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# SIMULATION OF THE DYNAMIC BEHAVIOUR OF THE SECONDARY CIRCUIT OF A WWER-440 TYPE NUCLEAR POWER PLANT

PART 1

*Hungarian Academy of Sciences*

**CENTRAL RESEARCH INSTITUTE FOR PHYSICS**

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ABSTRACT

This report describes the simulation model of the secondary circuit of a WWER-440 type Nuclear Power Plant. The goal of this modelling is to simulate normal and small abnormal transients in a Basic Principles Simulator. This paper continues the dynamic simulation of primary circuit of a WWER-440 Nuclear Power Plant /Reports KFKI-1983-127 and KFKI-1985-08/. At present the controllers of the secondary circuit are not simulated. In the end of the report some simulation results are presented.

А. Гач, Я.Ш. Яноши, Ж. Киш: Симуляция динамики второго контура АЭС типа ВВЭР-440. Часть 1. KFKI-1987-44/G

АННОТАЦИЯ

В отчете описана модель, которая служит для симуляции динамики второго контура АЭС с реакторами типа ВВЭР-440. Задачей модели является симулирование нестационарных процессов в номинальных режимах и при небольших авариях в принципиальном симуляторе. Данная работа - продолжение отчетов KFKI-1983-127 и KFKI-1985-08, в которых описывается модель первого контура АЭС с реакторами типа ВВЭР-440. Актуальное состояние модели не содержит регулирующие органы второго контура. В отчете даются некоторые симуляционные результаты.

Gács A., Jánosy J.S., Kiss Zs.: WWER-440 tipusu atomerőmű szekunderköre dinamikus tulajdonságainak szimulációja. I. rész. KFKI-1987-44/G

KIVONAT

Ez a riport egy VVER-440 tipusu atomerőmű szekunderkörének szimulációs modelljét irja le. A modellezés célja üzemi és kis üzemzavari tranziensek szimulálása egy Alapelvi Szimulátorban. A dolgozat folytatása a VVER-440 tipusu atomerőművek primerköri modelljeit leiró riportoknak (KFKI-1983-127 és KFKI-1985-08). Jelenleg a szekunderkör szabályozóit nem szimuláljuk. A riport végén néhány szimulációs eredményt közlünk.

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## NOMENCLATURE

The variables used in this report are listed below. For technical reasons Greek lettering has been replaced by Latin /e.g. *e* for eta, *d* for delta, *r* for ro, *z* for zeta/, and some letters need to be written in a more easily printable form /e.g. *G* for m dot, *W* for Q dot/:

*a* - heat transfer coefficient, $W/m^2/K$
*b* - rise of the parabola /efficiency - exhaust steam flow function/, $s^2/kg^2$
*c* - specific heat, J/kg/K
*C* - speed, m/s
*d* derivate of a function
*D* - difference of values
*e* - efficiency, -
*E* - energy, J
*F* - area, surface, open cross-section, $m^2$
*G* - steam flow, kg/s
*h* - specific enthalpy, J/kg
*k* - adiabatic exponent of the gas, -
*L* - work, J
*m* - mass, kg
*p* - pressure, bar
*P* - power, W
*r* - specific density, $kg/m^3$
*s* - specific entropy, J/kg/K
*S* - Stodola number, $kg^2/bar^2/s^2$
*t* - temperature /relative/, $^oC$
*T* - time, s
*u* - various constants
*v* - " "
*V* - volume, $m^3$
*w* - constants for filters
*W* - heat flow, W
*x* - quality /steam-content/, -

Indices are explained by figures within the text; there is no overall indexing scheme but the indices are given for each point independently.

## 1. INTRODUCTION

The construction of dynamic models of a WWER-440 nuclear power plant has been under development in the Central Research Institute for Physics for several years. Up till now the dynamic behaviour of the primary circuit has been investigated and reported [2,3]; here, we describe the dynamic models of the secondary circuit.

The goal of our simulation project is to study the normal and close-to-normal states of the secondary circuit in order to construct models of a basic principles simulator of WWER-440 nuclear power plants. In this paper we present models of the main technological units of the secondary circuit. This model-development work is not yet complete and in Part 2 of this report models of the controllers and that of the electrical generator will be described.

Since 197o we have become relatively experienced in constructing steady state models of the secondary circuits of conventional power plants [1]. Recently a demand has emerged to study the dynamic behaviour of the secondary circuits of power plants and to optimize the transient processes. There are three reasons which support this need:

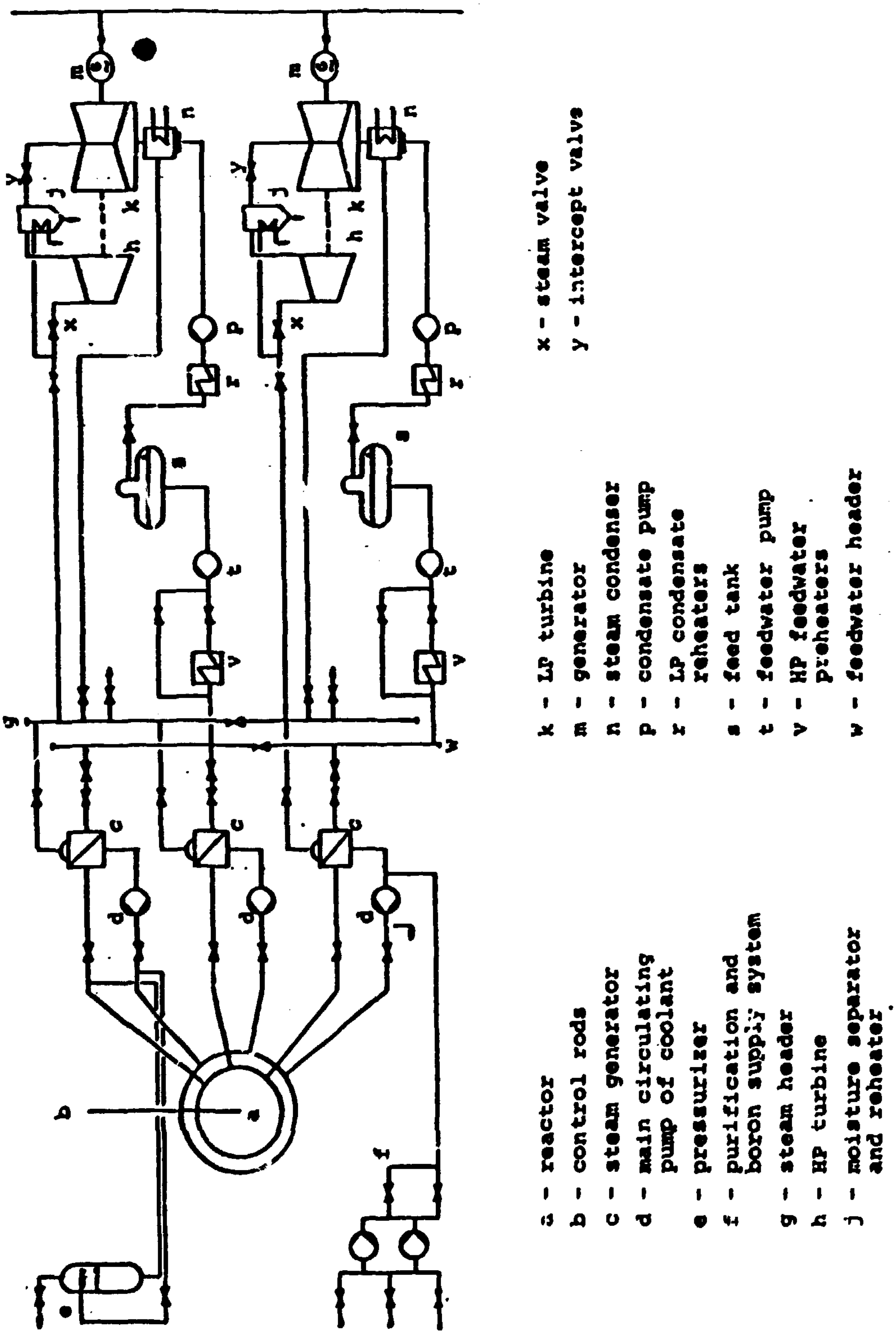
- the new technology of nuclear steam generation,
- the strict safety requirements,
- the very high investment costs.

During the development of the models of the secondary circuit our endeavour was to construct mathematical models which can be used to optimize the control systems of nuclear power plants. This goal defines the validity limits of our modelling approach: the process dynamics is only approximated but the transient characteristics are described correctly. We have constructed a modular system in which the components represent the main physical processes separated from each other by the weakest possible coupling. The coupling paramaters are selected from measurable ones in order to use real transient data for tuning the models.

The scope of the simulation of the WWER-440 basic principles simulator is shown in Fig.1.

In Fig.2 the corresponding model structure is given with the coupling parameters connecting the submodels to each other. In this report the following submodels are presented /see Part A of Fig.2 /:

- turbine model,
- moisture separator and reheater model,
- heater models,
- condenser and condensate pump model,
- deaerator, feedwater tank and pump model,
- check valve model.




