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

PART 2

Hungarian Academy of Sciences

CENTRAL RESEARCH INSTITUTE FOR PHYSICS

BUDAPEST

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J. Doorenbos, A. Gács, Zs. Kiss: Simulation of the dynamic behaviour of the secondary circuit of a WWER-440 type Nuclear Power Plant. Part 2. KFKI-1987-81/G

ABSTRACT

This report describes the dynamic simulation models of the most important controllers of the secondary circuit of a WWER-440 type Nuclear Power Plant i.e. the hydraulic turbine controller and the level controls of the condenser hotwell and that of the feedwater tank. Simulation results are also presented. This paper continues the descriptions of the dynamic simulation models of the primary circuit (Reports KFKI-1983-127 and KFKI-1985-08), and the secondary circuit (Report KFKI-1987-44/G) of this type nuclear power plants.

Я. Дооренбос, А. Гач, Ж. Киш: Симуляция динамики второго контура АЭС типа ВВЭР-440. Часть 2. KFKI-1987-81/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 коитура. В отчете даются некоторые симуляционные результаты.

Doorenbos J., Gács A., Kiss Zs.: WWER-440 tipusu atomerőmű szekunderköre dinamikus tulajdonságainak szimulációja. II. rész. KFKI-1987-81/G

KIVONAT

Ez a riport egy VVER-440 tipusu atomerőmű szekunderköre legfontosabb szabályozóinak dinamikus szimulációs modelljét irja le, igy a hidraulikus turbinaszabályozást valamint a kondenzátorzsomp és a tápviztartály vizszintszabályozóit. Szimulációs eredményeket is bemutatunk. Ez a dolgozat folytatása az ilyen tipusu atomerőművek dinamikus szimulációs modelljeit leiró riportjainak (primerkör: KFKI-1983-127 és KFKI-1985-08; szekunderkör: KFKI-1987-44/G).

1. INTRODUCTION

A basic principles simulator is under construction in our Institute for WWER-440 type Nuclear Power Plants. The purpose of this simulator is to provide a means of studying the normal and close-to-normal operating states of WWER-440 power plants and to help in teaching the essential control concepts of these units.

The first part of this report /KFKI-1987-44/G/ has already been published and it describes the models of the main technological units of the secondary circuit. The scope of simulation is now extended to the most important controllers of the secondary circuit; the models of these controllers are described in the present paper. The controllers to be discussed are the following:

- the hydraulic turbine controller,
- the level control of the condenser hotwell,
- the level control of the feedwater tank.

2. THE HYDRAULIC TURBINE CONTROLLER

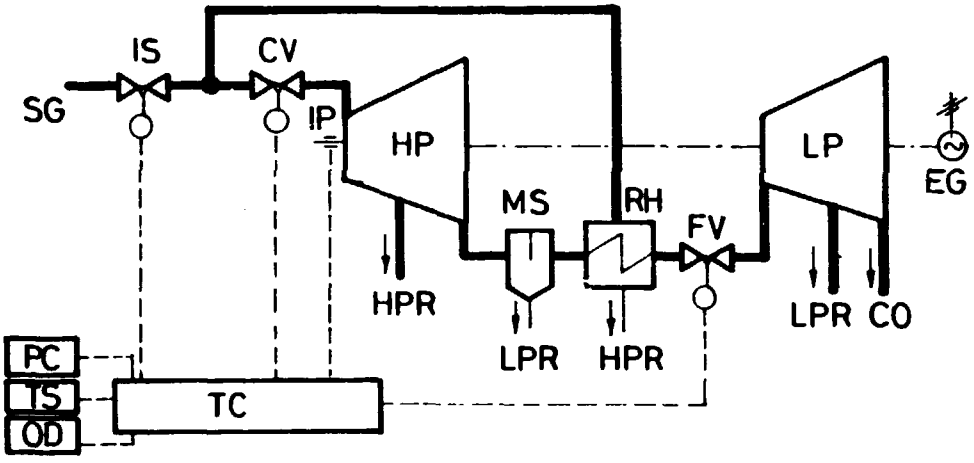
The steam turbine model is discussed in Part 1 of this report. The high pressure and the low pressure turbines are connected through a moisture separator and reheater unit. The inlet flows of both turbines are to be controlled.

The principle of nozzle control is well-known: various arrangements of valves are employed as automatic devices. Due to the large output and the high-pressure steam, a hydraulic controller unit is also needed for the steam valves of the K-220-44 type turbines used in WWER-440 type power plants.

As a means of simplifying our approach, we can divide the steam valves into groups, viz.

- isolation valves /to prevent admission of steam to both the turbine and the heating side of the reheater unit in the case of rapid transients/,
- main steam control valves /to set the inlet steam pressure of the high pressure turbine according to the load to be followed/,
- flap valves /to provide protection for the low pressure turbine during transients/.

Each of these groups can be simulated by a single valve model, i.e. one isolation valve /IS/, one control valve /CV/, and one flap valve /FV/, as shown in Fig.1.



TC hydraulic turbine controller
PC inputs from power control unit /setpoint/
TS " " techn. signals /protections, etc./
OD " " operator's desk /inhibitions, etc./
SG main steam from steam generators
IS isolation valve
CV control valve
IP impeller /turbine speed/
HP high pressure turbine
HPR to high pressure reheater of feedwater
MS moisture separator
RH reheater of working steam
FV flap valve
LP low pressure turbine
LPR to low pressure reheater of condensate
CO exhaust to main steam condenser
EG electrical generator

Fig.1 Steam valves and control

The modelling of the hidraulic turbine controller is necessary to accurately simulate normal transients that occur during the operation of the turbine. The controller has a great influence during transients such as load change and transition to the no-load state. The controller is also the only connection to the isolation valves, the control valves, and the flap valves. All control functions must be effected through the use of the hydraulic controller unit.

2.1. Controller Operation

A simplified diagram of the controller showing the most important functional blocks is given in Figure 2. The basic principles of operation of each of the blocks are important for simulation purposes, but the exact details are unimportant and unnecessary.

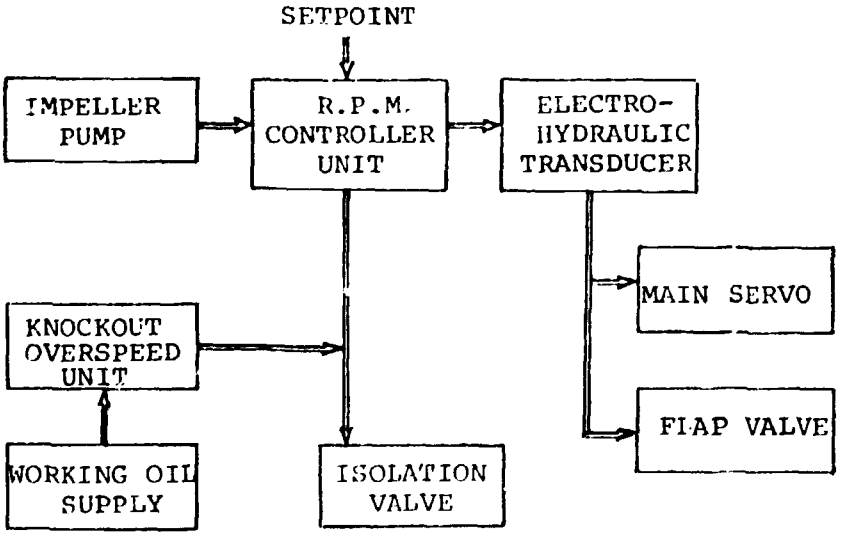


Fig.2 Simplified Controller Diagram

The steam valves are held open by oil pressure against springs that close the valves when the oil pressure is decreased or lost. The valve positions are varied by varying the controlling oil pressures.

