NUREG/IA-0017

International Agreement Report

Assessment of TRAC-PF1/MOD1 Against a Loss-of-Grid Transient in Ringhals 4 Power Plant

Prepared by
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Office of Nuclear Regulatory Research
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Washington, DC 20555

July 1989

Prepared as part of
The Agreement on Research Participation and Technical Exchange
under the International Thermal-Hydraulic Code Assessment
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STUDSVIK ENERGITEKNIK AB

Project BIV - 670219

Swedish State Power Board

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ICAP

Assessment of TRAC-PF1/MOD1 Against a Loss of Grid Transient in Ringhals 4 Power Plant.

ABSTRACT

A loss of grid transient in a three loop Westinghouse PWR has been simulated with the frozen version of TRAC-PF1/MOD1 computer code. The results reveal the capability of the code to qualitatively predict the different pertinent phenomena and the data comparison was quite encouraging. Accurate predictions of the system response required careful determination of the boundary conditions simulating the turbine governor valves and steam dump valves behaviour. An explicit modeling of the steam generator internals was also found to be important for the results. It was also revealed that the pressurizer system including spray and heaters and their operation should be modeled in some detail for proper response.

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NP111 M-LK

EXECUTIVE SUMMARY

A TRAC-PF1/MOD1 simulation has been conducted to assess the capability of the code to predict a loss of grid transient.

The measured data was obtained from a loss of grid test at full power operation conducted in Ringhals 4 power plant. Ringhals 4 is a Westinghouse PWR with three loops and two turbines of Stal-Laval design. The nominal thermal power is 2775 MW and 915 MW electrical. It is equipped with three Westinghouse steam generators model D3 with a feedwater preheater section located at the cold leg side and a division is made of the feedwater flow between this lower feedwater inlet and the top inlet at the upper part of the downcomer.

During the test the total feedwater was apportioned so that about 10% of the flow was delivered to the top inlet and the rest to the preheater. The circulation ratio at this conditions was about 2.43. The test was manually initiated by releasing the station breakers thereby isolating the plant from the external grid. At that instant about 95% of the turbine load was rejected and the plant started to automatically regulate to a power level corresponding to the house load demand by means of steam dumping and control rod insertion.

In the TRAC-simulation only a single loop representation was used and the core was modeled with a neutron point kinetics specified with beginning-of-cycle conditions. The complete model comprised 30 components made up by 133 nodes with the boundary condition components excluded.

The boundary conditions were taken from the test recordings and were:

- The flow area of the turbine governor valves
- The flow area of the steam dump valves
- The reactor coolant pump revolution speed
- The feedwater flow and temperature
- The control rod position
- The spray cooling power in the pressurizer steam volume.

For the calculation the control rod position was converted into a table providing the control rod reactivity versus time.

The result of the simulation revealed the importance of proper modeling of steam generator secondary side internals and valve characteristics. Also modeling of the signal processing devices was found to be of importance for simulating time delays. Although adequate information to provide a proper modeling of the time delays was scarce the outcome of the data comparison was quite encouraging. On the average the calculated steam flow was within about 8 %, the calculated steam line pressure within 2 % and the calculated steam generator level within about 10 % of measured values. On the primary side the calculated mass flow was within about 3 %, the calculated core power within 3 %, the pressurizer level within 1% and the calculated pressure within about 5 % of measured values.

From the run statistics it was found that the 40 s transient made use of 550 time steps requiring 1813 CPU-seconds on a CDC CYBER 180-835 computer. The time step sizes were forced to be within the following limits. During the first 10 s of the transient the maximum time step was set to 0.05 s while for the rest of the transient 0.1 s was specified.

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1 INTRODUCTION

The International Thermal-Hydraulic Code Assessment and Applications Program (ICAP) is being conducted by several countries and coordinated by the USNRC. The goal of ICAP is to make quantitative statements regarding the accuracy of the current state-of-the-art thermal-hydraulic computer programs developed under the auspices of the USNRC.

Sweden's contributions to ICAP relate both to TRAC-PWR (1) and RELAP5 (2). The assessment calculations of RELAP5 are being conducted by Studsvik Energiteknik AB for the Swedish Nuclear Power Inspectorate while the TRAC calculations are being carried out as a joint effort between the Swedish State Power Board and Studsvik. The assessment matrix is shown in Table 1.

In this report the results of an assessment of TRAC-PF1/MOD1 against a loss of grid transient is presented. The ability of TRAC to simulate this transient is assessed by comparison to data from a loss of grid test conducted at full power condition in the Ringhals 4 power plant.

The background for the transient is the Swedish electrical grid blackout which occurred on Dec 27, 1983. Because of a grid disturbance in the south of Sweden scram occurred in eight out of nine operating nuclear reactors and a relatively long time passed before the reactor plants could resume the electricity production.