<u>Fig.1</u> The Scope of the Simulation

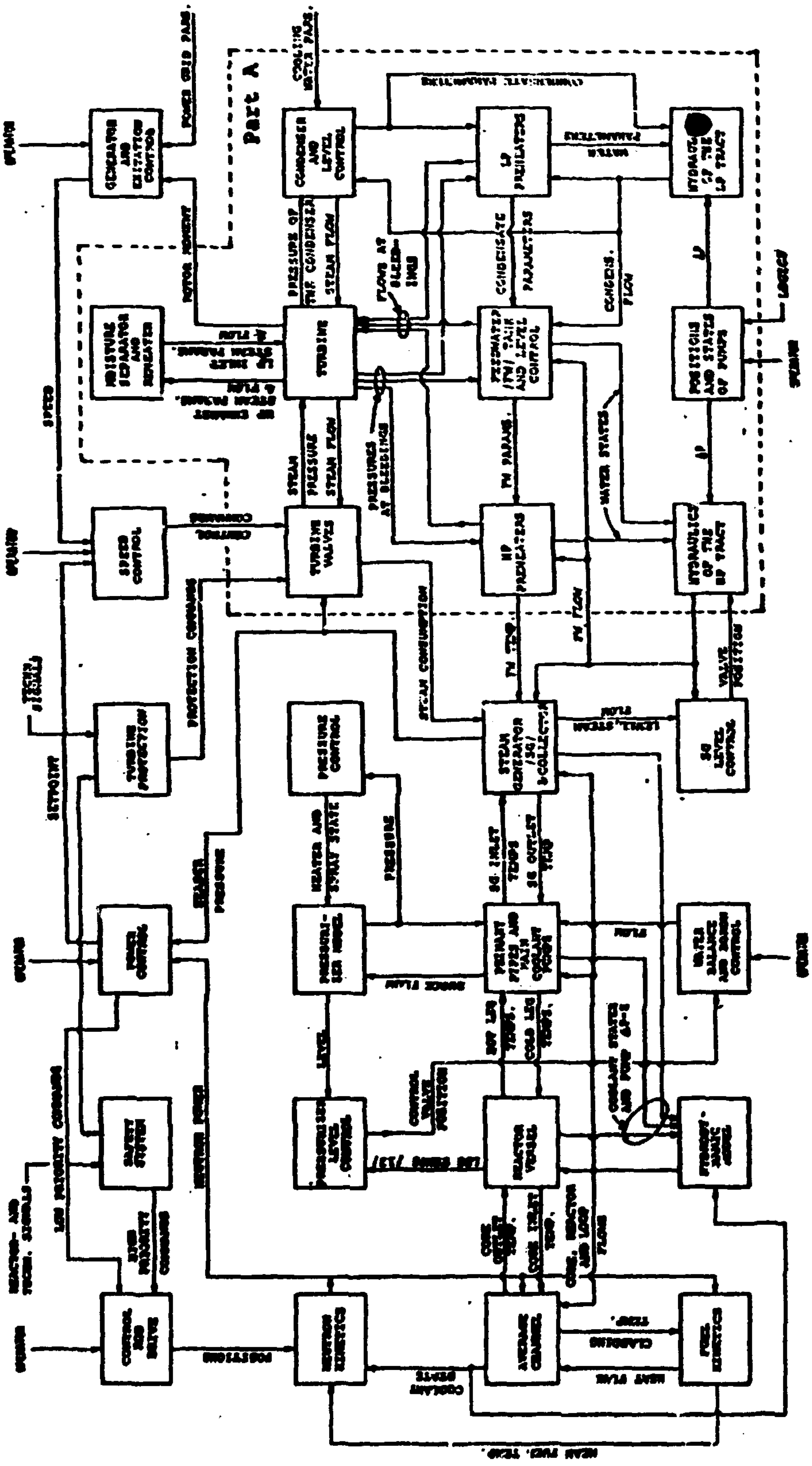



Fig.2 The Modular Model Structure

## 2. MODELLING THE SECONDARY CIRCUIT

Steady state models of the secondary circuit were elaborated at a high level for design and construction purposes [1] and a great deal of effort was devoted to making dynamic models of the primary circuit [2,3] as - in relation to safety aspects - the reactor part is the most important. While these were key questions, there was no great need for dynamic models of the turbine, the moisture separator and reheater unit, the condensate- and feedwater-heaters; nor was very much attention paid to the other parts of the secondary circuit either.

One reason for this could be that these were considered to be components known from conventional power plants. Nevertheless, the secondary circuit of a nuclear power plant is fundamentally different from a conventional one, e.g. the turbine operates with steam at a much lower temperature and greater water content therefore it is not possible to simply apply the results from conventional turbines. In addition, experience has shown that on the one hand many of the shutdowns of nuclear power plants are due to problems with turbines and other conventional parts, and on the other hand proper modelling of safety systems and that of the turbo-generator unit might help to prevent serious accidents [4-8].

The development of the models of the secondary circuit was started in 1984, as a continuation of the modelling activity on the primary circuit.

The building of a modular structure is a reasonable approach [9] , but it was not our intention to develop a new simulation language for general use. Instead, we have continued to use of the well-proved code FORSIM [lo,11]. This latter is a general simulation package for the automatic integration of coupled sets of ordinary differential equations, and partial differential equations in one and two spatial dimensions. Therefore the secondary circuit model is described by a number of subroutines and the FORSIM system then automatically integrates the equations with changing time steps and by the chosen integration method. For this, a version of FORSIM-5M is adapted and improved, as a simulation tool [12].

Another tool had to be the calculation package of the properties of water and steam. There are various skeleton table libraries for several materials and for general use. MAPLIB is one of the best known packages [13,14,15]. Instead of using such a widespread but complex system, we developed a compact but accurate code for calculating the thermodynamic properties of water and steam [16], based on the revised version of the internationally recognized 1967 IFC formulation [17]. This is used for the rapid and accurate approximation technique described in Chapter 3.

As for the model itself, we first examined various representations of models. Some of them do not represent details of thermodynamic transients; non-linear and linearized models are combined; some models are valid for slow transients and/or provide full-load, design-point calculations only; and, in general, different degrees of sophistication are involved [2o,26, 27,28].

Having investigated dynamic models implemented on hybrid computers [29,3o,31], our studies showed that extensive simplification is possible utilizing submodels with limited complexity and even by means of equations reduced to allow calculations performed by hybrid computers, and the careful use of lumped parameters enables accuracy and stability to be retained.

At the beginning of the elaboration of the submodels of the secondary circuit, such reasons motivated the decision to start extensive reserach, so at present most component-models have several versions complying with requirements of different depths [18].
The most important results are given in Points 2.1. to 2.4.

There is another goal for using two models for each equipment: it can be presumed that the initial state of simulation is a stationary one so it can be calculated by steady state models setting time derivates by zero. Dynamic simulation can be started from steady state only. Thus two models are necessary:

- a steady state model: initial values have to be calculated for all the integral variables before starting the simulation, presuming that there are no changes in these values so all the first order time derivates are zero;
- a dynamic model: this uses the initial values, and the disturbances as functions of time.

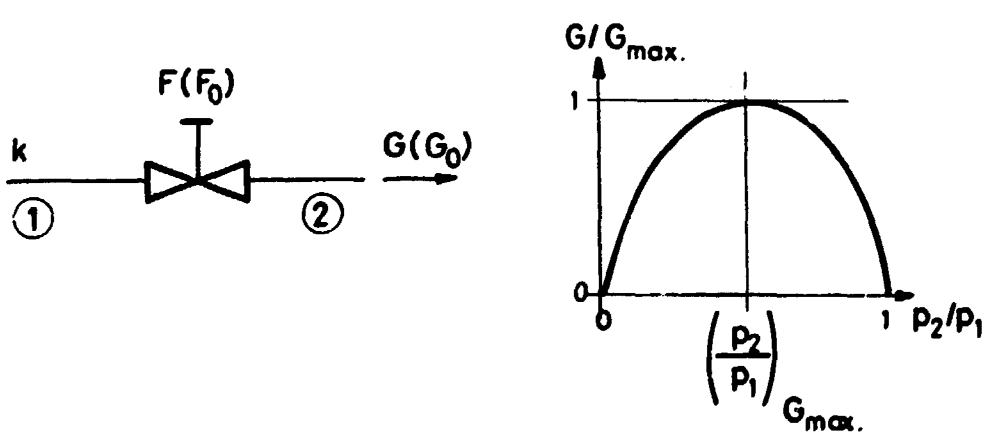
Therefore for each component shown in Fig.2, there are two submodels: a steady-state and a dynamic one, e.g. an LP turbine submodel used in the steady state code and another one for the dynamic code. Consequently, in Points 2.1. to 2.4., the assumptions and solutions are discussed relating to the components and their use in the submodels. The tunable parameters available are for adjusting the model to the measured real transients. These and the simulated parameters are all mentioned in this Chapter.

## 2.1. Modelling the turbines

Our first turbine model contained both the HP and LP turbines and the moisture separator - reheater unit. It comprised 69 equations, 18 of which were differential. Due to the large size, it was too complicated for studying the effects of each parameter one by one. The next, complex model was simplified to a series of point models set up stage by stage. This was convenient for investigating the stiff character of the equations, i.e. the time constants of the simulated processes differ from each other by many orders of magnitude /e.g. the steam-flow through the turbine in contrast to the temperature-change of the casing/. The result of this analysis made it possible to simplify the turbine model again. Because of the limited input-output facilities provided by the control desk of the simulator [19] and to comply with the requirements of short execution time, the present model is quite compact and the separate HP and LP submodels are point-models of concentrated parameters. The required accuracy is achieved by the proper fitting of the tunable parameters. The energy, continuity and moment equations and the appropriate assumptions are discussed in Points 2.1.1.-2.1.3.

2.1.1. Steam Valve

The steam consumption of the turbines is controlled by the steam valve /a set of control nozzles, see Fig.3, cf. element marked "$x$" in Fig.1/. A simple approach [25] is acceptable.




the indices: 0 - stationary state
1 - inlet side
2 - outlet side

Fig.3 Steam Valve

The mass flow of steam through this valve is considered as a subcritical /subsonic/ flow [2o].
Neither tunable parameters, nor simulated ones are needed for this model.
Assuming that the expansion is adiabatic, the work $L$ of the unit mass $m$ is:

$$\frac{L}{m} = \frac{k}{k-1}\left(\frac{p_1}{r_1} - \frac{p_2}{r_2}\right) \qquad /2.1.1.-1./$$

but this decrease of the thermal energy is also equal to the increase of the kinetic energy:

$$\frac{L}{m} = \frac{c_2^2}{2} - \frac{c_1^2}{2} \qquad /2.1.1.-2./$$

Accepting that the speed of steam at the inlet is negligible compared with that at the outlet, the latter can be expressed; then, by substituting it in the following equation of the steam flow $G$ through the area $F$ /the open-section of the valve/ we get:

$$G = F\, c_2\, r_2 = \ldots = F\sqrt{2\,\frac{k}{k-1}\,p_1\,r_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]}$$

/2.1.1.-3./

When $F, k$ and $c_2$ are constants, the steam flow is a function of the pressure-ratio $p_2/p_1$ only. As is shown in Fig.3, this flow has a maximum: the critical /sonic/ flow. From Eq. /2.1.1.-3./ we get:

$$\left(\frac{p_2}{p_1}\right)_{G_{max}} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \qquad /2.1.1.-4./$$

Equations /2.1.1.-3. and - 4./ are used in the steady-state model to get a stationary cross-section $F_0$ from the given steam-consumption $G$ and steam parameters $/p_1, v_1; p_2/$ .

