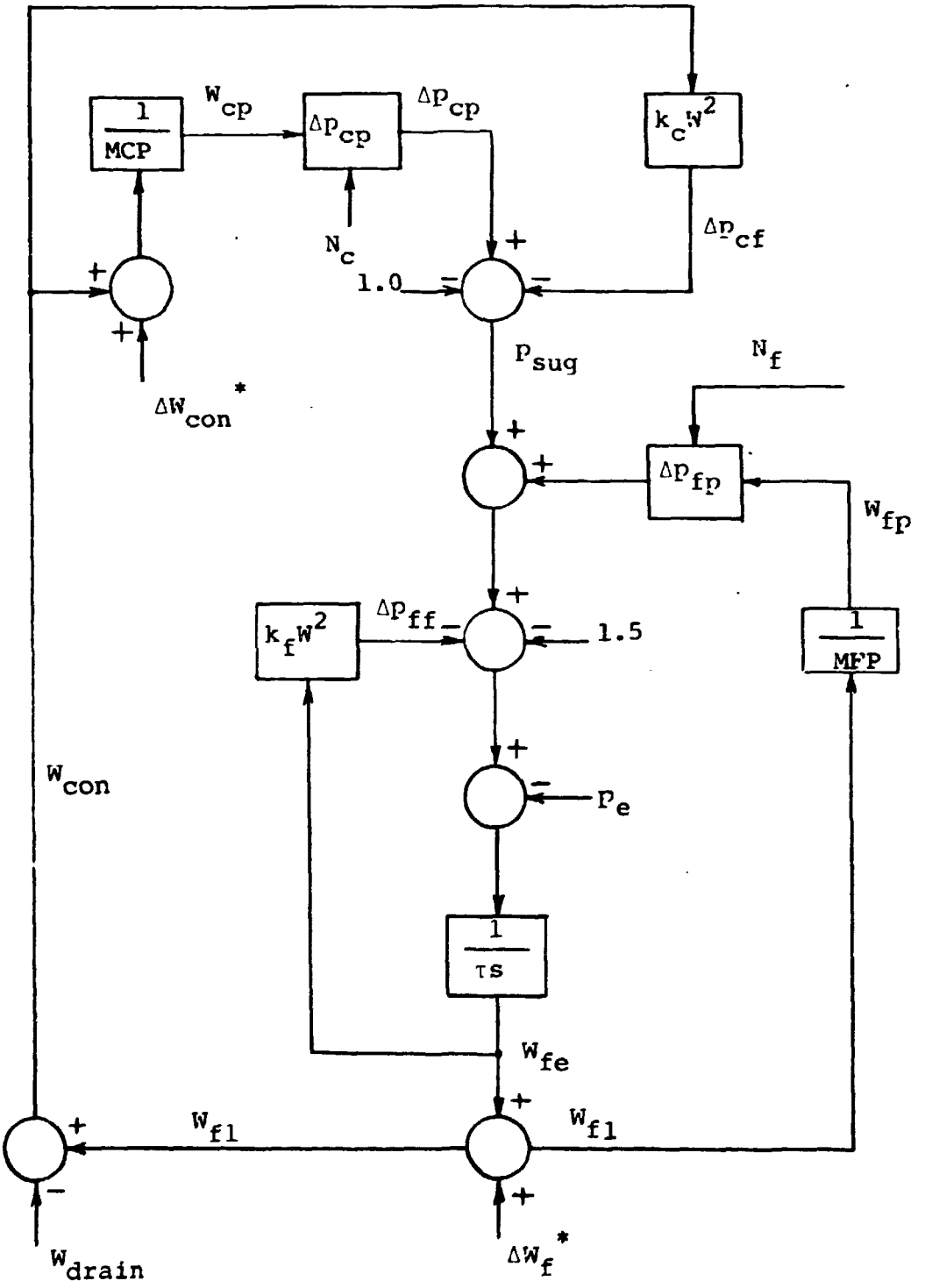
$$T_{fdi}(t) = T_{fo}(t-\tau_1)$$
$$T_{fi}(t) = T_{fdo}(t-\tau_2) \qquad (3.25)$$

where τ_1 and τ_2 are determined as the transportation time for a water particle from one heater stage to the next. The inlet temperature to the reactor is the delayed outlet temperature for feedwater heater No. 5.

3.5. The feedwater flow

The feedwater is pumped from the condenser through the heaters to the reactor by the condensate and feedwater pumps. The drainage is pumped from steam cooler No. 1 to the feedwater string before drain cooler No. 2. The condensate pumps run at constant speed, while the feedwater pump speed is controlled by the feedwater control system.