The isolation valves are held open by the working oil pressure. The working oil is supplied at a constant pressure of approximately 16.5 bar. This pressure holds the isolation valves in their fully opened state during normal operation.

Both the control valves and the flap valves are controlled by the control oil pressure. The control valves are controlled indirectly by a servomotor. The control oil pressure is determined by the R.P.M. controller unit. This control oil pressure being derived from the working oil pressure by means of a variable gap that allows some of the control oil to flow out. The working oil enters through a fixed gap and exits through a variable gap. This is illustrated in Figure 3. The control oil flows out though the variable gap to atmospheric pressure of approximately 1 bar. The control oil pressure ranges from to approximately 1.2 bar when the variable gap is fully open to the working oil pressure of 16.5 bar when the gap is completely closed.

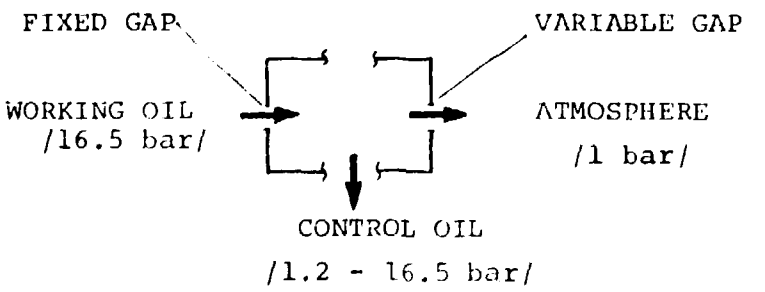


Fig.3 Determination of Control Oil Pressure

The size of the variable gap is determined by the relative positions of the setpoint sleeve and the impeller piston. The setpoint sleeve is connected to the setpoint unit so that an increase in the setpoint causes the gap size to decrease and the control oil pressure to increase;this opens the control valves. A decrease in the setpoint increases the gap size, decreases the control oil pressure, and closes the control valves.

The impeller piston position is determined by the impeller oil pressure. An impeller pump is connected to the axis of the turbine. The pressure of the oil from the impeller pump is proportional to the rotational speed of the turbine. This oil pressure moves the impeller piston inside the setpoint sleeve so that an increase in rotational speed causes an increase in impeller oil pressure which causes the gap to open. The opening of the gap lowers the control oil pressure and closes the control valves until a new equilibrium point is established.

An electro-hydraulic transducer unit has also been installed on the controller. Under normal conditions, it does not affect the operation of the turbine. In the case of a loss of load, however, the electro-hydraulic transducer plays an important role in the control of the turbine. The electro-hydraulic transducer provides a quick response to a trip of the generator breakers. When the breakers are tripped, a current function is generated that causes the electro-hydraulic transducer to allow some of the control oil to flow out. This flow of control oil lowers the control oil pressure and closes the control valves. The control oil flow is proportional to the current. Without this quick response, the turbine would trip because of an overspeed condition before reaching its no-load state because the response time of the rest of the controller is too slow.

The overspeed protection /OP/ and the knockout magnet /KM/ close all three types of valves. The OP and the KM operate in the same way: the working oil is allowed to flow out and the working oil pressure drops thereby causing the isolation valves to close. This drop in working oil pressure also causes the control oil pressure to drop and the control and flap valves to close.

2.2. Modelling the Controller

Although the actual controller is quite complex, it can be simplified and broken down into several block operations for simulation. The following six blocks were chosen for the simulation:

1. Calculation of a pseudo servo position from turbine speed and the setpoint
2. Calculation of a multiplier to simulate the electro-hydraulic transducer.
3. Calculation of the position of the isolation valve.
4. Calculation of a corrected pseudo servo position.
5. Calculation of the control valve position and the effective cross sectional area.
6. Calculation of the flap valve position.

A block diagram of the necessary inputs, calculations, and outputs is given in Figure 4.

First the size of the gap is determined. The impeller piston position is determined by a linear relationship from the turbine rotational speed, bypassing the calculation of the impeller oil pressure. The setpoint sleeve position is determined by a linear relationship from the setpoint input. These two positions are then subtracted to determine the size of the gap.

In the actual controller, the control oil pressure is determined by the gap size, This relationship is quite non-linear.

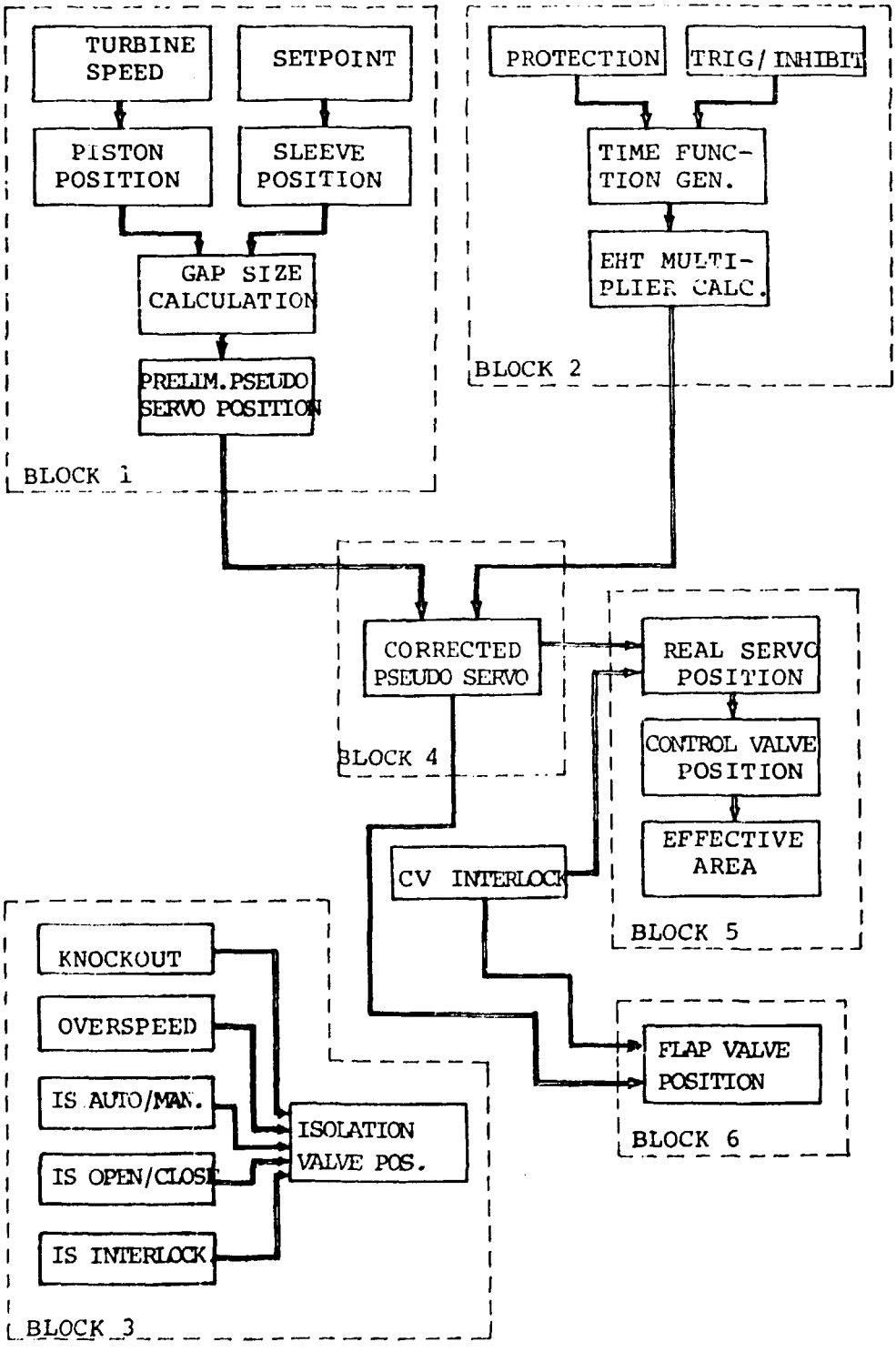


Fig. 4 Block Diagram of Controller Model

The relationship between the control oil pressure and the main servo position is also quite non-linear. However, the relationship between the gap size and the main servo position is nearly linear so that a linear approximation may be used. This also eliminates the unnecessary intermediate calculation of the control oil pressure. Since the control oil pressure is not displayed on the basic principles simulator, its calculation is not required.