The event illustrated the advantage in being able to transfer the reactors from nominal service load into a stable operation mode with external load rejected and maintaining this mode while the external grid is being restored. In this mode the turbine load corresponds to the in-plant electricity consumption only (giving it the name "house turbine operation") and excess steam is dumped to the condenser. During the "December blackout" the success of this kind of operation would have shortened the outage times considerably.

Already in the design specifications of the plants the possibility of "house turbine operation" was recognized. Thus sufficient dump system capacity is provided to accomodate such an operation. However, the transition to "house turbine operation" has previously always failed resulting in tripping of the reactor. Review of the "blackout" event and previously unsuccessful tests indicated that the transient interaction between important system components and control systems was the main reason for small or non-existent trip margins.

After modification of the steam dump valve control along with introduction of some other measures a successful "house-turbine" transition test was demonstrated in Ringhals 4 PWR on Sept 7, 1985. The test was carried out at beginning-of-cycle fuel conditions. The plant recording system had been extended making the test suitable for code assessment purposes. This test is used as an assessment case for the frozen version of TRAC-PF1/MOD1.

A description of the plant and the test transient is given in section 2. The nodalization is described in section 3 and the steady-state calculation is reviewed in section 4. Data comparison is outlined in section 5 and the conclusions are given in section 6.

Table 1

ICAP Assessment Matrix - Sweden

Code	Facility	Type		Description
		Sep.effect	Integral	
RELAP5	Marviken21	X		Subcooled Critical Flow
RELAP5	Marviken11	X		Critical Flow, level swell
RELAP5	FIX-II		X	Recirculation Line (10%) break
RELAP5	FIX-II		X	Recirculation Line (31%) break
RELAP5	FIX-II		X	Recirculation Line (200%) break
RELAP5	LOFT		X	Cold Leg Break (4") pumps off
RELAP5	LOFT		X	Cold Leg Break (4") pumps on
RELAP5	FRIGG	X		Subcooled Void Distribution
RELAP5	FRIGG	X		Critical Heat Flux
RELAP5	RIT	X		Post Dryout Heat Transfer
TRAC-PF1	Ringhals		X	Loss of Load

2 PLANT AND TEST DESCRIPTION

Ringhals 4 is a 3-loop, 2 turbine PWR of Westinghouse-Stal Laval design. The power is normally 2775 MW thermal and 915 MW electrical. It is equipped with three Westinghouse steam generators model D3 with a feedwater preheater section located at the cold leg side. The feedwater is divided between the top feedwater inlet, which enters into the upper part of the downcomer, and the preheater section at the lower end of the riser. Normally only a smaller part of the total feedwater flow is delivered to the top inlet and the rest to the preheater.

In the preheater section the flow is apportioned due to the flow restrictions in support plates etc. According to specifications at nominal load and no top feedwater 54.5 % of the flow is fed to the upper part of the riser (U-tube boiler section) while the remaining flow is fed downward and enters the lower end of the riser on the hotleg side where it mixes with the downcomer flow. The circulation ratio at this condition is specified to be 2.43.

The "house turbine" transition test was conducted with the plant initially operating at full power and about equally loaded turbines. The external 400 kV electrical grid was connected to the plant in a way to facilitate manual initialization of the test.

The test was initiated by manually releasing the station breakers thereby isolating the plant from the external grid. At that instant about 95 % of the turbine load was rejected while still the steam flow was maintained and consequently the turbines started to accelerate. The turbine acceleration limiter was activated causing fast closure of the turbine governor valves. Subsequently these valves were automatically adjusted to cover the in-plant load demand. Also the control rod insertion system and the steam dump system were activated.

The turbine overspeed resulted in an increase of the internal electrical grid frequency which in turn caused an instantaneous overspeed of the reactor coolant pumps. The increased reactor coolant flow resulted in a somewhat lower core temperature and thus an increase in reactor power occurred. After a few seconds the reactor power started to decrease because of reduced pump speed which originated from the turbine frequency regulation, control rod insertion of the D-bank and increased moderator temperature.

The mismatch between the core power and the turbine load was controlled by dumping steam to the condensers. Due to delays in the control and steam dump activation system a temporary interruption of the steam flow and heat removal occurred when the turbine governor valves were closed. Thus the heat was temporarily stored in the system and the pressure increased.

Once the steam dump valves were activated, the produced steam could be removed and the pressure decreased. The control rod insertion of the D-bank continued at maximum speed until a signal on low rod position was achieved at about 35 s into transient. At the reception of this signal the emergency boration was manually initiated as well as manual regulation of the control rods for prolonged "house turbine operation".