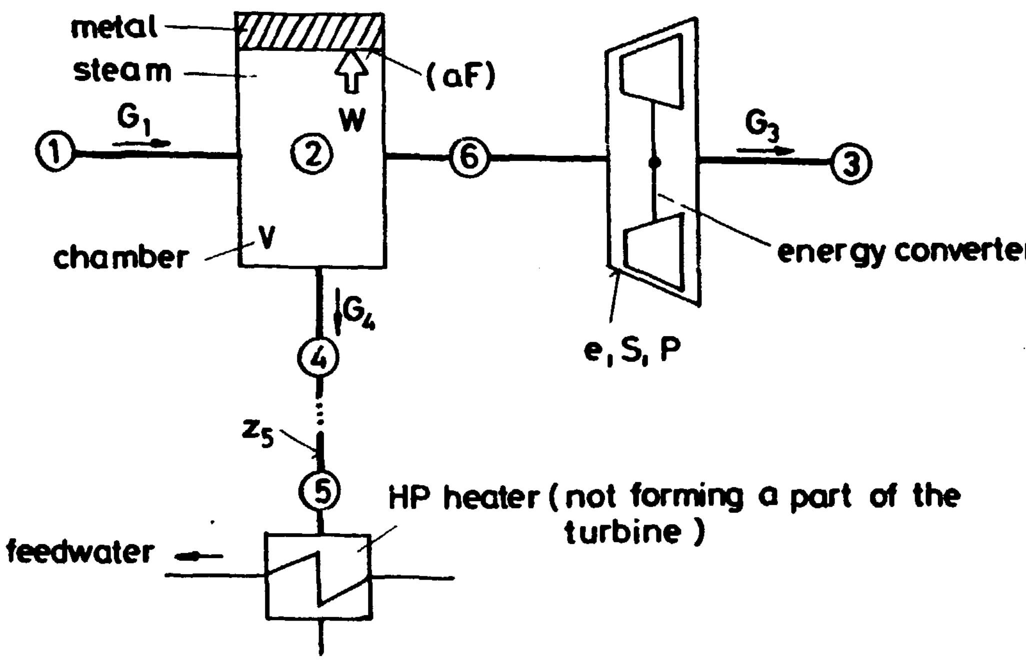
In the dynamic model, this $F_0$ may be changed by giving closing-opening commands to the steam valve from the switchboard. Taking into account other disturbances /steam parameters, etc./ as well, the proper inlet steam flow can be calculated.

## 2.1.2. HP Turbine

The point-model of the HP turbine /see. Fig.4, cf. element marked "$h$" in Fig.1/ represents all its stages and has two parts: one is a chamber for storing and distributing inlet steam, the other is an energy converter, assuming the blading /nozzle-and blade-rows/ of the stages as one unit. Bleed steam is led to the HP heater /cf. element marked "$v$" in Fig.1/ but this connection is effected by the actual valve positions /bypass/ and flows /deaerator heating/ discussed in Point 2.4.3.

The tunable parameters available for the chamber are:

- its volume $/V_2/$ ;
- the equivalent mass of the wall $/m_w/$ , and the product of the coefficient $/a/$ and the surface $/F/$ of heat transfer from the steam to the wall $/aF/$.




the indices:
0 - nominal load
1 - inlet point
2 - storing chamber
3 - outlet point
4 - bleed steam to the heater
5 - HP heater
6 - intermediate point
*A* - outlet point assuming adiabatic expansion
*w* - wall

Fig.4 HP turbine

The simulated parameters are as follows:

- specific density in the chamber /$r_2$/,
- specific enthalpy in the chamber /$h_2$/,
- temperature of the wall /$t_w$/.

The derivates of the above parameters must be zero in the steady state model, and in the dynamic model they need to be calculated.

The theoretical problem can be divided into two parts. So far as the chamber is concerned, $h_2$ and $r_2$ define the state properties of the steam $/p_2, t_2, s_2, x_2/$. The mass balance equation gives the mass-difference $DG_2$; $G_6$ is equal to $G_3$; and so the first differential equation is :

$$\frac{dr_2}{dT} = \frac{1}{V_2}\frac{dm_2}{dT} = \frac{DG_2}{V_2} = \frac{G_1-G_3-G_4}{V_2} \qquad /2.1.2.-1./$$

Here $G_4$ can be calculated at every time step from:

$$Dp_4 = p_2 - p_5 = z_4 \frac{G_4^2}{r_2} \qquad /2.1.2.-2./$$

where $z_4$ is a constant defined by the steady state /the stationary values of $p_2, p_5, r_5, G_4$ are given/. Because of the check valve built into the extraction pipe and depending on the parameters of the HP heaters and the deaerator, value $G_4$ may have a different value /see Point 2.4.3./.

Assuming isothermic conditions /no heat transfer to or from the wall/ and no bleed flow, the change of energy stored in volume $V_2$ is:

$$\frac{dE}{dT} = \frac{d/m_2h_2/}{dT} = m_2\frac{dh_2}{dT} + h_2\frac{dm_2}{dT} = m_2\frac{dh_2}{dT} + h_2/G_1-G_6/ \qquad /2.1.2.-3./$$

On the other hand:

$$\frac{dE}{dT} = G_1h_1 - G_6h_6 \qquad /2.1.2.-4./$$

but substituting $G_1$ for $G_6$ and $h_2$ for $h_6$ /due to the point model/, the enthalpy of the chamber is a function of the inlet steam parameters only:

$$m_2 \frac{dh_2}{dT} + \emptyset = G_1 /h_1 - h_2/ \qquad /2.1.2.-5./$$

hence:

$$\frac{dh_2}{dT} = \frac{G_1/h_1-h_2/}{V_2 \ r_2} \qquad /2.1.2.-6./$$

The heat transfer from the steam stored in the chamber to the equivalent mass of the wall is described by:

$$W_w = /aF/_w \ /t_2-t_w/ = m_w \ c_w \ \frac{dt_w}{dT} \qquad /2.1.2.-7./$$

where $c_w$ and $/aF/_w$ are assumed as constants and the outer heat insulation of the wall is considered as ideal.

Hence the second differential equation is:

$$\frac{dt_w}{dT} = \frac{/aF/_w \ /t_2-t_w/}{m_w \ c_w} \qquad /2.1.2.-8./$$

In our case, the energy balance is not affected by the bleed flow /if any/, therefore with heat transfer $/W_w/$ the third differential eqvation is:

$$\frac{dh_2}{dT} = G_1 /h_1-h_2/ - W_w \qquad /2.1.2.-9./$$

The amount of heat stored in the metal of the stage is taken into account in Eqs. /2.1.2.-7,-8,-9./, but such effects due to the masses of water drops and of condensed water are neglected /see the modelling of the LP turbine/.

In the dynamic model these three time derivates /Eqs. 2.1.2.-1.,-8.,-9./ are to be calculated for the digital simulation, which provides the actual values of $r_2, t_w$ and $h_2$ . In the steady state model these coefficients are equal to zero, consequently:

$$\left.\begin{array}{l} G_3 = G_1 - G_4 \\ t_w = t_2 \\ \text{/so } W = 0\text{, hence:/} \\ h_2 = h_1 \\ \text{/and } p_1 = p_2\text{, therefore: /} \\ r_1 = r_2 \end{array}\right\} \qquad /2.1.2.\text{-1o.}/$$

This means that states 1 and 2 are identical in the steady state, but are /or may be/ different during transients, see Fig.5.

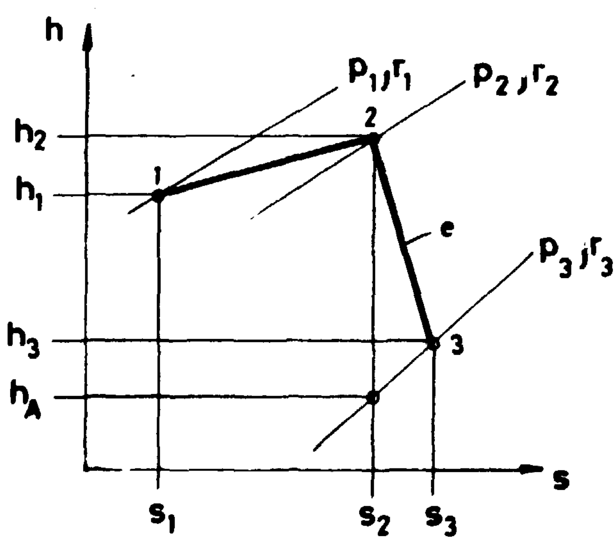



Fig.5 Discontinuity of the Expansion Line of the HP turbine due to the Transients /demonstrated on the entropy-enthalpy surface/

The second part of this theory is the description of the energy conversion. It is assumed that the mass flow of steam through the turbine follows the cone-law equation of Stodola [21] for the performance of stages at varying loads.
Neglecting the effect of the absolute temperature, this law defines the flow as a function of pressures at the intermediate and outlet points, and substituting $p_2$ for $p_6$, we get:

$$S = \frac{G_3^2}{/p_2^2 - p_3^2/} = const. \qquad /2.1.2.-11./$$

The thermodynamic efficiency and the expansion line vary with the flow. The exact calculation of the efficiency would need a detailed analysis of the various losses at the nozzle and blade rows of every stage, and other partial losses as well. However, for our accuracy requirements, it is sufficient to follow the changing conditions, so the dependence of the efficiency on the steam flow is approximated by a curve of second order:

$$e = e_o - b\ /G_o - G_3/^2 \qquad /2.1.2.-12./$$

where index "$o$" refers to the nominal load, and $e_o$, $G_o$ and $b$ are constants, defining the maximum of the efficiency $/e_o/$ at the nominal value $/G_o/$ of the exhaust steam flow $/G_3/$.

The expansion line /see Fig.5/ is defined by:

$$h_3 = h_2 - e/h_2 - h_A/ \qquad /2.1.2.-13./$$

where $h_A$ is the specific enthalpy after adiabatic expansion to the exhaust pressure $p_3$. Hence we can calculate the power output as:

$$P = G_3\ /h_2 - h_3/ \qquad /2.1.2.-14./$$

With regard to energy conversion, the steady state model calculates constant $S$, the dynamic model uses the actual value of $h_2$ from digital simulation.

Here we mention two numerical problems.

A very accurate method /see Point 3/ was essential in fitting two variable surface functions /or their splines/ to the saturated area of the entropy-enthalpy diagram, to get accurate values of the properties of steam, because we need functions of specific density and specific enthalpy for computing pressure and specific entropy. As the execution time of the dynamic model was limited, and here we use function $p/r,h/$ in every time step, it was suitable to use this function also in the steady-state model /where $p$ and $h$ are given, so $r$ was to be approximated successively/ to ensure accuracy as well as to keep the differential coefficients to exactly zero.