There is one minor problem with this approach, though. The main servo position ranges from Ø to 32Ø mm over a control oil pressure range of 4.2 to 1Ø.5 bar. This is less than the actual control oil pressure range of 1.2 to 16.5 bar as is shown in Figure 5. To solve this problem, a pseudo servo position is calculated instead. This pseudo servo position ranges from Ø to 783 mm to cover the entire control oil pressure range. The real main servo position can be determined by subtracting 3Ø8 from the pseudo servo position. This offset of 3Ø8 mm corresponds to a control oil pressure of 4.2 bar, the zero point of the real servo position. After the subtraction, the result is limited so that it is between Ø and 32Ømm.

The next step in the simulation process is to simulate the effect of the electro-hydraulic transducer. As mentioned earlier, the electro-hydraulic transducer only affects the operation of the controller in the case of a protection. This protection initiates a current time function to activate the electro-hydraulic transducer. The state of the protection input is checked. If the protection is activated, and it is not inhibited, the current function is initiated. This current function is simulated by a piece-wise linear function. The current applied to the electro-hydraulic transducer is then converted to a multiplier scaled from Ø.Ø to 1.Ø from another piece-wise linear function. This multiplier will be multiplied by the pseudo servo position to obtain a corrected pseudo servo position and to simulate the decrease in control oil pressure.

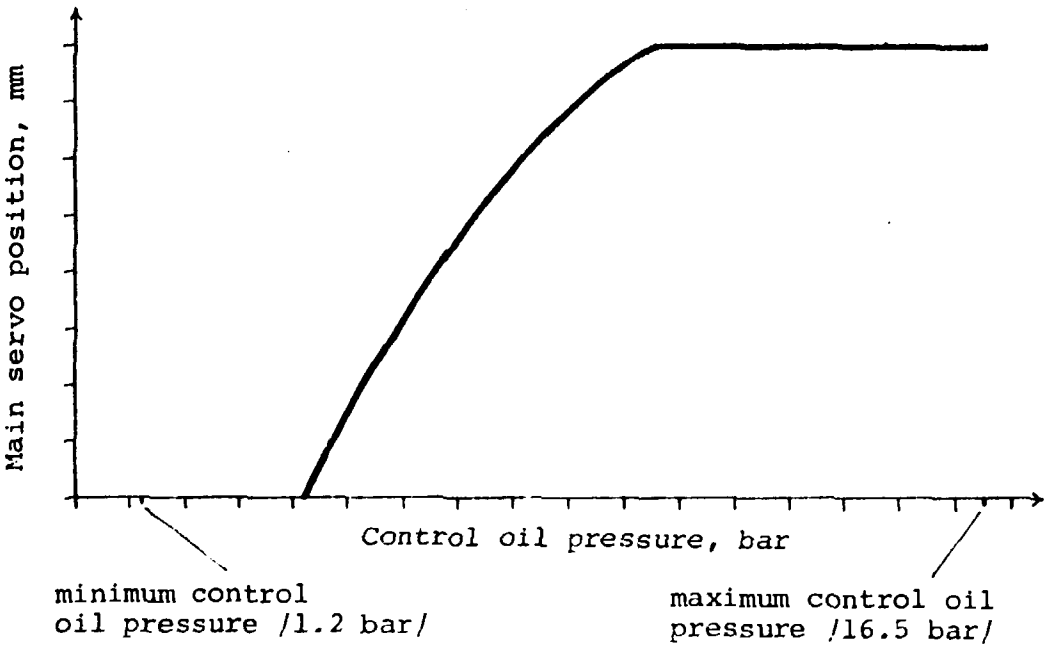


Fig.5 Main Servo Position versus Control Oil Pressure

Next, the position of the isola - valves is determined. For simulation purposes, the isolation valves are replaced by a single valve. The same is true for the control valves and flap valves. The positions are calculated as a fraction of completely open state;thus, their values are between $\phi.\phi$ and $1.\phi$. If a knockout or overspeed condition is active, the isolation valve position is set immediatley to zero since its closing time of $\phi.45$ s is much less than the simulation step size of $1.\phi$ s. If the rotational speed of the turbine is below a certain level, a second press of the knockout switch allows a knockout state to be deactivated.

The simulator also allows for the manual control of the isolation valve. If the isolation valve is under automatic con-

trol, it will be fully open unless an overspeed or knockout condition is active. In that case, it will be completely closed.

In manual mode, it is possible to open and close the valve in steps by issuing open and close commands.

The interlock signal from the plant logics is also simulated. This signal prohibits the opening of the isolation valve until the interlock signal returns to its inactive state.

After calculating the isolation valve position, the corrected pseudo servo position is calculated. This is done based on the preliminary pseudo servo position and the state of the knockout and overspeed protections. If either an overspeed or a knockout is active, the pseudo servo position is set to zero because the working oil pressure is lost, and the control oil pressure is at its minimum value. Otherwise, the preliminary pseudo servo position is multiplied by the multiplier that was determined earlier.

The corrected pseudo servo position is then used to determine the real servo position. The offset is subtracted, the real servo position is filtered to simulate the inertia of the servo and the control valve, and the final position is limited to be between $\emptyset$ and $32\emptyset$ mm. If the control valve interlock signal is active, the real servo position is prevented from increasing. The real servo position is then translated to the control valve position by a piece-wise linear function. The valve position is a value between $\emptyset.\emptyset$ and $1.\emptyset$ the effective cross sectional area of the control valve is also calculated. The relationship between the position of the control valve and its effective area is nearly linear. Therefore, a linear relationship is assumed. This value is also between $\emptyset.\emptyset$ and $1.\emptyset$.

Finally, the flap valve position is calculated. Its position

is determined by a linear relationship from the pseudo servo position. It is also filtered to account for its inertia. Like the real servo position, the position of the flap valve is not allowed to increase if the control valve interlock signal is active. The flap valve position is also a fractional value.

2.3. Testing the Controller Model

The hydraulic turbine controller model was independently subjected to a series of tests to verify that the model adequately simulates the actual controller. In the first of these tests, the setpoint was heldconstant, and the rotational speed of the turbine was increased. The real servo position and the control valve position both close as is expected. A plot of the results of this test is shown in Figure 6.

The second test involved keeping the rotational speed constant while decreasing the setpoint. In this test, the real servo position and the control valve position also decreased as expected. The results of this test are given in Figure 7. Notice that the flap valve also began to close since the opening of the valve overlaps a little with the opening of the control valve.