The feedwater system has two strings with pumps in each string, but is here described as one string with one pump. The pressure and flow relations are shown in the diagram at Fig. 6. The two flow terms ΔW_f and ΔW_{con} represent bypass flows from pump outlets to the condenser at low flows in the pumps. These bypass flows are used in order to keep the pumps in a desirable working area at low power, and are normally equal to zero. Details are not included here. The friction pressure drop in the line is divided in two parts: one after the feedwater pump and one after the condensate pump. This is sufficient to give the pressure p_{sug}



*) Min. flow correction.

Fig. 6. Feedwater flow computation diagram.

in front of the feedwater pump, which is needed for the control system, and is also sufficient for the feedwater flow calculation by the momentum balance equation for the feedwater line:

$$\dot{W}_f = (\Delta p_{cf}+\Delta p_{fp}-\Delta p_{cf}-\Delta p_{ff}-\Delta p_{sta}-p_e)/\tau$$
$$\Delta p_{cf} = K_c W_{con}^2 \quad , \quad \Delta p_{ff} = K_f W_{fe}^2 \qquad (3.26)$$

where $\tau = (L/A)\ 10^{-5}$ for the two lines together. The term Δp_{sta} represents the gravitation pressure difference between the two ends of the line.

The pump pressures Δp_{cp} and Δp_{fp} are calculated as for the reactor recirculation pumps. The equations are:

$$\Delta p_{cp} = a_c + b_c W_{cp} + c_c W_{cp}^2$$
$$\Delta p_{fp} = a_f N_f^2 + b_f N_f W_{fp} + c_f W_{fp}^2 \qquad (3.27$$

where N_f is the normalized feedwater pump speed. In the first equation the pump speed, which is constant, is included in the coefficients.

Symbols for the turbine and feedwater heaters

Pressures

p_v : Before the regulating valve
p_h : " the HP-turbine
p_{ho} : After " " "
p_t : In the reheater
p_ℓ : Before the LP-turbine
p_c : After " " "
$p_{fv}(i)$: In the steam extraction lines.

Flows:

G_v : Through the regulating valve
G_h : Into the HP-turbine
G_t : Into the reheater

G_{ti} : To the moisture separator compartment
G_{to} : To the superheater compartment
G_{ro} : Out from the reheater
G_{ℓ} : Into the LP-turbine
G_m : To the reheater primary side
$G_{fv}(i)$: Steam to the steam coolers
$G_f(i)$: Feedwater through heaters
$G_d(i)$: Drainage " "
W_{fe} : Feedwater to reactor
$W_{f\ell}$: " " " feedwater pumps
W_{fp} : " " through one pump
W_{con} : " " from condensate pumps
W_{cp} : Condensate through one pump
W_{drain}: Drainage to the feedwater line.

Enthalpies, specific:

h_v : Before regulating valve
h_h : At HP-turbine inlet
h_{ho} : " " " outlet
h_{ti} : At inlet to moisture separator compartment
h_{to} : " " " superheater compartment
h_{ro} : At reheater outlet
h_{ℓ} : At LP-turbine inlet
h_c : At LP-turbine outlet
h_m : For condensate out from reheater primary side
$h_{fv}(i)$: For steam to steam coolers
$h_w(i)$: For drainage out from steam coolers
Δh_h : Enthalpy drop in HP-turbine
Δh_{ℓ} : " " " " LP-turbine

Enthalpies, total:

H_{hp} : HP-turbine inlet volume
H_{rtv} : Moisture separator compartment
H_{rhv} : Superheater compartment
$H_{\ell p}$: LP-turbine inlet volume

Entropies, specific:

S_h : HP-turbine inlet
S_{ℓ} : LP-turbine inlet

Steam qualities:

X_h : HP-turbine inlet
X_{ho} : " " outlet
X_c : LP-turbine outlet

Thermal efficiencies:

γ_h : HP-turbine
γ_ℓ : LP-turbine

Powers:

E_h : HP-turbine
E_ℓ : LP-turbine
E_t : Total for turbine

Temperatures:

T_{to} : In moisture separator compartment
T_{ro} : Outlet from superheater
T_{tr} : Tube wall in superheater
$T_{fv}(i)$: Steam inlet to steam coolers
$T_{tf}(i)$: Tube wall in steam coolers
$T_{fo}(i)$: Feedwater outlet from steam coolers
$T_{fi}(i)$: " " inlet to steam coolers
$T_{td}(i)$: Tube wall in drain coolers
$T_{fdo}(i)$:Feedwater outlet from drain coolers
$T_{fdi}(i)$: " " inlet to drain coolers
$T_{do}(i)$: Drainage outlet from drain coolers
$T_{di}(i)$: " " inlet to drain coolers