After 15 minutes stable operating conditions were maintained with a core power level about 40 percent and the test was terminated. Unfortunately the reactor scrammed during the synchronization of the generators with the external grid frequency. Due to imbalances in the magnetization of the generators low voltage on the internal grid occurred which resulted in a shut down of the ordinary generators and a start up of the auxiliary diesel power supply. This sequence also implied that one reactor coolant pump stopped and the reactor was scrammed on the occurrence of low coolant flow in one loop concurrent with high core power level (higher than 34 %).

The major subject of concern for this type of transients is the power mismatch in the beginning of the transient caused by interacting processes associated with closure of the turbine governor valves and opening of the steam dump valves. Review of the "December blackout" and analyses of unsuccessful load rejection tests revealed that the major cause for large transient overshoots and scram were delays in the steam dump system. These delays resulted in a heat build up in the primary coolant and a challenge to the DNB protection system.

Also the protection limit on negative rate of change in neutron flux could be challenged because of the increased moderator temperature. The decrease in neutron flux would be accentuated by the decrease in pump speed and the insertion of control rods. Analysis has shown that the limit on high positive rate of change in neutron flux could be challenged when the pump speed first increased due to the turbine frequency swings.

Prior to the test modifications of the hardware in the plant were made in order to reduce the delays in the steam dump system. Also setpoints were modified to increase the operational margin for both positive and negative rate of change in neutron flux. The limit on negative rate was changed from -5 per cent to -8 while on the positive side the limit was increased by a corresponding amount.

3 CODE AND MODEL DESCRIPTION

The simulation of the "house turbine" transition was made with version 12.4 of the TRAC-PF1/MOD1 computer code (2). The program was run on a CDC CYBER 180-835 computer under NOS 2.5 operating system with no SCM and LCM partition of the memory. Instead the central processor primary memory was used together with an extended memory capability. TRAC was also locally modified to allow writing of signal variables and control block output on a separate file for later plotting with a separate program. Thus the EXCON and TRAP programs were not used for producing the graphics.

In the simulation only a single loop representation was used as shown in Figure 1. Differences between the three loops were considered to produce effects of secondary order during the transient. It should be noted that trip margins are usually dependent on conditions in individual loops. For instance, the trip on DNB-protection is initiated by the conditions in the two most extreme loops. Since the symmetry between the loops was not perfect this will influence the comparison between calculated and measured trip margins.

The TRAC-model of Ringhals 4 nuclear steam supply system is depicted in Figures 1 and 2. The nodalization comprised 14 components with 73 nodes on the primary side and 16 components with 60 nodes on the secondary side making a total of 30 components with 133 nodes if the boundary condition components are excluded.

3.1 Primary system nodalization

The reactor core, denoted by component 5, was divided into seven vertical nodes; five nodes representing the active core and one unheated inlet and outlet node respectively. The neutron kinetics was simulated by a point model with beginning-of-cycle characteristics.

The upper plenum (component 6) and hot leg inlet was divided into three nodes. The hot leg and surge line, denoted 710 , was represented by a tee-component with five nodes in each branch.

The pressurizer was modeled by two pipe components; the upper one representing the part of the pressurizer expected to contain steam only throughout the transient. The water phase occupied about 50 per cent of the pressurizer height. This nodalization was chosen to be able to introduce coolant spray explicitly into the steam phase. The spray effect was provided by means of a predefined negative power added to the fluid (steam) in component 60. Two PORVs (power operated relief valves) were modeled (components 52 and 53) on top of the pressurizer. For this specific transient the PORVs did not open and the valves were assumed to be blocked in the calculation.

The primary side of the steam generator is modeled by 18 nodes; one each for the inlet and outlet plena and sixteen nodes for the U-tube.

The cold leg leading from the steam generator to the vessel inlet was represented by 10 nodes; five on each side of the recirculation pump. The vessel inlet section was modeled by three nodes, and the downcomer and lower plenum were represented by two nodes and one node, respectively.

3.2 Secondary system nodalization

An outline of the steam generator is depicted in Figure 2. Feedwater was supplied to the steam generator at two locations. Ten percent of the feedwater was supplied as top feed into the downcomer and ninety percent into the preheater section near the outlet of the cold leg side of the U-tube. In the model the ratio between the top feed and preheater feed was kept constant even under transient conditions because the actual distribution was unknown.

The hot leg boiler and the explicitly modeled preheater section were both divided into five nodes with the node boundaries located at the same elevations. The U-tube boiler was represented by three nodes. An ideal separator which allowed only steam to escape upwards was assumed at the top of the riser. A connection was made between the separator node and the upper separator drain flow path. The downcomer was represented by totally nine nodes.

The steam generator level measurement represented by a differential pressure, was explicitly modeled in order to estimate dynamic contributions from flow and mass content.