In the third test, an increase in power output was simulated. First, the speed decreased as the load increased. This was followed by an increase in the setpoint. Both of these effects cause the control valve to open, to increase the speed back to its nominal value and to increase the speed back to its nominal value and to increase the power output. The results of this test can be found in Figure 8.
The last test was a simulation of a turbine trip to self consumption. In this situation, the speed increases rapidly as the turbine loses its load. The electro-hydraulic transducer causes the control valve to close quickly to prevent an over-

speed situation. The control valve interlock signal becomes active and remains active until about 25 s after turbine trip. At that time, the valves are allowed to open, and the controller operates the turbine in its no-load state. The plot of the results of this test is given in Figure 9.

These four tests verify the operation of the turbine controller model during normal transients. The results agree quite closely with actual performance data.

2.4. Integrated Testing

Once the turbine controller model had been tested independently, it was integrated into the turbine model for further testing and verification. The last two tests mentioned above /a change of power output and a trip to self consumption/ were repeated with the integrated model. Once again the performance of the simulator models agreed with actual plant data. The results of these two tests can be found in Figures 1Ø and 11.

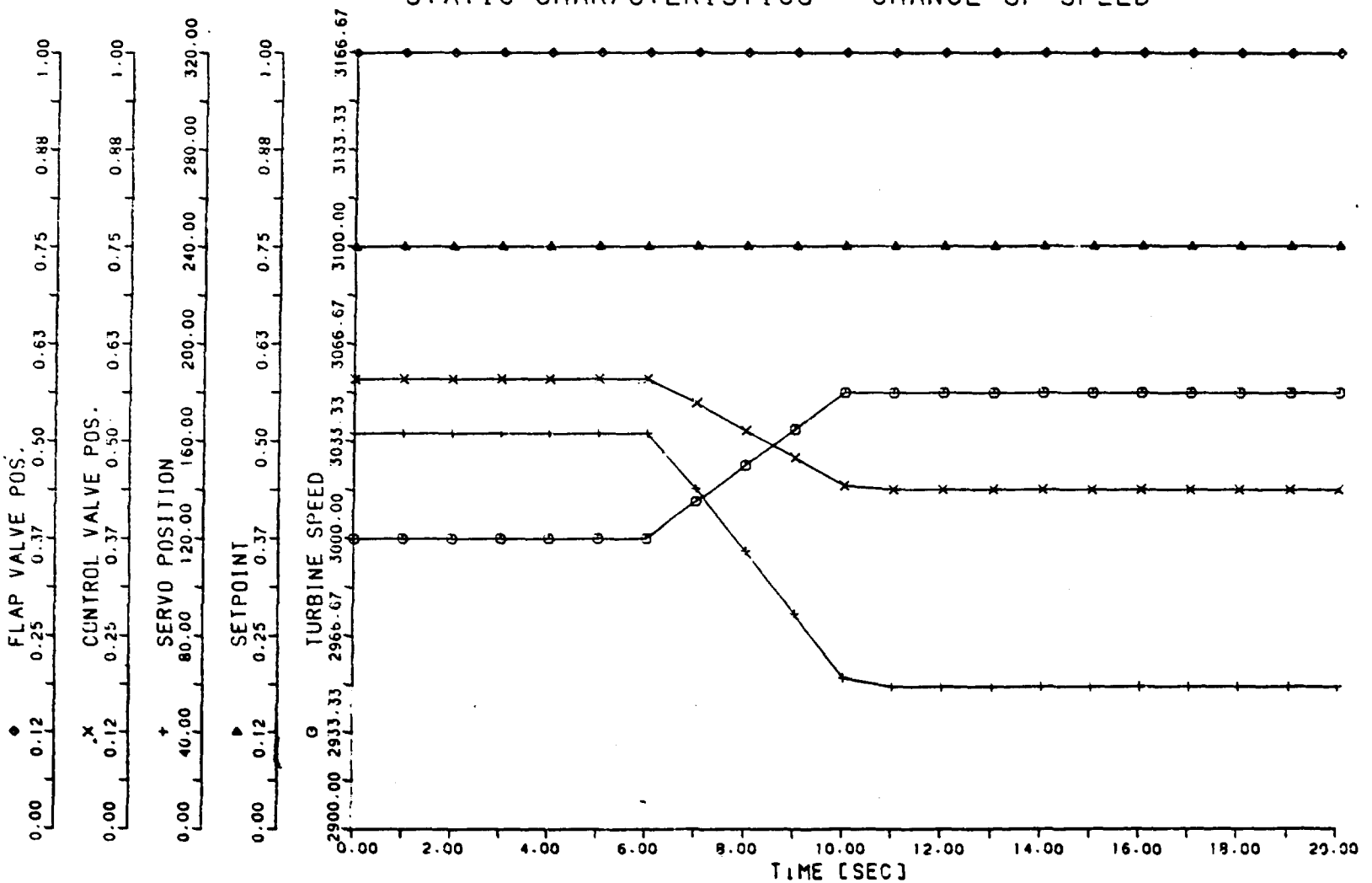

STATIC CHARACTERISTICS - CHANGE OF SPEED
FLAP VALVE POS.
CONTROL VALVE POS.
SERVO POSITION
SETPOINT
TURBINE SPEED
TIME [SEC]

Fig. 6.