The other problem was the slight differences of the enthalpies resulting in unacceptable errors. In view of this a relative scale of enthalpies was introduced.

### 2.1.3. LP Turbine

The point-model of the LP turbine /cf. element marked "$k$" in Fig.1/ is identical with that of the HP turbine /see Point 2.1.2./, as the actual inlet /$G_1$/, bleed /$G_2$/ and exhaust /$G_3$/ flows are doubled and so the two-flow LP turbine is modelled by only one unit which also has different $V_2$, $m_w$, $F_w$, $z_2$, $S$, $b$ and $G_0$ values. Bleed steam is led to the LP heater /cf. element marked "$r$" in Fig.1/. Obviously, using Fig.5, if a point is in the overheated region, the isobars and the iso-densities are not coincidental lines but intersect at the given point.

The results of examinations of our previous models enabled us to assume that:

- the flow through the stages is homogeneous,
- there is always steam in the turbine, i.e. the steam is either overheated /supersaturated, e.g. before the LP turbine/ or undercooled /saturated but not wet, e.g. before the HP turbine/.

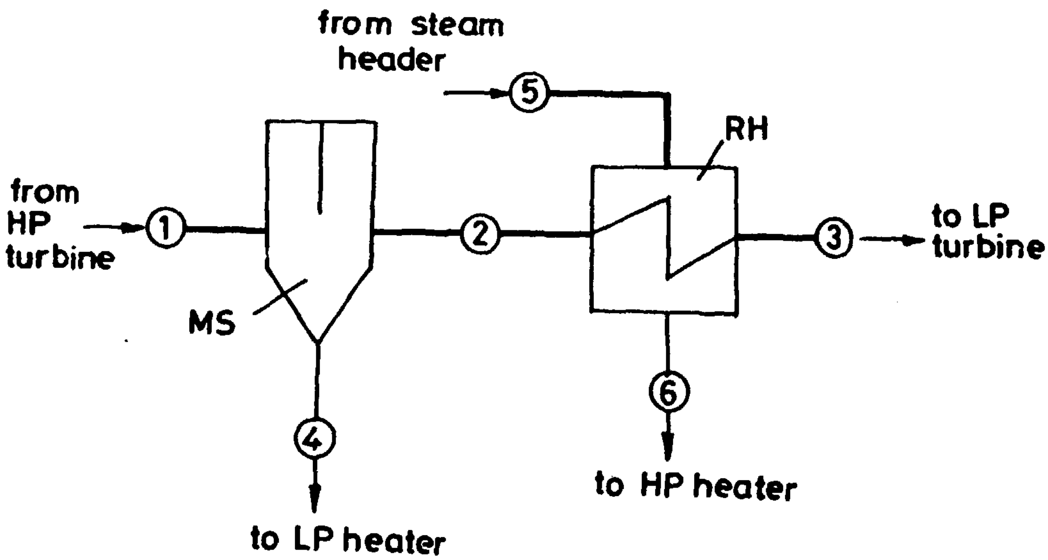
The use of these assumptions even for the exhausts of both turbines is very important. In the LP turbine, expansion starts in the supersaturated region and proceeds into the saturated region. Where the expansion starts, only steam is present. This expansion continues when crossing the saturation line, and if there was thermal equilibrium, steam condensation would start at the crossing of the saturation line and would increase during the expansion in the saturated region,and here would be a two-phase flow of steam and water. However, experiments have shown that there exists a metastable expansion at a great distance below the saturation line, where only steam is present. According to the Wilson theory for the description of this phenomenon [22, 23], we can assume that there is no spontaneous condensation and this makes it possible to neglect the effects of

- the condensing steam,
- the condensate stored in the rows of the stages and
- the vaporization of this condensate in both HP- and LP turbines,

even during transients.

## 2.2. Modelling the Moisture Separator and Reheater Unit

The MSRH unit /moisture separator and reheater, see Fig.6, cf. element marked "*j*" in Fig.1/ is connected between the HP and LP turbines to overheat the working steam and to increase the overall efficiency of the power plant in this way. This unit is built in one structure and consists of an MS and two stages of RH.




the indices: 1 - HP-exhaust

2 - inlet of reheater unit /RH/

3 - LP-inlet

4 - drainage from moisture separator /MS/ to LP condensate heater

5 - heating steam from the main steam header /before control valve/

6 - condensate of RH to HP feed heater

*a* - average

*in* - heat flow into the wall of the tube /outer surface/

*ln* - logarithmic mean value

*h* - heated steam

*out* - heat flow from the wall of the tube /inner surface/

*p* - previous value

*s* - saturation point

*w* - wall

Fig.6 MSRH Unit

The reheater may be modelled as one unit because on the one hand the parameters of the steam between the two units are not needed and on the other hand, the heating steam of Stage 1 /from a bleed line of the HP turbine/ may be included in that of Stage 2, so the calculation of the HP turbine is faster. Another simplification is that the pressure-drop due to the flow-resistance of the whole MSRH unit is concentrated to the MS as a function of flow and density:

$$Dp = p_1 - p_3 = z \frac{G_1^2}{r_1} \qquad /2.2.-1./$$

where $z$ is calculated for the steady state and is kept as a constant in the dynamic model.

If we take the efficiency of the moisture separator to be loo%, the steam flow at the inlet of the RH is dry $/x_2 = 1/$. As the steam quality of the HP exhaust $/x_1/$ can be calculated from state properties $/h_1, p_1/$, the mass balance of the MS is:

$$G_4 = G_1 - x_1 G_2 \qquad /2.2.-2./$$

As for the reheater, it is assumed that the outside heat isolation is ideal, the heating steam is never overheated, its condensate is always saturated /there is no aftercooling/, and the isobaric specific heat of the working steam is constant.

The logarithmic mean temperature is defined by:

$$Dt_{ln} = \frac{t_3 - t_2}{ln\left(\frac{t_5 - t_2}{t_5 - t_3}\right)} \qquad /2.2.-3./$$

In a heat-exchanger even if it is heated by condensing steam, the heat transfer must follow the changing conditions. The type of this RH is a construction with pipes for the heating steam and the heated working steam flows outside. The heat flux is determined by the condensation in the heating pipe, the heat conduction of the wall, and the outside convection between the wall and the working steam. The determinant factor is the last one so the product of the coefficient and the surface of heat transfer is calculated as function of the working steam flow:

$$/aF/ = u + v\ G_2 \qquad /2.2.-4./$$

where $u$ and $v$ are tunable constants.

The heat flow can be expressed by several formulae. In the steady state model, $W$ is obtained form

$$W = G_5\ /h_5 - h_6/ \qquad /2.2.-5./$$

In the dynamic model, the RH unit has a heat storage capacity so the average temperature of the heated working steam may be defined by the difference of the heat flow into and from the tube, and by the heat capacity of the wall and the heated steam. Substituting half of this latter sum of capacities by a constant thermodynamic inertia $I$ [24], we get:

$$\frac{dt_a}{dT} = \frac{1}{2}\left(\frac{dt_{11}}{dT} + \frac{dt_{12}}{dT}\right) = \frac{W_{in}-W_{out}}{m_w c_w + m_h c_h} = \frac{W_{in}-W_{out}}{2I} \qquad /2.2.-6./$$

where $I$ is a tunable parameter and:

$$W_{in} = \left(aF\right)\ Dt_{ln} \qquad /2.2.\ -7./$$

$$W_{out} = G_2\left(h_3 - h_2\right) \qquad /2.2.-8./$$

where

$$G_2 = \frac{W_{in}}{x_5/h''-h'/} \qquad /2.2.-9./$$

Finally, a differential equation is given:

$$\frac{dt_3}{dT} = \frac{W_{in} - W_{out}}{I} - \frac{dt_2}{dT} \qquad /2.2.-10./$$

Preventing possible numerical instabilities resulting from short time steps $/T-T_p/$, the last term is filtered using values from the previous time step: $t_{2p}$ and $dt_{2p}$. This digital filtering is:

$$\frac{dt_2}{dT} = \frac{t_2-t_{2p}}{(T-T_p)} \left[1 - exp\left(\frac{T_p-T}{w}\right)\right] + \frac{dt_{2p}}{dT} exp\left(\frac{T_p-T}{w}\right) \qquad /2.2.-11./$$

where $w$ is a tunable constant.

Other solutions are to be used in the steady state model to ensure accuracy:

- the specific enthaply at the outlet of RH $/h_3/$ is calculated by successive approximation using the same function as used in the dynamic model: $h_3/p_3, t_3/$;
- the product $/aF/$ is determined by two equations: Eq. /2.2.-4./ and Eq. /2.2.-7./ For this reason we have introduced a correction factor as the ratio of the value $/aF/$ from Eq. /2.2.-7./ to the other one and tuning parameters $u$ and $v$ are multiplied by this factor when $G_2$ is calculated.