Heat capacities:

$C_{tr}(i)$: Tube wall in superheater
$C_{ts}(i)$: " " " steam coolers
$C_{fs}(i)$: Feedwater " " "
$C_{td}(i)$: Tube wall in drain coolers
$C_{fd}(i)$: Feedwater " " "
$C_d(i)$: Drainage in drain coolers

Volumes:

V_h : HP-turbine inlet
V_t : Moisture separator compartment
V_r : Superheater compartment
V_ℓ : LP-turbine inlet.

4. THE STEAM LINE MODEL

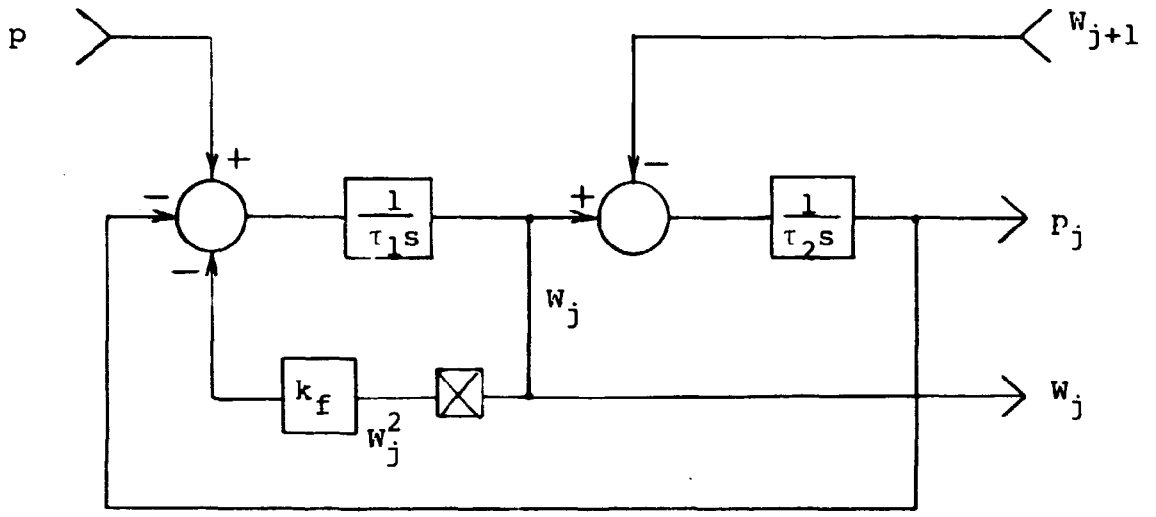
The steam line is a 120 m long tube with a flow area of 0.66 m^2. Such a long line cannot transmit pressure and flow variations immediately, and, what is more important, load variations will cause resonance oscillations with waves running backward and forward along the line. The damping of the oscillations depends on the frictional pressure drop, the higher friction the better the damping.

The model must be constructed by a series of subsection each described by lumped parameters. The number of sections depends on the dimension of the line and the damping. Experiments for the present case have shown that 20 sections give a very good performance, 10 sections is also good, and 5 will still reproduce the main characteristic. As a detailed simulation of the line itself is outside the scope of the present work, 6 sections have been chosen.

The single section is described by mass- and momentum balance as shown in the diagram at Fig. 7. The equations are:

$$\begin{aligned} \dot{W}(j) &= (p(j-1) - p(j) - k_f W(j)^2)/(\tfrac{\Delta L}{A}\ 10^{-5}) \\ p(j) &= (W(j) - W(j+1) - \dot{m})/(A\Delta L \tfrac{\partial \rho_g}{\partial p}) \end{aligned} \qquad (4.1)$$

where



$$\tau_1 = \frac{\Delta L}{A} 10^{-5} \qquad \tau_2 = \Delta V\left(\frac{\partial \rho_{gs}}{\partial p} + \frac{\rho_{gs}}{h_{fg}} \frac{\partial h_{gs}}{\partial p}\right)$$

Fig. 7. Steam line section computation diagram.

$$\dot{m} = A\Delta L \frac{\rho_{gs}}{h_{fg}} \frac{\partial h_{gs}}{\partial p} \dot{p}(j) \qquad \text{for wet steam}$$

and $\dot{m} = 0$ for superheated steam.