The steam line was divided into a number of tee-components and the secondary pressure (steam line pressure) was measured in the tee-component 752. Also safety and relief valves were connected to the steam line. None of these valves were activated during the transient although the activation logic was modeled. The part of the steam line denoted by 753 was represented by a valve component with two nodes simulating the main steam stop valve.

The steam flow was divided into two streams - one for each turbine - in the steam line header (754) which was represented by three nodes. The line for each turbine was further split into two flow paths - one containing the turbine valve and the other containing the dump valve. Time dependent characteristics of these valves were given as boundary conditions for the thermal-hydraulic problem.

4 STEADY-STATE CALCULATION

Prior to the transient simulation the TRAC model was adjusted to replicate the plant stationary pre-test conditions. This was done by means of a step-wise procedure starting with identification of the appropriate steady-state of the TRAC steam generator component which handled all the primary-to-secondary heat transfer (enclosed by dashed line in Figure 2). It is usually difficult to attain a specified heat transfer without applying special measures in this type of modeling. The reason can partly be found in the donor cell formulation but also in the substantial amount of counterflow and crossflow in the secondary side riser. The crossflow enhances the heat transfer considerably and is not taken into account in the ordinary heat transfer correlations.

Key parameters in the steady-state adjustments were the primary temperature, secondary pressure and the heat transfer distribution on the hot-leg boiler, the preheater, and U-tube boiler. By increasing the heat transfer area in these three sections the total measured heat transfer could be attained at a given primary flow as well as the apportionment of the heat transfer between the sections according to design specifications. Evaluation of flow distribution from the preheater revealed that 55 percent of the feedwater flow would enter the U-tube boiler while the rest would enter the hot-leg boiler. Appropriate loss coefficients were applied to the preheater to achieve this ratio. Also, by applying appropriate loss coefficients, the specified flow distribution on the secondary side was met. It was found that the nominal heat transfer areas in the hot-leg boiler and the U-tube boiler had to be increased by factors 1.33 and 1.559, respectively. The multipliers for the preheater were 1.29 in the upper part, 3.46 at the feed water inlet and 1.82 in the lower part. The high number at the feed water inlet was justified by an expected large crossflow at this location.

The rest of the steam generator components were added and, by adjusting the downcomer loss coefficients and the secondary side liquid inventory, the specified circulation ratio and downcomer liquid level were attained.

The primary side loop and the steam line system were then adjusted to meet the steam generator condition and as a final step the different subsystems were joined together. With only minor corrections the complete model could be brought to the stable plant pre-test condition.

5 DATA COMPARISON

The simulation was made using a single loop representation. The measured thermal-hydraulic data were obtained for each loop, thus an averaging procedure had to be applied in order to provide data for an average single loop. The averaged parameters were

- Cold leg temperature
- Hot leg temperature
- Mass flows
- Secondary pressures
- Steam flows
- Feedwater flows and temperatures
- Steam generator levels

It was noted that the averaged measured values had some minor inconsistencies and did not completely satisfy the heat and mass balance. Small shifts in the measured valves were therefore introduced to meet the TRAC-requirements on an adequate steady-state.

5.1 Boundary conditions used in the simulation

The time histories of the boundary conditions were either obtained directly from the test recordings or taken as average values from the three loops. In the simulation the following boundary conditions were used:

- The flow area of the turbine governor valves
- The flow area of the steam dump valves
- The reactor coolant pump revolution speed

- The feedwater flow
- The feedwater temperature
- The control rod position
- The spray cooling power in the upper part of the pressurizer

The areas of the turbine governor valves and the steam dump valves as a function of time are depicted in Figure 3. In the calculations it was assumed that the turbine valves were closed over a period of 0.4 s according to specified characteristics and thereafter remained closed throughout the transient. In reality these valves were fully closed only for some seconds and then regulated to a partly open position corresponding to the plant internal power demand of approximately 5 per cent of nominal power. This was taken into account by adding capacity to the steam dump valves corresponding to this extra steam flow. A reasonable downstream boundary condition for the steam line could thus be provided.

The pump speed as a function of time is given in Figure 4. This boundary condition was taken from the recording in the plant. By comparison with the measured data of the primary massflow it was realized that the recorded pump speed data inherently contained a time delay of approximately 0.95 s which has been accounted for in Figure 4. The oscillatory behaviour of the pump speed was caused by the plant internal electrical grid frequency control that required some time to stabilize.

The feedwater flow boundary conditions were controlled by the steam flow and steam generator level and are given in Figure 5. The feedwater temperature is given in Figure 6. The rapid secondary pressure rise caused void collapse and an apparent low level which at first increased the feedwater flow. Thereafter, the feedwater flow decreased due to decreased steam flow. The feedwater boundary conditions were provided by the two fill-components 741 and 744 by means of trip controlled tables.