## 2.3. Modelling the Regenerative Heating

### 2.3.1. LP Condensate Reheaters

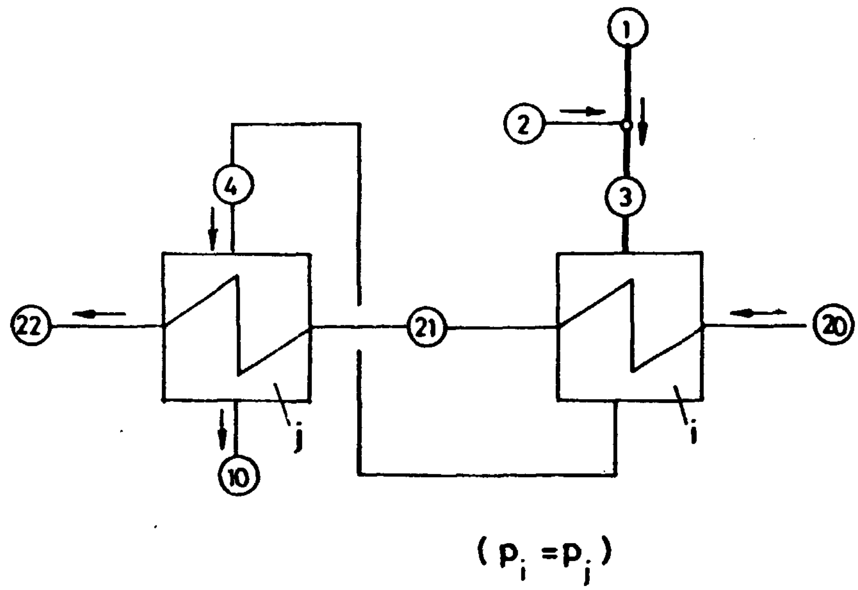
The indices used in this Point /Figs. 7, 8, 9/:

1 - bleed flow from the LP turbine
2 - separated moisture from MS
3 - heating steam mixture
4 - saturated steam from "i"
5 - saturated steam component of inlet heating steam to "j"
6 - saturated water component of inlet heating steam to "j"
7 - condensing steam in "j"
8 - saturated steam stored in "j"
9 - saturated water collected in the hotwell of "j"
1o - saturated water as the drain of "j" to the main steam condenser after the LP turbine
11 - saturated water remaining in the hotwell of "j"

2o - condensate from main condensate pump
21 - partially heated main condensate from "i"
22 - reheater condensate to deaerator
3o - heat flow form overheated steam to main condensate
31 - heat flow from condensing steam
32 - heat flow to main condensate in "j"

c - main condensate
d - drain
f - water
g - steam
i - bleed cooler
j - bleed condenser
p - values from previous time step
w - wall

The regenerative condensate reheating consists of five heat-exchangers, most of them have an aftercooler or a drain pump. It is possible to reduce them to one heater model /see Fig.7, cf. element marked "*r*" in Fig. 1/. The actual construction is of condenser type. A parallel flow model is chosen to avoid several redundant successive approximations.




<u>Fig. 7</u> LP Heater

This is divided into two heat exchangers to follow the real process: first, the overheated heating steam /from LP bleed flow/ is cooled until its saturation and, second, the saturated heating steam is condensed. Therefore, the first is a bleed cooler, the second is a bleed condenser. It is assumed that the separated moisture /from MS/ is mixed with the bleed steam prior to their flow into the heaters.
Further assumptions are, that

- both the separated moisture and the condensate of the heating steam are always saturated water,
- the heating pipes are never below the condensate level,
- the outer heat insulation is ideal.

The theory of the point models of these heaters has three parts:

- a mixer,
- a cooler,
- a condenser of the bleed steam.

Mixing is defined by simple mass- and heat balance equations:

$$G_3 = G_1 + G_2$$

$$h_3 = \frac{G_1 h_1 + G_2 h_2}{G_3} \qquad /2.3.1.-1./$$

In transients, when $h_3$ is not greater than its saturation value, the heating steam is not overheated, so point 3 is identical with point 4 /and the cooler model is "bypassed"/.

As for the bleed cooler "$i$", the logarithmic mean temperature is /cf. Fig.8/:

$$Dt_{ln_i} = \frac{/t_3 - t_{2o}/ - /t_4 - t_{21}/}{ln\left(\frac{t_3 - t_{2o}}{t_4 - t_{21}}\right)} \qquad /2.3.1.-2./$$

and the heat transfer coefficient $/a_i/$ is function of the main condensate flow $/G_{2o}/$:

$$a_i = u_i + v_i G_{2o} \qquad /2.3.1.-3./$$

where $u_i$ and $v_i$ are tunable parameters.

The storage capacity of bleed cooler "$i$" is included in the inertia of bleed condenser "$j$", as the latter is always calculated, but "$i$" may be omitted. Therefore heat flow $W_{3o}$ is defined by various formulae:

$$W_{3o} = G_3/h_3 - h''/ = G_{2o} c_{2o} /t_{21} - t_{2o}/ = a_i F_i Dt_{ln_i} \qquad /2.3.1.-4./$$

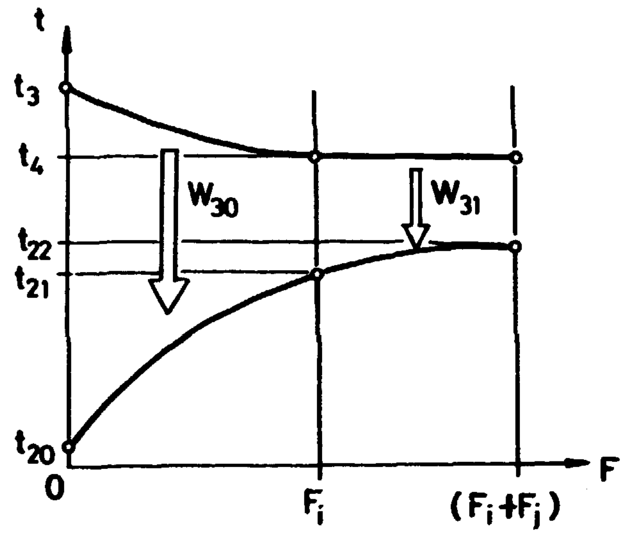



Fig. 8 Temperature-Surface Diagram for the LP Heater Model

Combining the first and second expressions, we can get the intermediate temperature $t_{21}$; and from the first and third, the cooling surface of the heat transfer $/F_i/$ is given.

As for the bleed condenser "$j$", the logarithmic mean tempara-ture is /cf. Fig. 8/:

$$Dt_{ln_j} = \frac{t_{22}-t_{21}}{ln\left(\frac{t_4-t_{21}}{t_4-t_{22}}\right)} \qquad /2.3.1.-5./$$

and the heat transfer coefficient $/a_j/$ is again a function of the heated flow $/G_{2o}/$:

$$a_j = u_j + v_j G_{2o} \qquad /2.3.1.-6./$$

where $u_j$ and $v_j$ are tunable parameters.

The calculation is based on the flow-distribution principle shown in Fig.9.

The quality /steam-content/ of the heating steam $/x_4/$ is defined by its property parameters $/h_4,\ p_i/$, and the steam flow can be divided into saturated steam and saturated water flows:

$$G_5 = x_4 G_4$$

$$G_8 = G_4 - G_5 \qquad /2.3.1.-7./$$

Certainly one of these flows may be zero /depending on $x_4$/as the heating steam may be saturated steam $/x_4=1/$, or the bleed cooler is bypassed if it is saturated water $/x_4=0/$ or a mixture of steam and water.

The total outer surface of the pipe is given $/F/$ and the part needed for cooling is calculated $/F_i/$, so the area available for condensing the heating steam $/F_j/$ is the remainder. This defines the transferable heat:

$$W_{31} = a_j/F-F_i/\ Dt_{ln_j} = G_7\ /h''-h'/ \qquad /2.3.1.-8./$$

and hence the condensed flow $G_7$ is given. The remaining steam flow is retained in the volume and this is a simulated parameter; so the first differential equation is:

$$G_8 = \frac{dm_8}{dT} = G_5 - G_7 = x_4 G_4 - \frac{a_j/F-F_i/Dt_{ln_j}}{h''-h'} \qquad /2.3.1.-9./$$

In the dynamic model, the heat storage capacity and the average temperature of the heated water are defined by the difference of the heat flow from the condensing steam to the tube $/W_{31}/$ and that from the tube to the condensate $/W_{32}/$:

$$dW = W_{31} - W_{32} = a_j/F-F_i/Dt_{ln_j} - G_{2o}c_{2o}/t_{22}-t_{21}/ \qquad /2.3.1.-1o./$$

Here water temperatures $t_{21}$ and $t_{22}$ are functions of time, but $t21$ was already calculated from Eq./2.3.1.-4./ and its derivate can be estimated as:

$$\frac{dt_{21}}{dT} = \frac{t_{21} - t_{21p}}{T-T_p} \qquad /2.3.1.11./$$

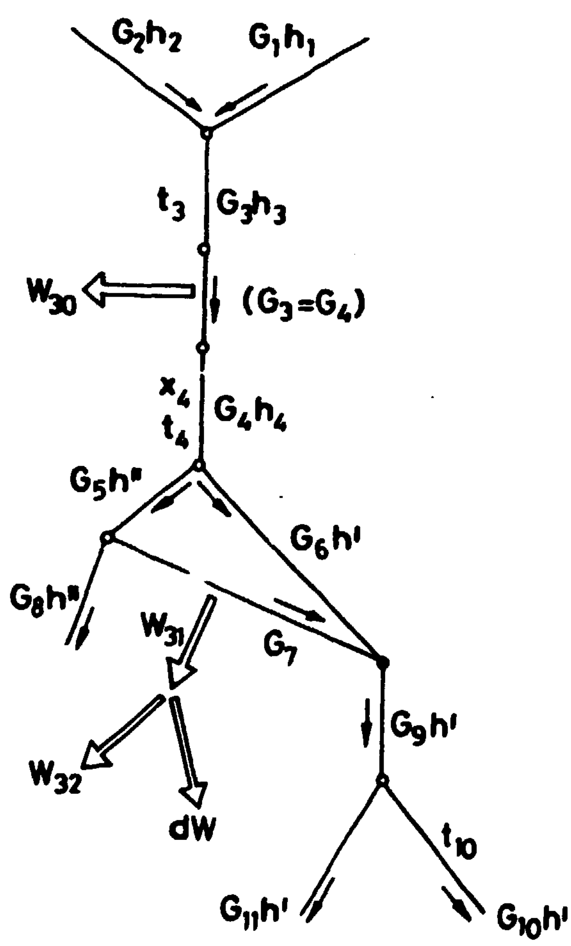



Fig. 9 Mass and Heat Flows in LP Heater Model

The same storage capacity and thus the average temperature of the main condensate is also defined by the heat capacity of the wall and the heated water. Substituting half of this latter sum of capacities by a thermodynamic inertia $I$ [24] , we get:

$$\frac{dt_a}{dT} = \frac{1}{2}\left(\frac{dt_{21}}{dT} + \frac{dt_{22}}{dT}\right) = \frac{W_{31} - W_{32}}{m_w c_w + m_c c_c} = \frac{dW}{2I}$$

/2.3.1.-12./

Experience has shown that it is practical to take $I$ to be in direct proportion to the main condensate flow and specific heat:

$$I = w_c\, G_2\, c_2$$

/2.3.1.-13./

where $w_c$ is a tunable parameter and hence the second differential equation for the digital simulation of the LP heater is:

$$\frac{dt_{22}}{dT} = \frac{dW}{w_c G_2 c_2} - \frac{dt_{21}}{dT}$$

/2.3.1.-14./

As the level control in the hotwell is assumed as ideal,there is no drain stored in it: $G_{11}$ is zero and $G_{1o}=G_9$ . But, for numerical reasons, a digital filter needs to be used to get an acceptable value of the drain leaving the heater /$G_{1o}$/; this is the third differential equation:

$$\frac{dG_{1o}}{dT} = \frac{G_9 - G_{1op}}{w_d}$$

/2.3.1.-15./

where $w_d$ is a tunable time-constant /$s$/.