The energy balance could have been used to calculate the enthalpy but as explained in Chapter 3.2 it is not used because the outlet value is set equal to the inlet value.

It is assumed that the steam is near saturation with a small fraction of humidity. That is reasonable as the steam from the reactor fulfils this condition and the transit time for a steam particle is so short that it remains in the wet condition for rather fast pressure excursions. This means that the pressure equation in (4.1) can be simplified as:

$$\dot{p}(j) = (W(j) - W(j+1))/A\Delta L\rho' \quad , \quad \rho' = \frac{\partial \rho_{gs}}{\partial p_s} + \frac{\rho_{gs}}{h_{fg}} \frac{\partial h_{gs}}{\partial p_s} \qquad (4.2)$$

The parameter ρ' has two terms where the first is 0.57 ± 0.02 kg/m^3 bar and the last about -0.03 kg/m^3 bar in the pressure range 55 to 70 bar. ρ' is consequently set equal to 0.54 kg/m^3 bar for all conditions. The validity of the above-mentioned assumption of wet steam condition can be evaluated by comparison with the value for ρ', which for slightly superheated steam in the same pressure range would be 0.50 kg/m^3 bar.

A more detailed model should involve the energy balance which together with the mass and momentum balance should give the mass flow, specific density and enthalpy. The pressure should then be found by an iterative calculation. The procedure that has been selected here reduces the number of state variables, avoids the time consuming iterative calculation, and still gives the main characteristics for the steam line.

Symbols for the steam line model

W : Mass flow
p : Pressure
A : Cross section for the steam line
ΔL: Length of one section
k_f: Friction coefficient.

5. THE ATOMATIC CONTROL SYSTEMS

The model incorporates three main control loops: the reactor re-circulation control, the feedwater control and the pressure control.

Details of the control circuits are confidential and cannot be given here, so the controllers are described as "black boxes" in general terms.

The control circuits incorporate protection devices that change the control mode for reactor trip (SS), turbine trip (TS), loss of electrical load (LR) and forced power reduction (NK).

5.1. The reactor recirculation control

A block diagram of the system is shown in Fig.8. For normal operation it has one input, the electrical power. The controller is of the proportional + integral type. To avoid fast variation in the pump speed the driving signal is passed through a step and ramp limit follow up circuit. The limits are dependent of the state of the plant.

For extreme conditions as given by the logic signals TS, LR and NK the pump speed is reduced to a minimum value and locked to that value.

The dynamics of the pump servo is approximated by a single lag function.

5.2. The feedwater control

This control circuit is more complicated with four analogue input signals as shown in Fig. 9. The signals are combined partly in a nonlinear way using several dynamic compensation circuits and with signal limitations, of which one has a variable limit, namely the upper speed limit. The two main paths are the flow error signal (steam flow-feedwater flow) and the reactor water level. They are added and corrected with smaller signals from steam flow and pressure to give the output signal. Reactor trip and low pressure behind the condensate pumps will decrease the upper limit for the pump speed and force the feedwater flow down to a low level.

The dynamics of the pump servo is approximated by a double lag function.