In order to use the reactivity as a boundary condition the control rod position, Figure 7, was converted into a trip controlled table giving the reactivity as a function of time.

The spray cooling power, Figure 8, was obtained from a recorded pressurizer control signal. In reality the balance between a small continuous spray flow and control heater power regulates the primary pressure. In the simulated transient a primary pressure increase will occur which implies an increased spray flow with a low approximately constant heater power level. Thus the control signal will essentially only influence the spray flow. By assuming a constant spray flow temperature (i.e the cold leg temperature) along with a completely vaporization of the spray in the pressurizer a resulting cooling power was obtained. Through proper specification of the pipe-component a trip controlled table was set up providing the power to be deposited directly in the fluid of component 60. In this way there was no need for modeling and adjusting of the pressurizer control system and the pressurizer system could be kept on a very basic level.

5.2 Results from the simulation

When the turbine valves were closing the steam flow experienced a rapid decrease with a later recovery when the steam dump values opened, Figure 9. The calculated steam flow was obtained by means of a differential pressure between the steam generator dome and a point downstream in the steam line. In order to simulate the registered signal a first order lag function was applied on the differential pressure with a time constant equal to 0.5 s. From Figure 9 it can be seen that the oscillatory behaviour obtained in the beginning of the transient was also found in the calculation but more pronounced although the frequency was about correct. A better comparison could probably be obtained if a more sophisticated signal processing had been simulated.

The dump valve capacity was prior to the simulation adjusted to provide specified flow at specified pressure. The figure reveals the correctness of this capacity at the prolonged dump condition.

Also the pressure upstream the turbine valves, Figure 10, indicates proper dump valve characteristics. The measured oscillations occurring after 14 s resulted from the control function of the turbine governor valves. This function was not simulated in the calculation.

The calculated steam line pressure signal registered at a point just downstream the steam generator showed about the same increase following the closure of the turbine valves as the measured data, Figure 11. An assumed first order lag function with a 0.5 s time constant was applied in the model in order to simulate the pressure signal processing. The maximum pressure was about 6.9 MPa compared to the measured value of 6.7 MPa.

The current calculated pressure response was considerably improved compared to a previous simulation as shown in Figure 11. The main differences between these two simulations were:

- review and modification of the turbine governor valve closure characteristics and the opening sequence of the steam dump valves.
- inclusion of steam line pipe walls and steam generator dome internals. The internals were modeled in some detail to simulate the proper thermal capacity and time response of structural material.
- modeling of the signal processing device. In the plant the measured steam line pressure was actually an output signal from some devices (amplifiers etc). In order to make at least a crude simulation of these devices a first order lag function was used with a 0.5 s time constant.

The second and third items were the major contributors to the obtained improvement and revealed the importance of proper modeling of thermal capacities and of having some understanding concerning the basic signal processing in the plant measurement systems.

The level in the steam generator downcomer calculated as a differential pressure between the two upper pressure taps in Figure 2 is depicted in Figure 12. An initial discrepancy between the measured and calculated values is apparent. When the secondary side pressure increased a void collapse in the riser occurred causing the downcomer level to decrease due to an "U-tube manometer effect". The decrease in downcomer level was less pronounced in the calculation compared to data and the reason for this discrepancy was believed to be twofold:

- the dynamical contribution of the downcomer flow was not properly accounted for in the model. The origin for this dynamical part in the pressure drop was the flow disturbances caused by the top feedwater flow, which added a directional momentum to the downcomer flow, and also the area change from the upper to the lower part of the downcomer. This caused three-dimensional flow effects which were ignored in the one-dimensional modeling approach used in the nodalization.

- the registered downcomer level was a processed differential pressure signal. The actual signal processing was not fully known but the importance of having this properly modeled is indicated in Figure 12 by comparing the unfiltered and filtered signals. The unfiltered signal was taken as the unaffected differential pressure while the filtered signal was obtained by applying a first order lag function with a 0.5 s time constant to the unfiltered signal.

Comparisons between calculated and measured cold and hot leg temperatures are shown in Figure 13. The cold leg temperature increased quickly due to the increase of the secondary side pressure which reduced the heat transfer rate between the primary and secondary side. Later on also the hot leg temperature started to increase when the hotter water arrived. The hot leg temperature increased slower though because of the reduced reactor power. Thereafter the temperatures slowly adjusted to the new power level.

The measured data was delayed because of measurement pipelines, temperature sensors etc. However, in the TRAC simulation the corresponding delay algorithm (a second-order transfer function) was not properly functioning and the calculated curves in Figure 13 were taken as the unaffected process variables. Thus the measured and calculated curves in Figure 13 are not directly comparable.