The description of the common pressure of the volumes is based on:

$$\frac{m_g}{r''} + \frac{m_f}{r'} = V_i + V_j = V = const.$$

/2.3.1.-16./

where $m_g$ and $m_f$ are the masses of steam and water respectively, stored in the whole LP heater volume. Deriving $V$ by time, as there is only one pressure for the common volume:

$$\frac{dp}{dT} = \frac{dp_g}{dT} = \frac{dp_f}{dT} \qquad /2.3.1.-17./$$

and using

$$\frac{dm_f}{dT} \approx 0$$
$$\frac{dm_g}{dT} = G_8 \qquad /2.3.1.-18./$$
$$m_g = m_8$$

we get

$$\frac{dp}{dT} = \frac{G_8}{r'' \left[ m_8 \frac{d(1/r'')}{dp} + m_f \frac{d(1/r')}{dp} \right]} \qquad /2.3.1.-19./$$

where $m_f$ is a constant, $m_8$ is given by digital simulation, and the derivates of the specific densities are property parameters /calculated at saturation/. Here the last member of the denominator is negligible as the saturated water is nearly incompressible.

## 2.3.2. HP Feedwater Preheaters

The feedwater preheating consists of three heat exchangers with a bypass line /cf. element marked "$v$" in Fig.1/. The point-model used for these is similar to that of the LP heaters. A difference is that when the bypass line is opened, the values of feedwater flow and temperature are not affected by this model. When the bypass is closed, the HP heater has to be calculated, but there is no need to cool the bleed steam, as it is always saturated, The result is that - in terms of Point 2.3.1.- we get:

$$\left.\begin{aligned} h_3 &< h'' \\ t_4 &= t_3 \\ W_{3o} &= 0 \\ F_i &= 0 \\ t_{21} &= t_{2o} \end{aligned}\right\} \qquad /2.3.2.-1./$$

The calculation described in Point 2.3.1. is followed, using the appropriate values:

- the heating steam /$G_1$/ is a part of the HP bleed flow /defined by the deaerator, see Point 2.4.3./,
- the drainage mixed with this prior to their flow into the heater is the condensate of the heating steam of the working steam reheater /RH/,
- the condensate of this heating mixture is drained off into the feedwater tank.

## 2.4. Modelling the Other Components

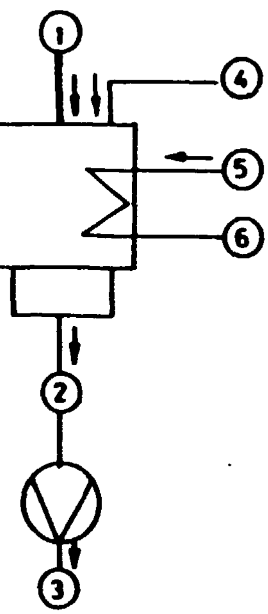
### 2.4.1. Main Steam Condenser and Condensate Pump

The main condenser and pump /see Fig.lo, cf. elements marked "*n*" and "*p*" in Fig.1/ do not need dynamic modelling.

The condenser has a relatively infinite volume which has an extremely weak coupling to other elements, so a simple approach is acceptable:

$$p = p' \; / \; t_6 \; / \qquad /2.4.1.-1./$$

and mass- and heat-flows are taken as independent values, no air-content is calculated and the main condensate pump is reduced to a pressure-difference only /$p_3 > p_2$ /.



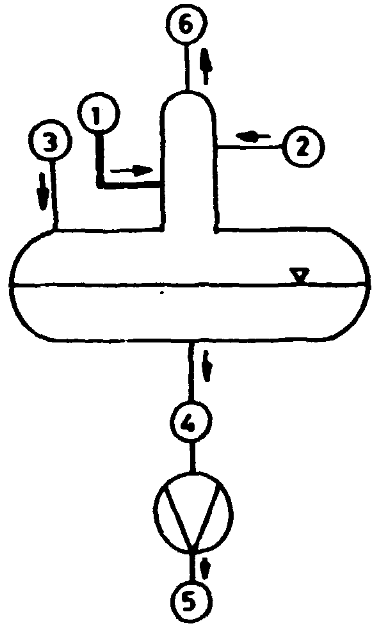

indices: 1 - exhaust steam from LP turbine
2 - main condensate to pump
3 - main condensate to LP heaters
4 - drainage from LP heaters
5 - cooling water inlet
6 - cooling water outlet

Fig.1o Main Steam Condenser and Condensate Pump

### 2.4.2. Deaerator, Feedwater Tank and Feed Pump

The point-model of the deaerator is a mixing condenser, and it is combined with a tank model /see Fig.11, cf. elements marked "*s*" and "*t*" in Fig.1/.

It is assumed that the level control of the tank is ideal, the pressure is constant /its control is perfect/, and there are no fumes /$G_6 = 0$ /.




indices: 1 - heating steam from HP extraction
2 - preheated main condensate
3 - drainage from HP heaters
4 - feedwater to feed pump
5 - feedwater to HP heaters
6 - fumes

Fig. 11 Deaerator, Feed Tank and Feed Pump

Heat- and mass-balance equations describe this model:

$$G_1 = \frac{G_4 /h_4 - h_2/ - G_3/h_3 - h_2/}{h_1 - h_2} \qquad /2.4.2.-1./$$

and as $G_4$ is determined by the steam consumption /assuming ideal level control of the steam generator/, the condensate flow is expressed as:

$$G_2 = G_4 - G_1 - G_3 \qquad / 2.4.2.-2./$$

Due to the check valve and the HP heater, the calculation of heating steam flow $G_1$ is also affected by a consideration of Point 2.4.3. The pump is reduced to a pressure-difference only $/p_5 > p_4/$.

2.4.3. Check Valves

During transients, when the load rapidly decreases, the pressure of the heaters may drop well below the boiling point defined by the temperature of the heated water. In this case evaporation starts in the condensates of the bleed steam and this would cause a return flow in the extractions which is not allowed for turbine protection reasons; therefore a check valve is built into these extraction pipes to prevent this effect. These valves are also modelled in order to follow the real process.

The theory is that the pressure difference $/Dp/$ between the turbine side $/p_1/$ and the heater side $/p_2/$ defines the maximum flow allowed by the valve:

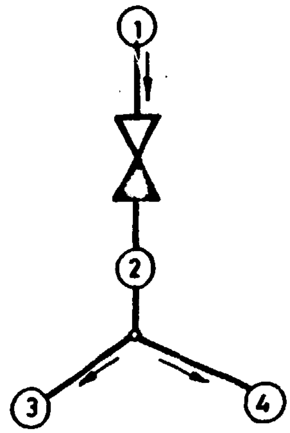
$$G_{2_{max}} = \sqrt{z\ r\ Dp} = \sqrt{z\ r\ /p_1 - p_2/} \qquad /2.4.3.-1./$$

where $z$ is a constant defined by the steady state.

But whenever $Dp$ is negative, a tunable time constant $/w$, in s $/$ is set and, until the time limit $/T+w/$ has expired, the flow $G_2$ remains zero. This is an approximation of the actual inertia of these valves and serves as a filter as well.

This approach is applied to the LP check valve, where the LP bleed steam $/G_1/$ is not divided into two flows but it is the heating steam of the LP heaters $/G_2/$. As value $p_2$ is the pressure of the LP heater, $G_2$ is equal to $G_{2max}$ whenever the valve is opened. The actual value $/G_{2max}$ or zero/ is used as value

"$G_4$" in Eq. /2.1.2.-2./ when it is used in Point 2.1.3.




<u>Indices:</u> 1 - extraction /HP turbine side/
2 - extraction /heater side/
3 - bleed steam for heating the deaerator
4 - bleed steam led to the HP heater
*max* - maximum

<u>Fig.12</u> HP Check Valve

As for the HP check valve /cf. Fig.12/, the bleed steam is used first for deaeration /$G_4$/ and the rest is for heating the HP heaters /$G_3$/.

The necessary heating steam of the deaerator /$G_4$/ is given by value "$G_1$" in Eq. /2.4.2.-1./, and the maximum possible flow through the valve /$G_{2max}$/ is defined by Eq. /2.4.3.-1./. Hence the heating steam flow of HP heaters is limited:

$$G_3 = G_{2max} - G_4 \qquad /2.4.3.-2./$$

This flow /$G_3$/ is used as value "$G_1$" in Eq. /2.3.1.-1./ when this calculation is done in Point 2.3.2. But, when $G_4 \geq G_{2max}$

then $G_3=0$ and $G_4=G_{2max}$, the last of these latter values is substituted for value "$G_1$" in Eq. /2.4.2.-2./, and Eq. /2.4.2.-1./ is omitted. This is acceptable as the water stored in the feed tank can be taken as a relatively infinite volume, so its coupling to other elements is weak.

These flows have to be summed up to get the bleed flow needed:

$$G_2 = G_3 + G_4 \qquad /2.4.3.-3./$$

This result is either less than or equal to $G_{2max}$ or - when the valve is closed - is equal to zero. Value $G_2$ is used as value "$G_4$" in Eq. /2.1.2.-2./ and it ensures that the turbine can be appropriately calculated.

## 3. FITTING TWO-VARIABLE SURFACE FUNCTIONS

In the secondary circuit model the thermodynamic properties of water and steam are approximated by a two-dimensional spline technique [32]. The method briefly described below leads to three times constinuously differentiable surfaces. They are, thus, sufficiently "smooth" for simulation purposes.

### Description of the spline technique

The surface $z = f/x,y/$ will be determined by a set of points: each lying on the surface. All the three coordinates of these points have to be given. This is done in the following way:

Let us first consider the perpendicular projection of the surface $z = f/x,y/$ to the plane $/x,y/$. Let $x_n$ be a strictly monotonously increasing series on this projection, $x_n$ need not be equidistant. Another strictly monotonously increasing series $y_{im}$ is chosen for each $x_i$ , $y_{im}$ need not be equidistant, but the number of elements in the series $y_{im}$ must be the same for all values of $i$. This procedure will determine a trapezoidal net, as shown in Fig.13.