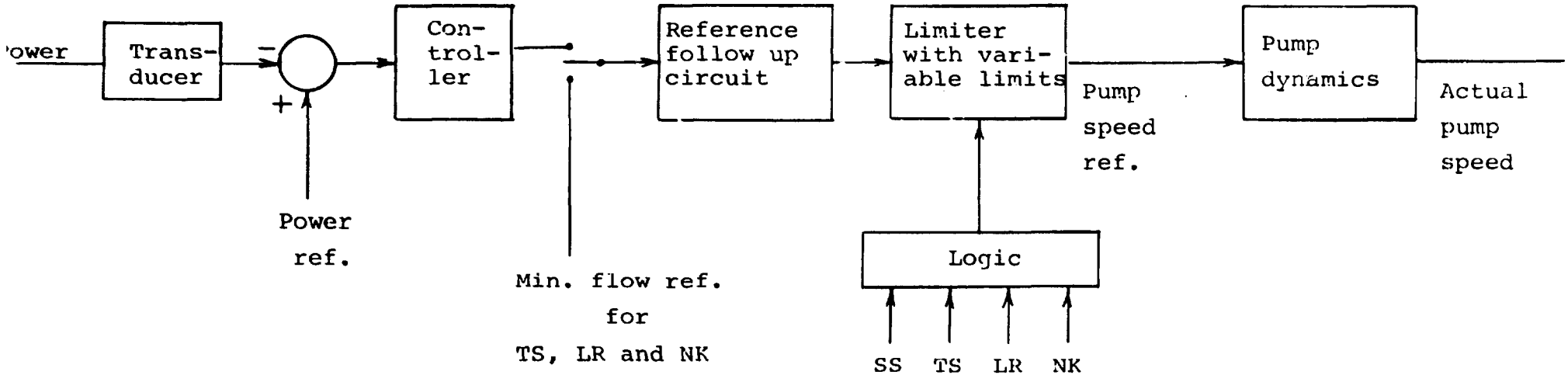
5.3. The pressure control

The pressure control circuit is the most complicated of the three circuits. The main function for normal operation is to control the steam pressure before the turbine using the regulating valve as actuator. The intercept LP-valve is then fully open and the bypass valve fully closed.

The regulating valve movements are limited by a step and ramp limiter with nonlinear performance for open and close operation.

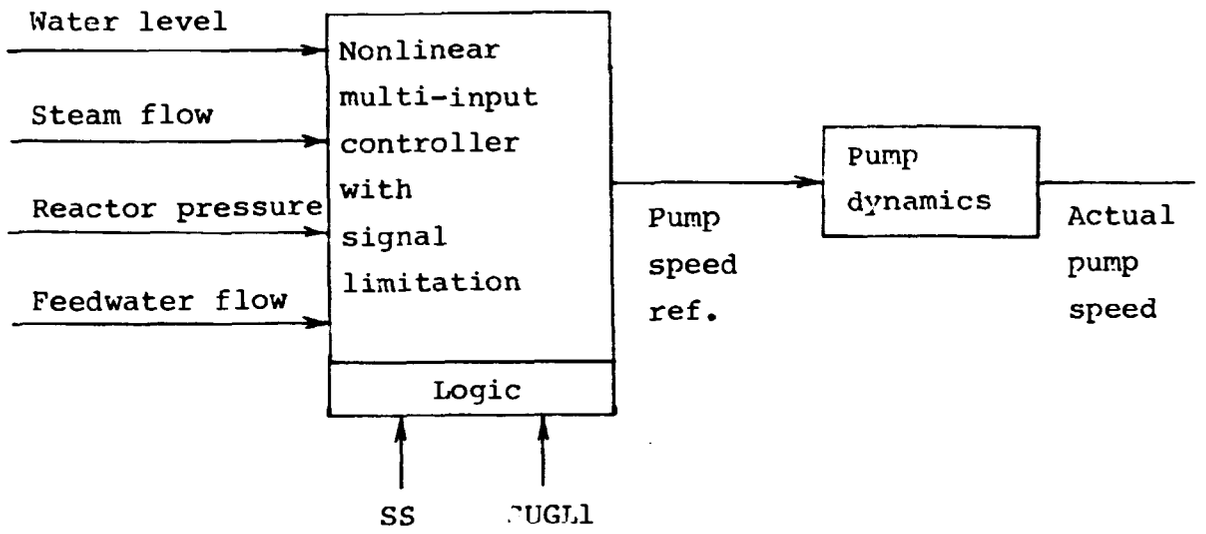
The turbine speed input is normally not in action; but an overspeed signal will close the regulating valve and open the bypass valve which then is used for the pressure control. This shift in operation mode is accomplised by minimum and maximum selection circuits. The same happens if a turbine trip occurs.

The intercept LP-valve will remain open until the regulating valve is nearly closed, and it will not close completely but remain a few per cent open.