Figure 14 shows the calculated and measured reactor coolant flow rate normalized to the steady state value. The measured flow signal was the output from an amplifier device which had to be accounted for. The transfer function was not known but an assumed first order lag function with 0.5 s time constant was suggested for the model. Both the unaffected mass flow, taken as a TRAC signal variable, and the output from the assumed transfer function are depicted in Figure 14. It is clear that the unaffected signal simulated the measured curve somewhat better than the processed signal which indicated that the assumed transfer function was not a proper model of the amplifier device.

Also the pump characteristics will influence the mass flow behaviour. TRAC internal pump homologous curves based on data for the LOFT test facility pump were used in the simulation with the rated conditions corresponding to the operating point of the actual plant pumps. It is very probable that the actual plant pump and the LOFT pump are not quite comparable in their behaviour when the pump speed is varying. No comparison between the pump curves was made. However it was believed that differences in pump curves were one reason for the differences between the calculated and measured mass flows.

Another reason could be found in the difficulty of determining a precise time point of the pump speed table in the simulation boundary condition. The scanning frequency used in the pump speed recordings was 1 Hz which results in an uncertainty of an order that could explain the differences between calculated and measured flow, at least during the very first period of the transient.

Comparisons between calculated and measured core nuclear power are depicted in Figure 15. The power increased quickly after start of transient because of the increased pump speed and primary flow which lowered the core-average temperature and improved the neutron moderation. At about 2 s into the transient the maximum power was obtained both in the calculation and in the measurement and the power started to decrease.

The power reduction occurred due to several combined effects. A small contribution came from the control rods that just had started to move into the core. The major contribution originated from the decrease in the primary coolant flow and the arrival of warmer cold leg water to the core.

The initial increase in nuclear power was underpredicted by about 40 MW in the calculation (1.4 %). This could most easily be explained if the moderator temperature reactivity feedback coefficient (α_n) was chosen too small. Apart from this and the uncertainty in the measurement there could be basically two more reasons for the discrepancy:

- A neutron point kinetics model was used to represent the conditions in the core. Thus the axial redistribution of the core coolant temperature when the flow increased was taken into account through weighting functions to arrive at the core coolant average temperature for the kinetics. If a one-dimensional model had been used the nodewise coolant temperature reactivity feedback would have resulted in a change in the axial power distribution and a somewhat different total core power would be expected.

- The increase of the coolant mass flow when the reactor coolant pump speed increased was less pronounced in the simulation than in the plant due to non-matching pump curves. This tendency was also found when comparing the calculated and measured coolant mass flows as outlined above.

From Figure 16 it is revealed that the minimum margin against trip on negative nuclear rate of change was about 2.5 % in the test whereas the corresponding margin was about 3.5 % in the simulation. This curve was essentially the time derivative of the nuclear power with a 2 s time constant included. Thus the discrepancies between the calculated and measured nuclear power curves also influenced the rate curves. The possible underestimate of α_n suggested above could in fact contribute to the discrepancies both for the initial positive peak and the subsequent negative dip of the nuclear rate curves in Figure 16. It should also be noted that the test was performed with a core at beginning-of-cycle. The trip on negative rate of change is more critical at end-of-cycle conditions and the accuracy of the prediction may have to be improved for adequate simulation of such conditions.

The pressurizer level trend was due to having a somewhat higher primary side heat up in the simulation than in the plant as seen for instance in Figure 13. This caused a larger liquid inventory expansion resulting in the somewhat higher level, Figure 17.

Also the pressurizer pressure experienced a rapid increase when the level started to rise but before the pressurizer spray was activated, Figure 18. The registered pressure was a processed signal and in the calculation this was simulated by means of a first order lag function with 0.5 s time constant. It has to be realized that this function was only an unrefined attempt to simulate the signal processing and consequently has to be included when viewing the discrepancy between the calculated and measured signals.

In the model the spray flow was not explicitly taken into account but was reformulated into an effective spray cooling power according to Figure 8. This negative power was added directly to the fluid (i.e. vapor) in component 60. Thus the pressure increase up to about 6 s when the cooling power was activated, could be expressed in terms of an adiabatic compression of an initially saturated steam volume corresponding to the level increase. This also resulted in a few degrees superheated vapor condition at 6 s.

In the plant the ordinary pressurizer control system was influencing the pressurizer response throughout the transient. Thus in the stationary pre-test situation a continuous spray flow was provided which was balanced by the pressurizer heater power. During the initial phase of the pressure increase this spray flow, although small in magnitude, was certainly enough to keep the condensation rate on a level that resulted in a more moderate pressure increase.

Thus it could be expected that the conditions in the upper vapor region of the pressurizer were somewhat dissimilar in the model and the plant during the test. In the model a superheated condition was prevailing with no condensation during the initial phase of the transient resulting in a higher pressure increase. In the actual pressurizer a saturated or somewhat-superheated vapor condition with a continuous condensation could be assumed resulting in a lower pressure increase.