The input variables of the method will be the points of this net /the basic points/, which are given by their coordinates together with the number of surface elements $n-1$ and $m-1$ in the $x$ and $y$ direction respectively. The coordinates of the basic points are stored in a three dimensional array $P\,(3,n+2,\ m+2)$

$$P\,(I1,\ I2,\ I3)$$

where $I1 = 1,2,3,$ refers to $x,y$ or $z$ respectively

$I2 = 2,\ldots,NN+2;$ $NN$ is the number of surface elements along $x$

$I3 = 2,\ldots,MM+2;$ $NM$ is the number of surface elements along $y$

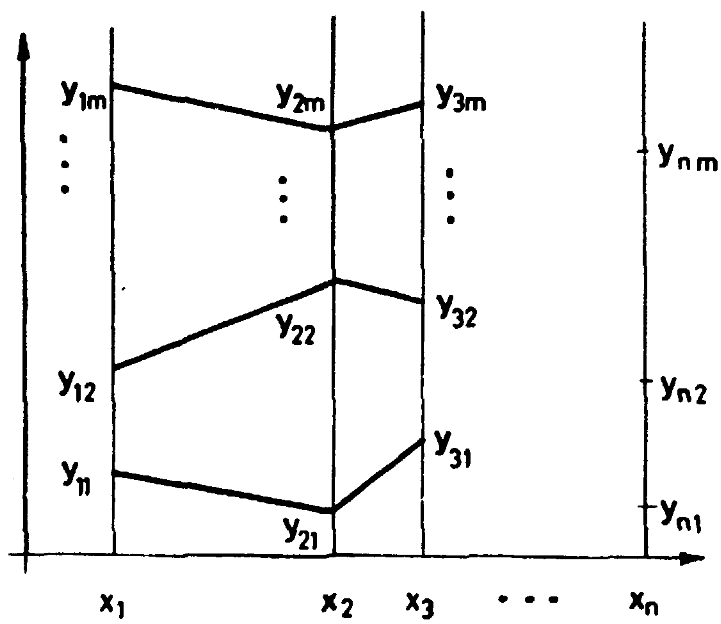



Fig.13

The set of supporting points needed for the spline interpolation will be obtained from the set of basic points by linear transformation expressed by the following equations:

$$\begin{bmatrix} 6 & 1 & & & & \\ 1 & 4 & 1 & & 0 & \\ & 1 & 4 & 1 & & \\ & & & \ddots & & \\ 0 & & & 1 & 4 & 1 \\ & & & & 1 & 6 \end{bmatrix} \cdot \begin{bmatrix} PK\,(I1,I2,2) \\ \vdots \\ PK\,(I1,I2,MM+2) \end{bmatrix} = 6 \begin{bmatrix} P\,(I1,I2,2) \\ \vdots \\ P\,(I1,I2,MM+2) \end{bmatrix}$$

supporting point coordinates /to be determined/ — basic point coordinates /given/

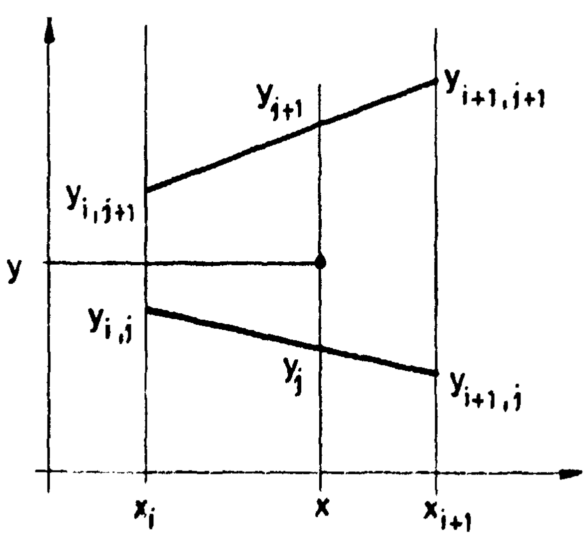
This set of equations is solved for $\forall I2$ ($I2 \in [2, NN+2]$, integer) and $\forall I1$ ($I1 \in [1,3]$, integer).
Elements of the PK array not given by the solution will be calculated as follows:

$$PK(I1,I2,MM+3) = 2PK(I1,I2,MM+2) - PK(I1,I2,MM+1)$$
$$PK(I1,I2,1) = 2PK(I1,I2,MM+2) - PK(I1,I2,3)$$
for $\forall I2(I2 \in [2,NN+2]$, integer)and $\forall I1(I1 \in [1,3]$, integer);

$$PK(I1,1,I3) = 2PK(I1,2,I3) - PK(I1,3,I3)$$
$$PK(I1,NN+3,I3) = 2\ PK(I1,NN+2,I3) - PK(I1,NN+1,I3)$$
for $\forall I3(I3 \in [2,MM+2]$, integer)and $\forall I1(I1 \in [1,3]$, integer);

$$PK(I1,1,1) = 2PK(I1,2,1) - PK(I1,3,1)$$
$$PK(I1,NN+3,1) = 2PK(I1,NN+2,1) - PK(I1,NN+1,1)$$
$$PK(I1,1,MM+3) = 2PK(I1,1,MM+2) - PK(I1,1,MM+1)$$
$$PK(I1,NN+3,MM+3) = 2PK(I1,NN+3,MM+2) - PK(I1,NN+3,MM+1)$$
for $\forall I1(I1 \in [1,3]$, integer ).

To obtain the desired function at any arbitrarily chosen $x,y$ coordinate-pair, the point $x,y$ is first located on the net by successive bisecting of distances, then the $x$ and $y$ coordinates are normalized between $0$ and $1$, as shown in Fig.14.




$$u = \frac{x - x_i}{x_{i+1} - x_i}$$

$$v = \frac{y - y_j}{y_{j+1} - y_j}$$

Fig. 14

Four third order weight functions are defined by the relative coordinates $u$ and $v$:

$$S_0(u) = \frac{1}{6}(1-u)^3$$

$$S_1(u) = \frac{1}{6}(3u^3-6u^2+4)$$

$$S_2(u) = \frac{1}{6}(3(1-u)^3 - 6(1-u)^2 + 4)$$

$$S_3(u) = \frac{1}{6}u^3$$

/$S_0(v)$ , etc. are obtained in a similar way/.

The value of the function at the desired point is given by the following expression:

$$F(u,v) = \sum_{j=0}^{3} \sum_{i=0}^{3} S_j(v)S_i(u)PK(3,NT-1+i, MT-1+j)$$

here the surface element containing the desired point is identified by serial numbers $NT$ and $MT$ in the $x$ and $y$ directions, respectively.

## 4. VERIFICATION OF THE MODELS

The coupling of all submodels of the secondary circuit made possible to form a dynamic simulation model of the thermodynamic behaviour of the secondary circuit of a nuclear power plant.

The tunable parameters are fitted to achieve good agreement with the available measurements. Here we present three examples. The units are as follows: time in s, flows in kg/s, pressures in bar, temperatures in $^{o}C$, power in MW.

The test shown in Fig.15 is to demonstrate the stability of the model. First the pressure of the main steam header is kept for more than a minute and the values of the calculated parameters show a steady state; then the steam pressure is decreased and increased, the responses proved correct: the turbine power follows the disturbance and returns to steady state but the feedwater flow has some delay due to the tanks and the piping /it returns to the initial steady state after lo minutes/.

It is important that the HP feedwater preheaters can be bypassed /see Fig. 16/: the feedwater flow is maintained but there is no water flow in the HP heater. The result is, that the feedwater temperature drops to that of the deaerator. The pressure of the heater increases /as there is no cooling/ and this leads to a reduced steam flow: there is no need for heating steam and only the condensate of the reheater /RH/ flows through the HP heater. The steady state is disturbed by a step-change but a new steady state is formed and there was no oscillation proving the stability of the model.

Finally a characteristic transient is shown in Fig 17: the control valve of the turbine is closed due to a protection /the breaker of the electrical generator is opened - the unit is isolated from the electrical power grid/, this leads to a rapid decrease of the turbine power, the main condensate and the feedwater start to cool down. When the power is reduced to a certain level /75% load/, the bypass valve is opened and the feedwater temperature drops to that value of its tank. After 4 seconds of waiting for the decrease of power, the control valve of the turbine is opened again to supply the self-consumption of the isolated unit. When the turbine power is enough for this, it is maintained, but the main condensate temperature is decreasing for a time, then it has an inflexion point due to the small increase of the inlet steam flow, and later it becomes steady. The main condensate and feedwater flows can follow these disturbances only with some delay because of the holdup time in the condenser, deaerator and piping /but a new steady state comes after 3 minutes/.

These and other results show acceptable characteristics of the responses also in the parameters without available site measurements. The maximum deviations were less than the expected, there was no oscillation in any case and the computer time was within the given limit.