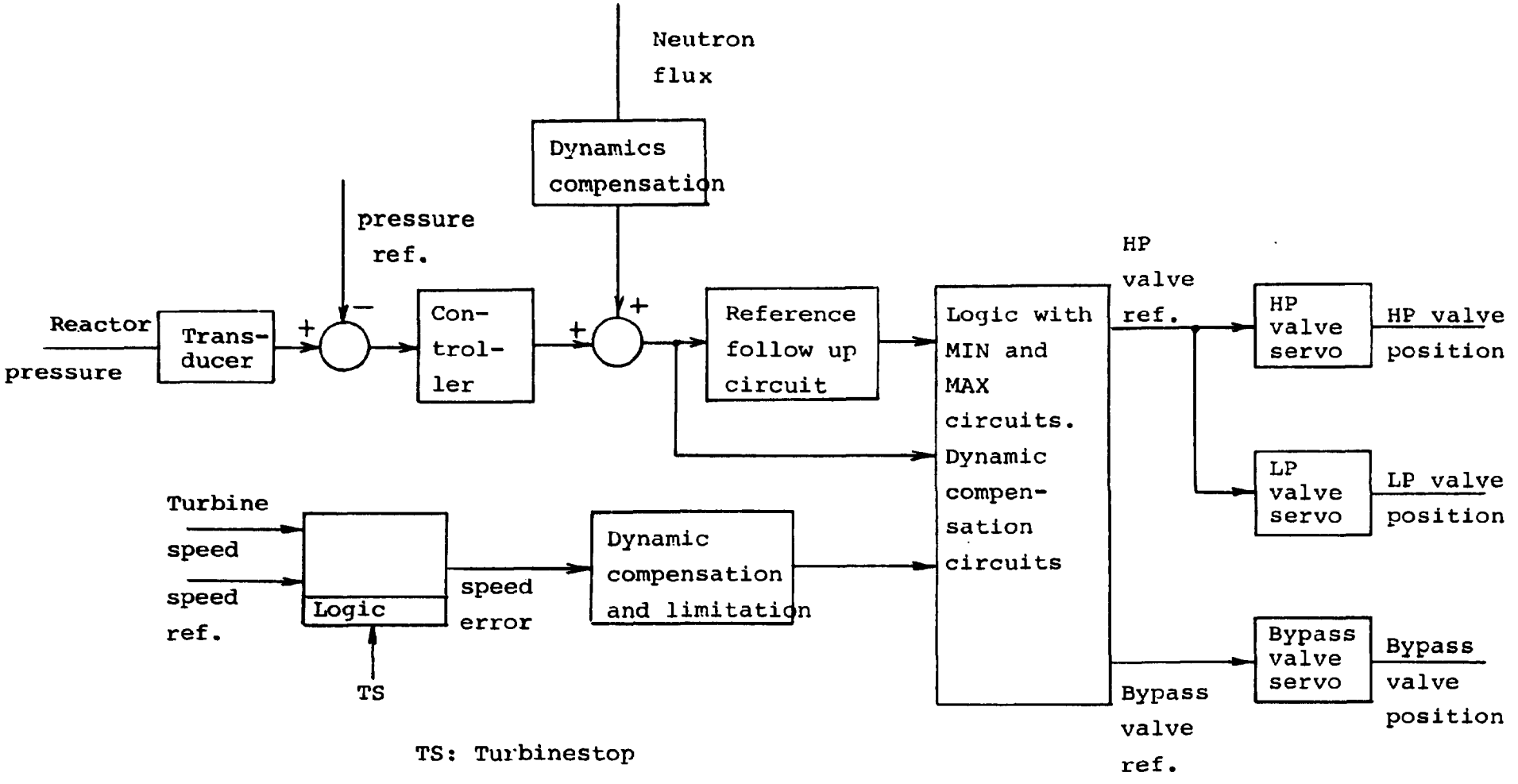
SS: Reactor trip
TS: Turbine trip
LR: Power reduction at loss of load
NK: RC-pump run down

Fig. 8. Reactor recirculation control system.



SS : Reactor trip
SUGL1: Low pressure from condensate pump.

Fig. 9. Feedwater control system.



TS: Turbinestop

Fig. 10. Pressure control circuit.

REFERENCES

CHRISTENSEN, P. La Cour (November 1974). Description of the Real Time PWR Power Plant Model PWR-PLASIM, Risø Report No. 318.

NONBØL, E. (December 1975). Development of a Dynamic Model of a BWR Nuclear Power Plant, Part I and II, Risø Report No. 335 and 336.

BAKSTAD, P. and SOLBERG, K.O. (August 1967). A Model for the Dynamics of Nuclear Reactors with Boiling Coolant with a new Approach to the Vapour Generating Process, Kjeller Report KR-121.

BABALA, D., BECH, N., and HAUGSET, K. (August 1970). ANDYCAP 1970, Vol. I - Theory, Scandinavian Reactor Dynamics Project, Report SRD-25.

SCHLECTHENDAHL, E.G. (July 1970). DYSYS - A Dynamic System Simulator for Continuous and Discrete Changes of State, Institut für Reaktorentwicklung, Karlsruhe, KFK-1209.

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The supplying of data by Sten Bergman from A/B Sydkraft in Sweden is greatly appreciated. These data have been fundamental for the model presented here.

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APPENDIX A

Static temperature relations for idealized steam and drain coolers

Steam cooler

The steam cooler is assumed to have the ideal configuration shown in Fig. A1.