Consequently the comparison between the calculated and measured pressurizer pressure as found in Figure 18 could be improved if the pressurizer system was modeled in more detail. However even if a more realistic modeling approach had been used, analysis has revealed a tendency that TRAC-PF1/MOD1 overpredicts the pressure increase of pressurizer insurges (3). A contributing factor appeared to be too little interfacial heat transfer between the vapor and the subcooled spray flow.

5.3 Estimation of Accuracy and Run Statistics

When making an estimation of the accuracy several sources of uncertainty should be taken into account (4) in order to sort out the code alone accuracy. For the simulation presented herein the major contributors to the total uncertainty were judged to be:

- σ_{exp}, the uncertainty in the measured data
- σ_{input}, the variability of the results due to uncertainty in initial and boundary conditions
- σ_{user}, the variability of the results due to nodalization, time step selection, modeling of head slabs, etc

However none of these were easily quantified for this specific transient. When using a test conducted in a real plant one has to limit the data comparison to those parameters that are monitored by the plant ordinary monitoring system and also realizing that this system is not intended for providing data of the kind found in test facility data acquisition systems. Different measured parameters were usually represented by signals which were output from some signal processing devices, the characteristics of which were not fully known. These devices of course had some influences on the signal response (time delays) which consequently could not be completely accounted for. Also the data obtained for each of the three loops had to be combined to provide single loop representative values, a procedure that added to the uncertainty of the data. It was also realized that in some cases the data scanning frequency was fairly moderate (up to 1 Hz) resulting in time resolution difficulties. All these were embedded in σ_{exp} thus making it diffucult to estimate.

Also σ_{input} and σ_{user} were found difficult to quantify. The initial conditions were taken from the stationary pre-test recordings after making the averaging to provide data for a single loop.

In this averaging process the small imbalances between the loops were smoothed out and also small adjustments of some parameters had to be done in order to fulfill the TRAC heat balance. Unfortunately the pre-test plant heat balance was not fully known and a more precise evaluation of that condition could not be performed.

The boundary conditions were taken from some recorded signals and thus inherently contained the sources of uncertainty mentioned above to find out the influences of the boundary conditions on the result a sensitivity study would have been required which is very involved when using a complex plant model. No such study was for that reason performed and the same was argued for the nodalization study when estimating σ_{user}. However a combined effect of including steam generator secondary side internals, a simple signal processing simulation and somewhat modified turbine- and steam dump valve area boundary conditions on the steam line pressure can be found in Figure 11.

The curve represented by the previous calculation shows the result without any internals and signal processing whereas these were included in the current calculation along with small corrections of the valve area versus time characteristics. This resulted in a decrease of the peak pressure with about 0.2 MPa and a more close comparison with the measured data.

Only a crude estimate of the total uncertainty was made by viewing measured and calculated signals in the diagrams, Table 2. Thus it could be expected that the code alone accuracy should be somewhat better than the values given in the Table.

Parameter	Uncertainty*) (%)
Core power	3.0
Primary mass flow	3.0
Pressurizer pressure	5.0
Pressurizer level	1.0
Steam flow	8.0
Steam line pressure	2.0
Steam generator level	10.0

*) Absolute value of difference between calculated and measured value related to measured value

Table 2. Estimated uncertainty of some parameters

During the simulation the time step size was limited by means of program input. This was done in order minimize the time delay (one time step) between fulfillment of trip conditions and the resulting actions (opening of valves etc). During the first period of the transient (0-10 s) where the severity was somewhat pronounced a maximum time step size of 0.05 s was allowed while during the rest of transient 0.1 s was specified. For the 40 s transient this resulted in 550 time steps and 1813 CPU-seconds or about 3.3 CPU-seconds per time step.

6 CONCLUSIONS

From the result of an analysis of a loss of grid transient test at full power conducted at Ringhals 4 PWR it is clear that version 12.4 of TRAC-PF1/MOD1 is capable of performing a favourable simulation. When using test results obtained in a real plant it is important to realize that the registered data often are not direct measures of different process variables but the output from some signal processing devices. Thus these devices also had to be modeled in a simulation in order to make a true data comparison. For the present analysis a detailed performance of these devices was not known and a simple first order lag function with an assumed time constant was used in most cases.

Despite these shortcomings the outcome of comparisons between calculated results and test data was quite encouraging although the results could even be improved if a more sophisticated modeling had been used to simulate the signal processing. On the average, the calculated steam flow was within about 8 %, the calculated steam line pressure within 2 % and the calculated steam generator downcomer level response within about 10 % of measured values. On the primary side the calculated mass flow was within about 3 %, the calculated core power within 3 %, the calculated pressurizer level within 1 % and the calculated pressurizer pressure within about 5 % of measured values.