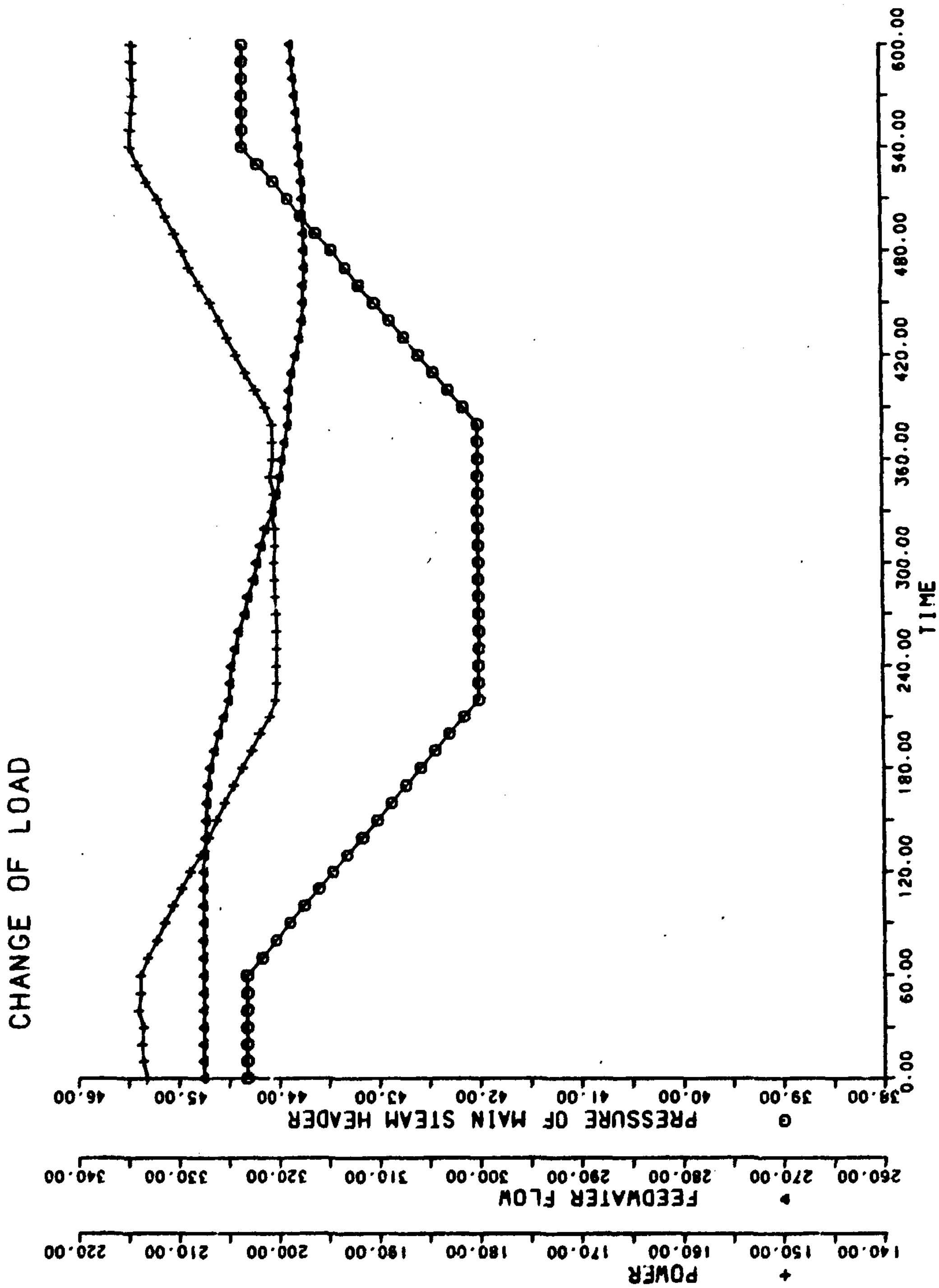

CHANGE OF LOAD
TIME
PRESSURE OF MAIN STEAM HEADER
FEEDWATER FLOW
POWER

Fig. 15

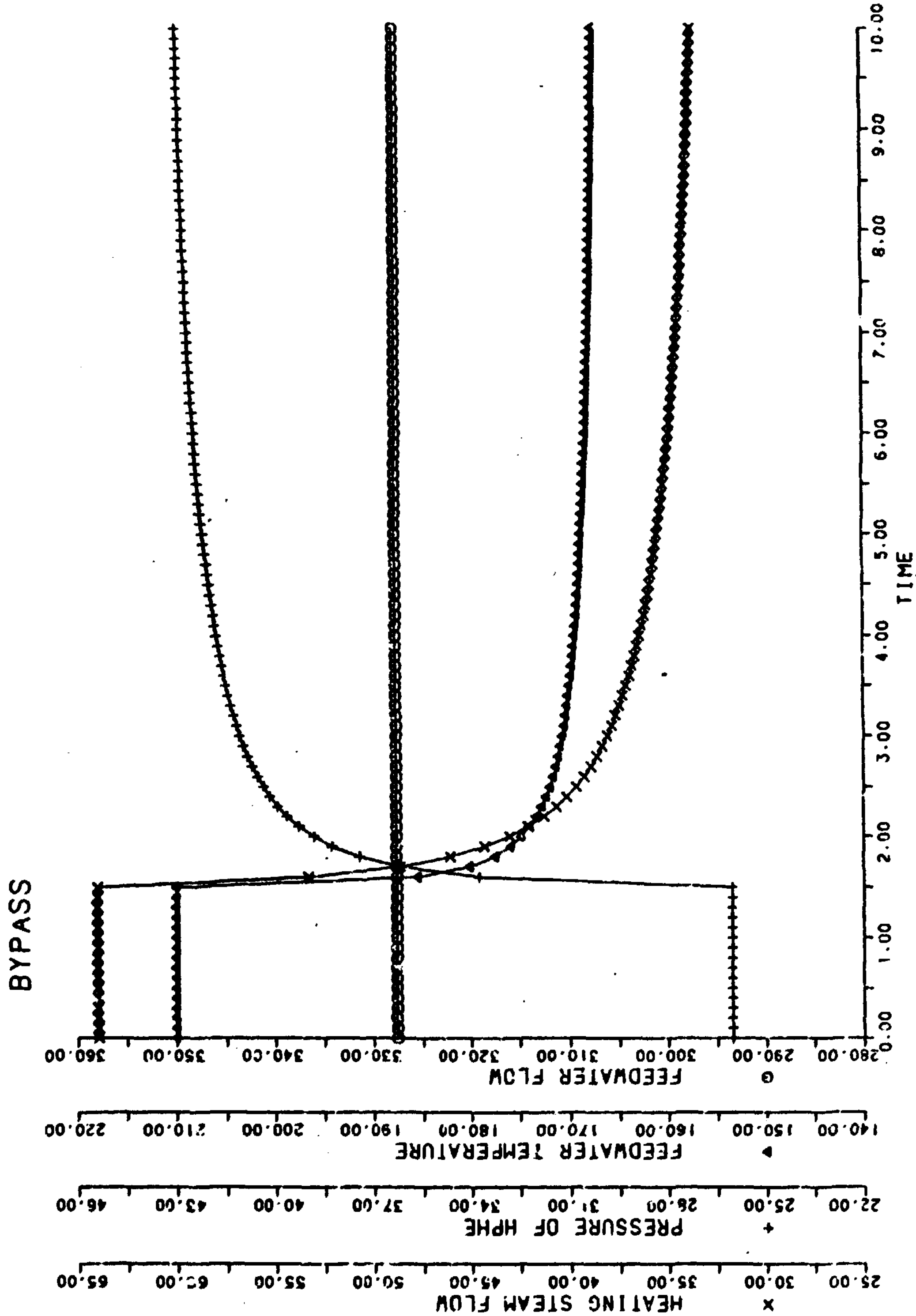

BYPASS
TIME
FEEDWATER FLOW
FEEDWATER TEMPERATURE
PRESSURE OF HPHE
HEATING STEAM FLOW

Fig. 16

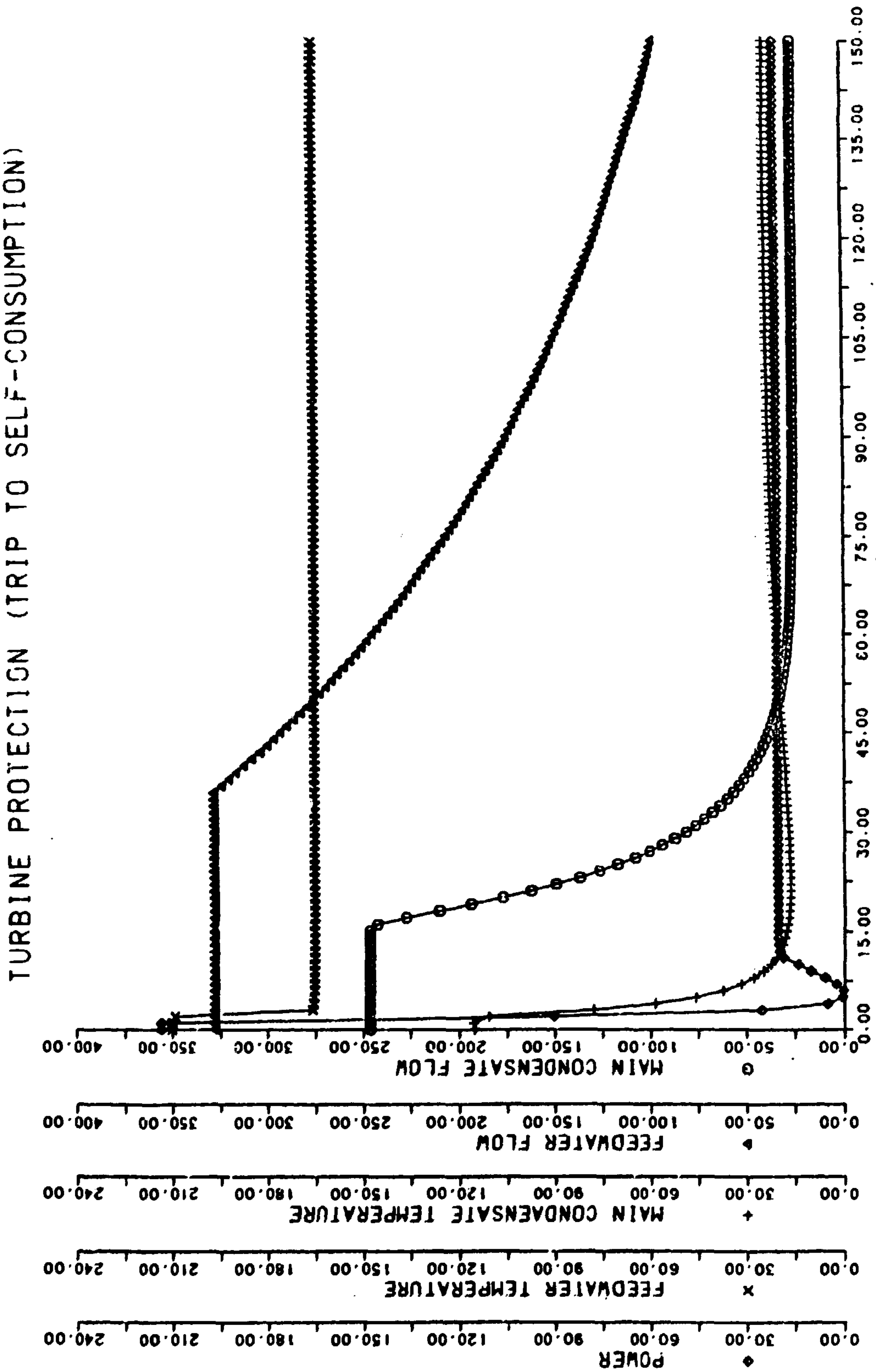

TURBINE PROTECTION (TRIP TO SELF-CONSUMPTION)
TIME
MAIN CONDENSATE FLOW
FEEDWATER FLOW
MAIN CONDENSATE TEMPERATURE
FEEDWATER TEMPERATURE
POWER

Fig. 17

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