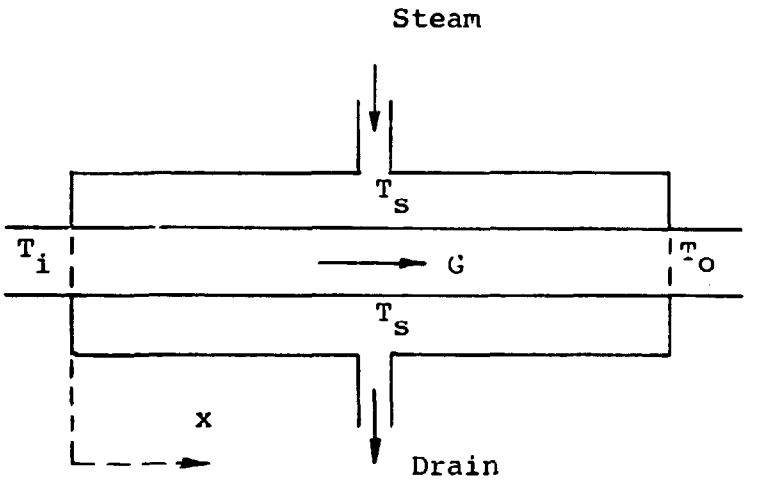

Fig. A1. Steam cooler configuration.

The steam temperature is assumed to be constant over space and the water flow G uniform over the tube cross section. The heat transfer across the tube wall is determined by a constant coefficient k per meter over the total length L meter.

The energy balance for a small section dx with a temperature increase dT gives:

$$Gc_p dT = k(T_s - T)dx \qquad (A.1)$$

which can be solved to give the known input-output relation:

$$\ln \frac{T_s-T_i}{T_s-T_o} = \frac{kL}{Gc_p} \qquad (A.2)$$

or an equation for the temperature T(x) at the distance x:

$$T(x) = T_s(1-E_x) + T_iE_x \quad , \quad E_x = \exp(-ax) \quad , \quad a = \frac{k}{Gc_p} \qquad (A.3)$$

The temperature T(x) can be expressed in several forms involving T_s, T_i and T_o; one of them is:

$$T(x) = T_i + (T_o-T_i)\frac{1-E_x}{1-E} \quad , \quad E = \exp(-aL) \qquad (A.4)$$

The mean value of T(x) is found by integration of eq. (A.4) and division with L. The result is:

$$T_m = T_i + \frac{T_o-T_i}{1-E}\left(1-\frac{1}{aL}(1-E)\right) \qquad (A.5)$$

which can be written in the following form giving T_m as a weighted mean value between input and output:

$$T_m = BT_i + (1-B)T_o \quad , \quad B = \frac{1}{aL} - \frac{E}{1-E} \qquad (A.6)$$

The total heat transfer can then be calculated as:

$$Q = kL\ (T_s-T_m) \qquad (A.7)$$

Static data has been used to calculate the coefficient kL from eq. (A.2). It appears to be fairly constant in the normal working range, so the only variable in the expression for B is the flow G.

Drain cooler

The drain cooler is assumed to have the ideal configuration shown in Fig. A2.

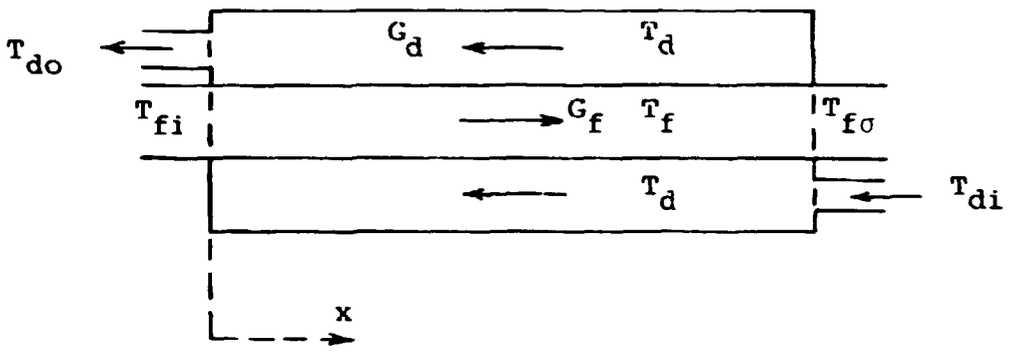

Fig. A2. Drain cooler configuration.