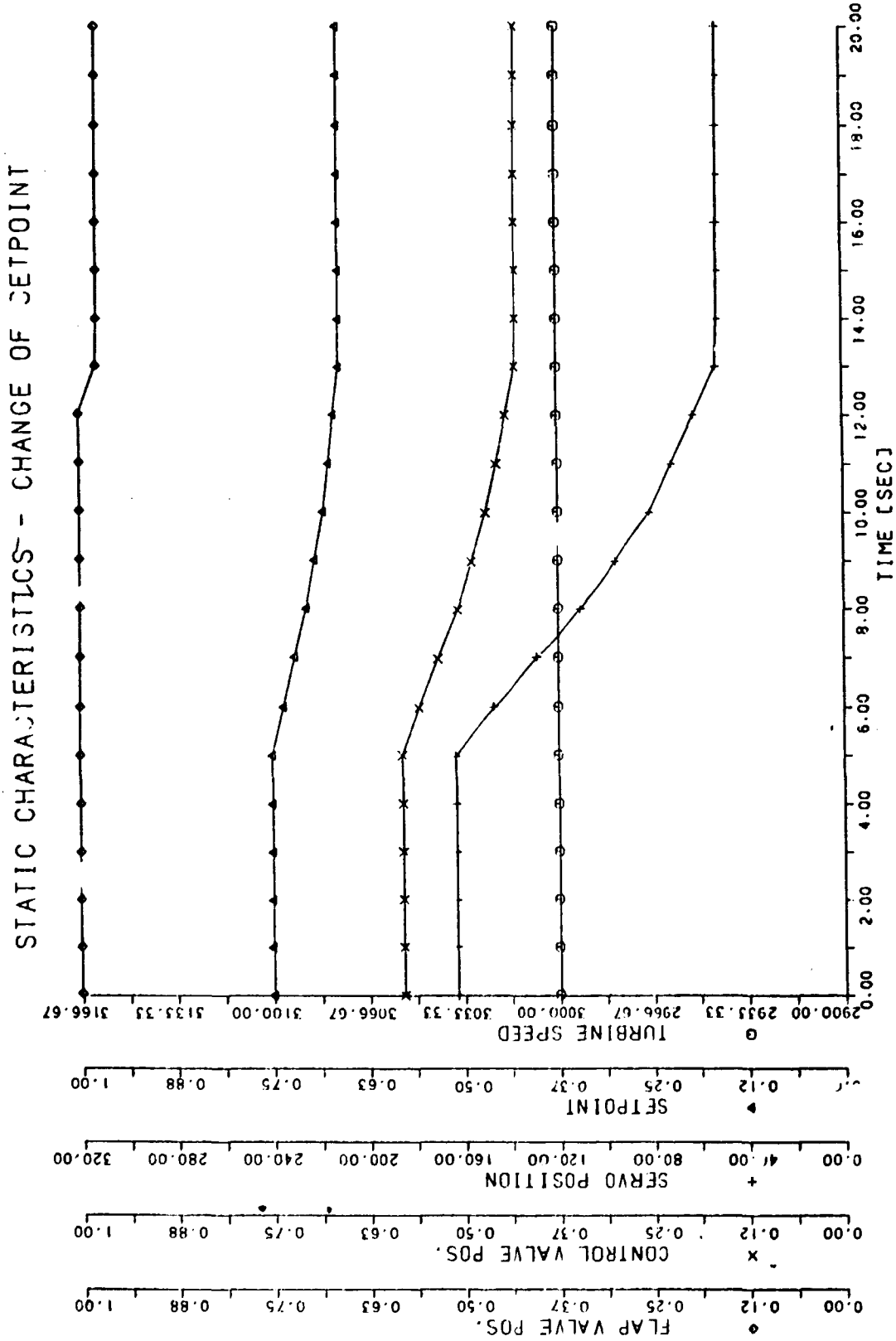

STATIC CHARACTERISTICS - CHANGE OF SETPOINT
TIME [SEC]
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
16.00
18.00
20.00
TURBINE SPEED
2900.00
2933.33
2966.67
3000.00
3033.33
3066.67
3100.00
3133.33
3166.67
SETPOINT
SERVO POSITION
CONTROL VALVE POS.
FLAP VALVE POS.
0.00
0.12
0.25
0.37
0.50
0.63
0.75
0.88
1.00
40.00
80.00
120.00
160.00
200.00
240.00
280.00
320.00

Fig. 7.

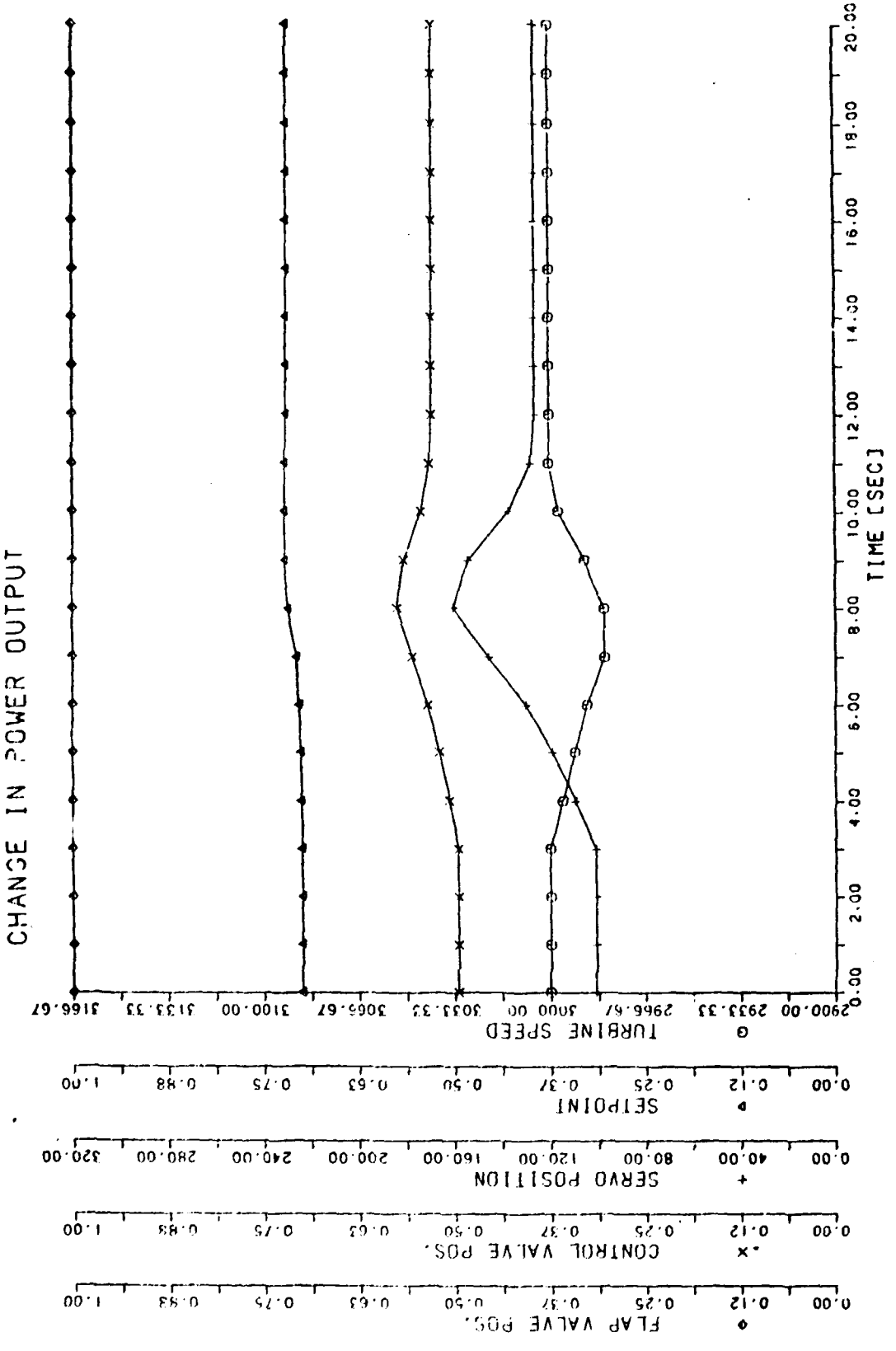

CHANGE IN POWER OUTPUT
TIME [SEC]
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.30
16.00
18.00
20.00
G
TURBINE SPEED
2900.00
2933.33
2966.67
3000.00
3033.33
3066.67
3100.00
3133.33
3166.67
SETPOINT
SERVO POSITION
CONTROL VALVE POS.
FLAP VALVE POS.
0.00
0.12
0.25
0.37
0.50
0.63
0.75
0.88
1.00
40.00
80.00
120.00
160.00
200.00
240.00
280.00
320.00

Fig. 8.

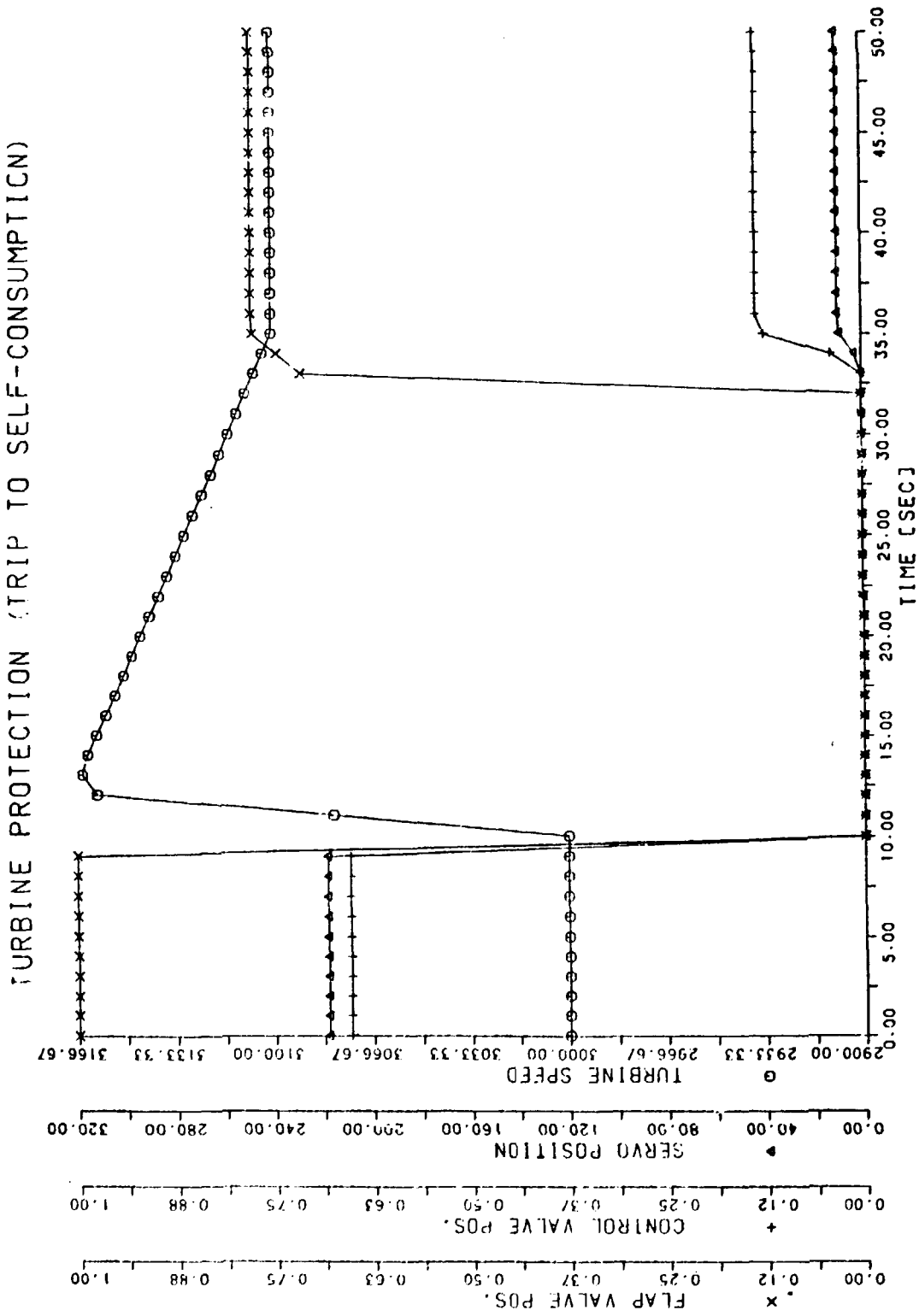

TURBINE PROTECTION (TRIP TO SELF-CONSUMPTION)
TIME [SEC]
0.00
5.00
10.00
15.00
20.00
25.00
30.00
35.00
40.00
45.00
50.00
TURBINE SPEED
2900.00
2933.33
2966.67
3000.00
3033.33
3066.67
3100.00
3133.33
3166.67
SERVO POSITION
0.00
40.00
80.00
120.00
160.00
200.00
240.00
280.00
320.00
CONTROL VALVE POS.
0.00
0.12
0.25
0.37
0.50
0.63
0.75
0.88
1.00
FLAP VALVE POS.
0.00
0.12
0.25
0.37
0.50
0.63
0.75
0.88
1.00

Fig. 9.

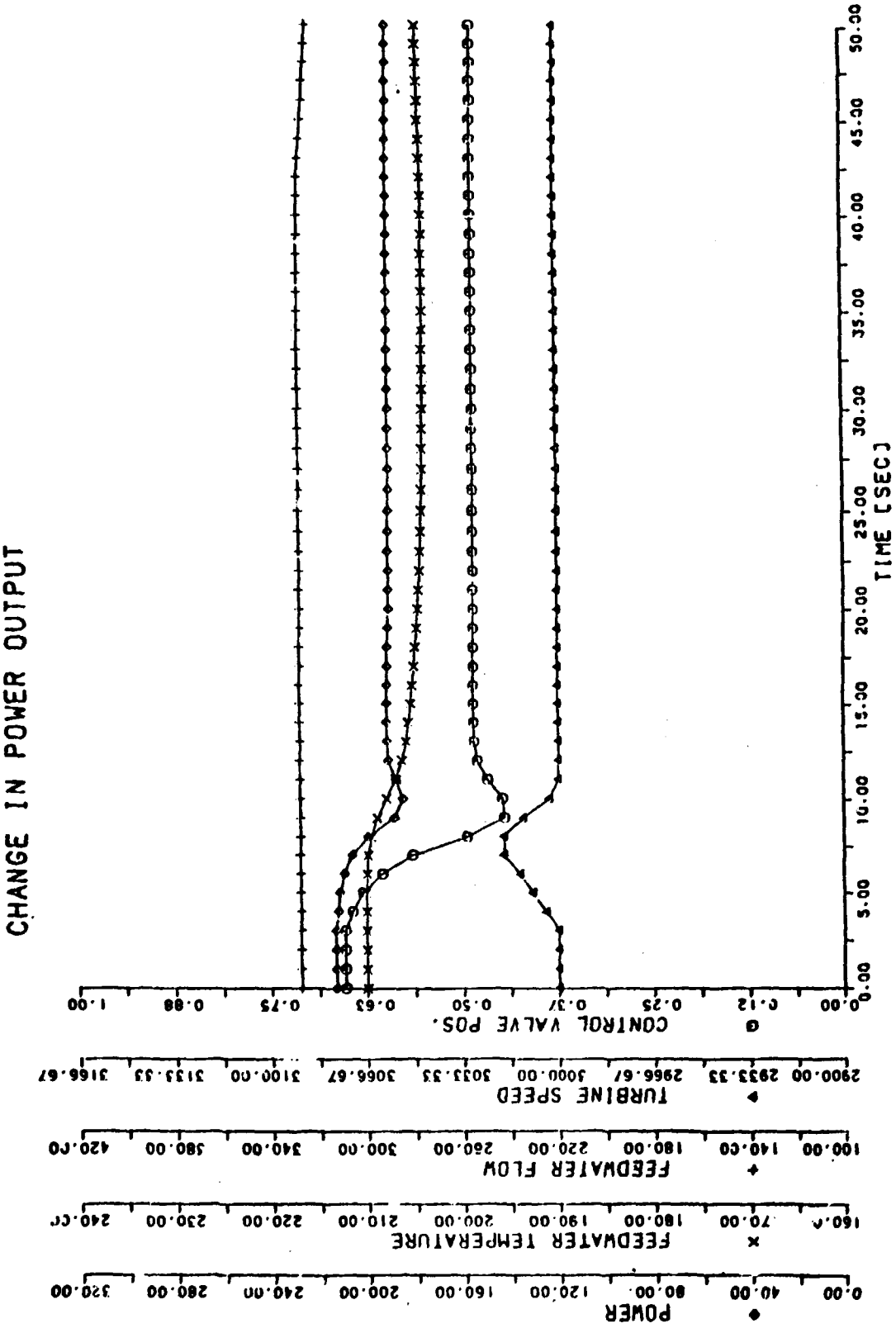

CHANGE IN POWER OUTPUT
TIME [SEC]
0.00
5.00
10.00
15.00
20.00
25.00
30.00
35.00
40.00
45.00
50.00
CONTROL VALVE POS.
TURBINE SPEED
FEEDWATER FLOW
FEEDWATER TEMPERATURE
POWER

Fig. 10.