The water flows are assumed to be uniform over the cross sections. The heat transfer across the tube is determined by a constant coefficient k per meter over the total length L meter. The energy balance gives for the two flows:

$$\frac{dT_f}{dx} = \frac{k}{G_f c_p}(T_d - T_f)$$

$$\frac{dT_d}{dx} = \frac{k}{G_d c_p}(T_d - T_f) \qquad \text{(A.8)}$$

In the following derivation three short notations will be used:

$$\frac{k}{G_f c_p} = a_f \quad , \quad \frac{k}{G_d c_p} = a_d \quad \text{and} \quad a_f - a_d = a \qquad \text{(A.9)}$$

and furthermore

$$E_x = \exp(-ax) \quad \text{and} \quad E = \exp(-aL)$$

Eq. (A.8) is a set of coupled first-order differential equations:

$$\frac{dT_f}{dx} + a_f T_f - a_f T_d = 0$$

$$\frac{dT_d}{dx} + a_d T_f - a_d T_d = 0 \qquad \text{(A.10)}$$

The solution to eq. (A.10) can be found in mathematical textbooks. The equations are first transformed to another set:

$$\frac{d}{dx}(T_f - T_d) + a(T_f - T_d) = 0$$

$$\frac{d}{dx}(a_d T_f - a_f T_d) = 0 \qquad \text{(A.11)}$$

The solution to eq. (A.11) is:

$$T_f - T_d = \frac{C_1}{a_f - a_d} \exp(-ax)$$

$$a_d T_f - a_f T_d = C_2 \qquad \text{(A.12)}$$

which gives the solutions for T_f and T_d:

$$T_f(x) = \frac{1}{a}\left(\frac{a_f}{a} C_1 E_x - C_2\right)$$

$$T_d(x) = \frac{1}{a}\left(\frac{a_d}{a} C_1 E_x - C_2\right) \qquad \text{(A.13)}$$

The two integration constants C_1 and C_2 can be found from the boundary value conditions:

$$C_1 = \frac{a^2}{a_f - a_d E}(T_{fi} - T_{di})$$

$$C_2 = a \frac{a_d E T_{fi} - a_f T_{di}}{a_f - a_d E} \qquad \text{(A.14)}$$

Introduction of (A.14) into (A.13) leads to:

$$T_f(x) = T_{fi} + (T_{fo} - T_{fi}) \frac{1 - E_x}{1 - E}$$

$$T_d(x) = T_{di} - (T_{do} - T_{di}) \frac{E - E_x}{1 - E} \qquad \text{(A.15)}$$

Eq. (A.15) has a form similar to (A.4) for the steam cooler, but the parameters E_x and E stand for different exponentials.

The mean value of T_f and T_d is calculated as for the steam cooler:

$$T_{fm} = T_{fi} + \frac{T_{fo}-T_{fi}}{1-E} \left(1-\frac{1}{aL}(1-E)\right)$$

$$T_{dm} = T_{di} - \frac{T_{do}-T_{di}}{1-E} \left(E-\frac{1}{aL}(1-E)\right) \qquad (A.16)$$

It is seen that the temperature mean values can be expressed in a similar way as for the steam cooler:

$$T_{fm} = BT_{fi} + (1-B)T_{fo}$$

$$T_{dm} = (1-B)T_{di} + BT_{do} \qquad (A.17)$$

$$B = \frac{1}{aL} - \frac{E}{1-E} \quad , \quad a = \frac{k}{c_p}\left(\frac{1}{G_f} - \frac{1}{G_d}\right) \quad , \quad E = \exp(-aL)$$

The total heat flow can be expressed as:

$$Q = kL(T_d-T_f)_m = kL(T_{dm}-T_{fm}) \qquad (A.18)$$

The fundamental equations (A.8) can be used to find another relation between inlet and outlet temperatures similar to (A.2):

$$\ln \frac{T_{di}-T_{fo}}{T_{do}-T_{fi}} = - aL \qquad (A.19)$$

which is used to find aL for a set of static temperature values and, as G_d and G_f are known, kL can be found. Also here k appears to be fairly constant.

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DESCRIPTION OF THE POWER PLANT MODEL BWR-PLASIM OUTLINED FOR THE BARSEBÄCK 2 PLANT

BY

P. la Cour Christensen

Date November 1979

Department or group

Department of Reactor Technology

Group's own registration number(s)

pages + tables + illustrations

Abstract A description is given of a BWR power plant model outlined for the Barsebäck 2 plant with data placed at our disposal by the Swedish Power Company Sydkraft A/B.

The basic equations are derived and simplifications discussed.

The model is implemented with a simulation system DYSYS which assures reliable solutions and easy programming. Emphasis has been placed on the models versatility and flexibility so new features are easy to incorporate. The model may be used for transient calculations for both normal plant conditions and for abnormal occurrencies as well as for control system studies.

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