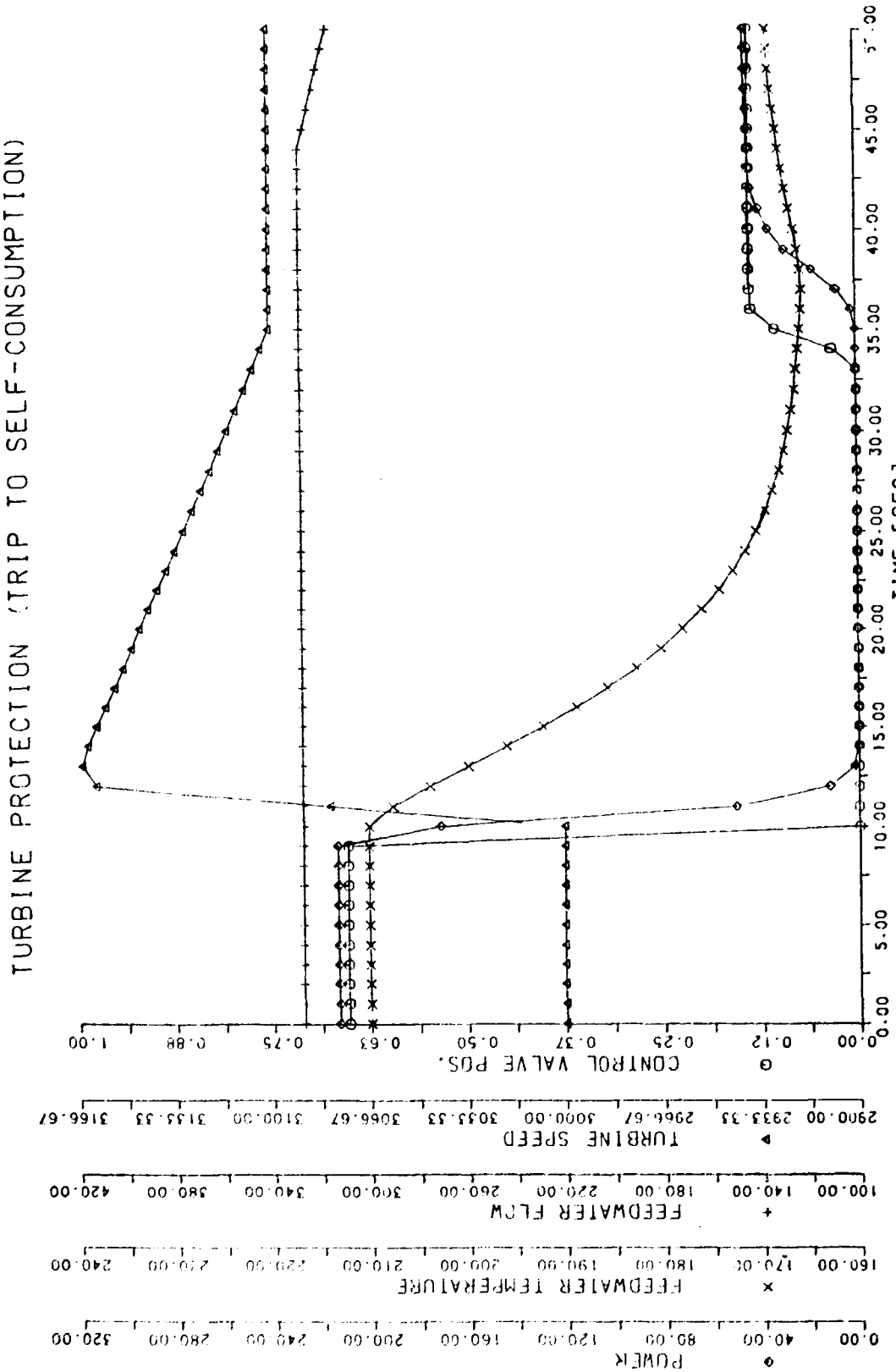
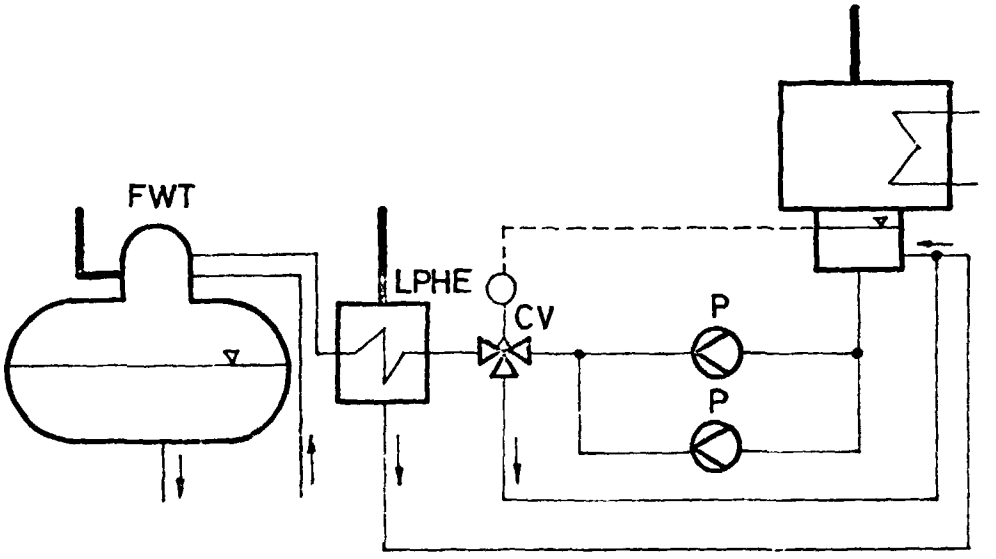

TURBINE PROTECTION (TRIP TO SELF-CONSUMPTION)
TIME [SEC]
0.00
5.00
10.00
15.00
20.00
25.00
30.00
35.00
40.00
45.00
G
CONTROL VALVE POS.
0.00
0.12
0.25
0.37
0.50
0.63
0.75
0.88
1.00
TURBINE SPEED
2900.00
2933.33
2966.67
3000.00
3033.33
3066.67
3100.00
3133.33
3166.67
FEEDWATER FLOW
100.00
140.00
180.00
220.00
260.00
300.00
340.00
380.00
420.00
FEEDWATER TEMPERATURE
160.00
170.00
180.00
190.00
200.00
210.00
220.00
230.00
240.00
POWER
0.00
40.00
80.00
120.00
160.00
200.00
240.00
280.00
320.00

Fig. 11.

3. LEVEL CONTROL OF THE CONDENSER HOTWELL

The model of this level controller is illustrated in Figure 11.



C	- condenser hotwell
P	- condensate pump
CV	- three way control valve
LPHE	- low pressure reheater /simplified representation/
FWT	- feedwater tank

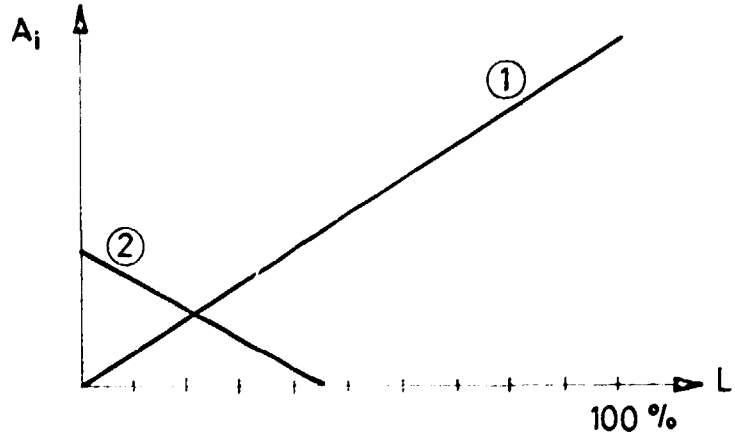
Fig.11

The operation of the controller is as follows: when the level in the condenser hotwell decreases, the control valve moves towards its close position thereby reducing the water flow to the feedwater tank and increasing the recirculation flow back to the condenser hotwell. If on the other hand the level in the hotwell increases, the control valve moves in the open direction which results in a reduction of the recirculation

flow and an increase of water flow into the low pressure reheater and feedwater tank.

For the simulation, some simplifications had to be made:

- the pressure maintained by the pumps should be constant,
- two pumps operate as long as the main steam valve /in font of the turbines/ is open more than 50% of its range, but only one pump operates when the opening of the main steam valve is less than 50%; the starting of the pumps and their trip after they are switched off are not represented in the model,
- the five low pressure reheaters are represented by a simplified model /see Part 1/ and the pressure drop in the reheater depends only on the number of operating pumps,
- the relations of the open cross sections /A_i/ of the three-way control valve are shown in Figure 12 as a function of valve shaft position /L/,



i=1: open cross section towards the reheater
i=2: open cross section towards the condenser hotwell /recirculation/.

Fig.12

- the flow rate across the control valve is calculated from the open cross section of the valve and the pressure drop in the valve:

$$G = A\sqrt{2\varrho\,\Delta p} \qquad /1/$$

where Δp = the pressure drop,
G = the flow rate
A = the open cross section
ϱ = the density of the condensate.

A block diagram of the control algorithm is shown in Figure 13.

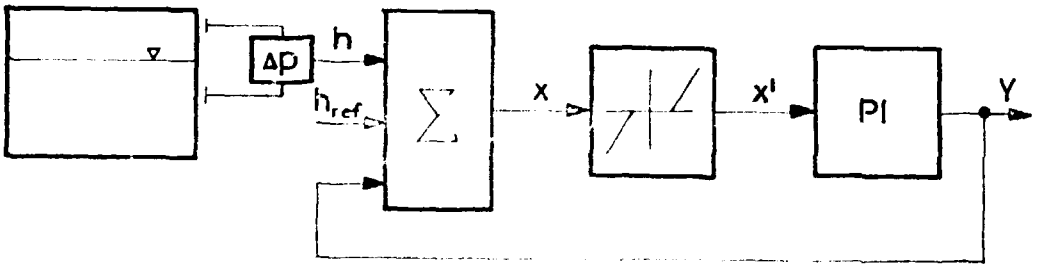


Fig.13

The water level is measured by the pressure-difference of the condenser hotwell. In the simulation model this level is obtained from the mass flow balance in the tank.
Error signal x is taken as

$$x = C_1 h + C_2 h_{ref} + C_3 \Delta Y \qquad /2/$$

where h = the actual water level
h_{ref} = the nominal water level as the reference signal of the controller,
ΔY = the incremental displacement of the control valve
C_1, C_2, C_3 = constants determined by measurements.

The input signal to the controller /X'/ is obtained after taking into account the hysteresis of the system. The controller is an ideal PI /proportional integrating/ controller with the transfer function:

$$Y/S/ = C\left\{1 + \frac{1}{sT_I}\right\} x' \quad /S/ \qquad /3/$$

where C = the proportional coefficient

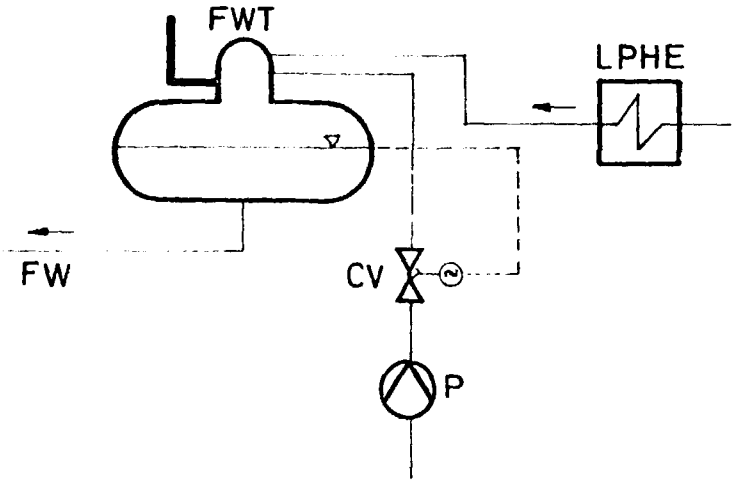
T_I = the time constant of the integration.

The solution of this differencial equation gives the Y position /relative opening compared with nominal opening/ of the control valve.

The speed of the valve position change is limited so that it is constant with realistic valve action times.

4. LEVEL CONTROL OF THE FEEDWATER TANK

The model of the water level controller of the feedwater tank is shown in Figure 14.



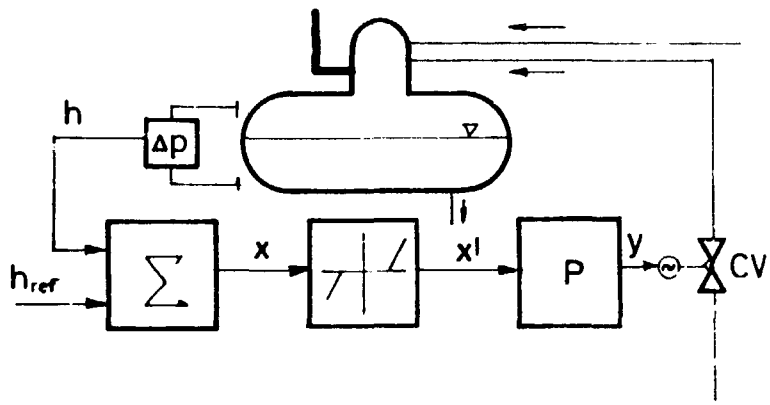
FW - feedwater
FWT - feedwater tank
LPHE - low pressure reheaters
/simplified representation/
CV - water level control valve
P - desalinated water pump

Fig. 14

During the nominal, steady state of the plant, desalinated water is allowed to flow into the secondary circuit to compensate the constant sedimentation losses. This water flows into the feedwater tank through the control valve. The desalination system, however, is not explicitly represented in the simulation, therefore the following assumptions are needed.

The pump supplying the desalinated water line is running continuously at full speed, and a constant pressure is maintained at the inlet of the control valve; similarly, the temperature of the desalinated water is taker as constant.

The block diagram of the algorithm for the feedwater tank water level control is shown in Figure 15.



FWT - feedwater tank
CV - control valve

Fig.15.

The deviation of the feedwater tank water level from its nominal value is used to calculate the required mass flow from the desalinated water system to the tank that will restore the nominal level in the tank. With pressures known the required mass flow determines the necessary cross section of the valve to be opened. The valve characteristic that gives the open cross section as a function of valve shaft position is simply obtained.

The mass flow rate of the desalinated water into the feedwater tank is finally modified by taking into account the limitation due to the realistic valve speed.

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