The analysis also revealed the importance of accurate modeling of valve characteristics and operating sequences as well as the modeling of structual material in the steam generators. It was also realized that a more sophisticated modeling of the pressurizer control system including spray and heaters would probably have improved the pressurizer pressure response during the insurge period. In the present analysis an excess pressure increase was obtained during the initial phase of the transient which was believed to be less pronounced if a correct pressurizer control system had been modeled. As a result the vapor in the pressurizer top now was superheated. The degree of superheating would have been quite less if a continuous subcooled spray had been simulated during the compression phase resulting in a condensation process throughout the transient.

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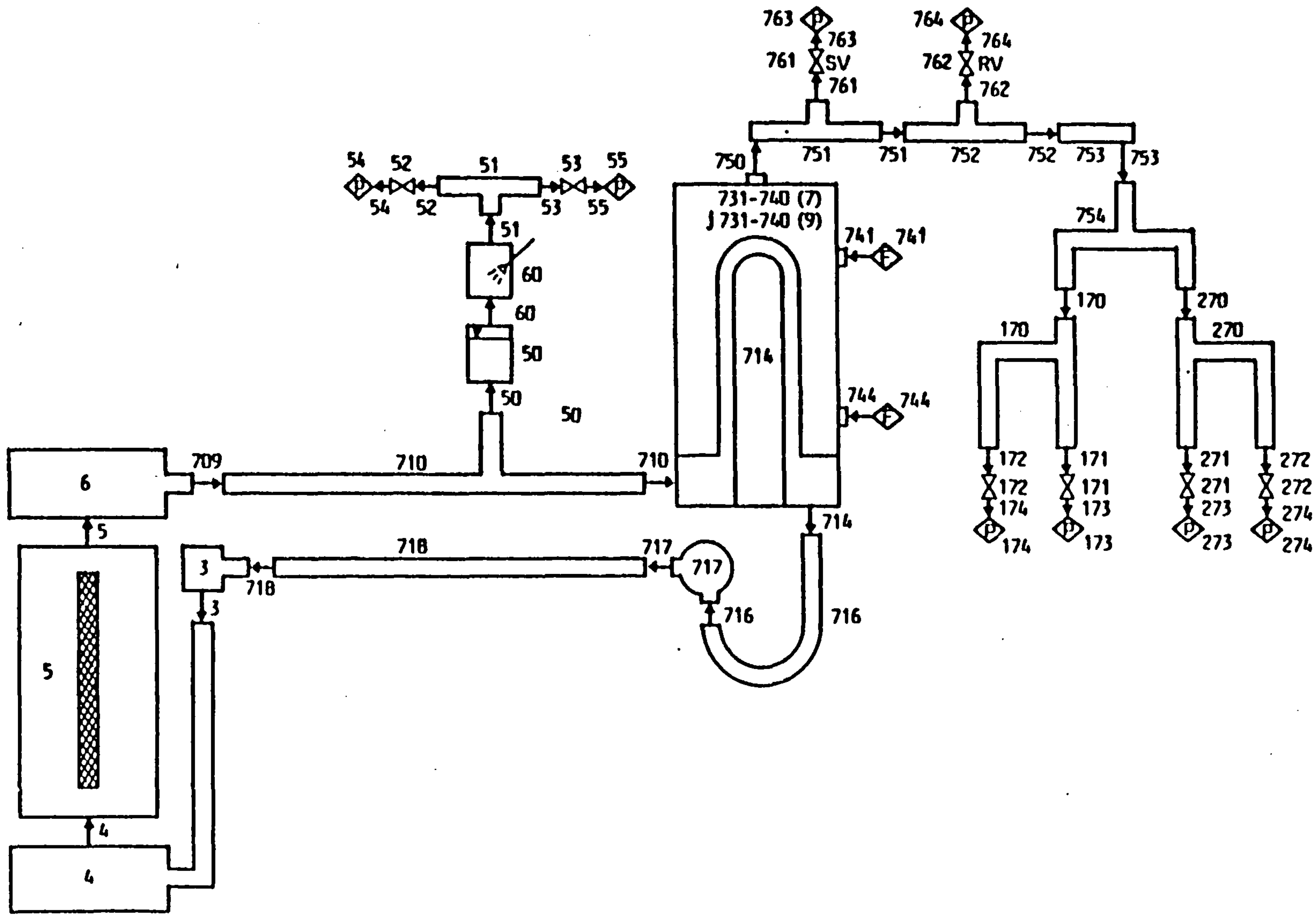

Figure 1. TRAC-PF1/MOD1 Model of Ringhals 4

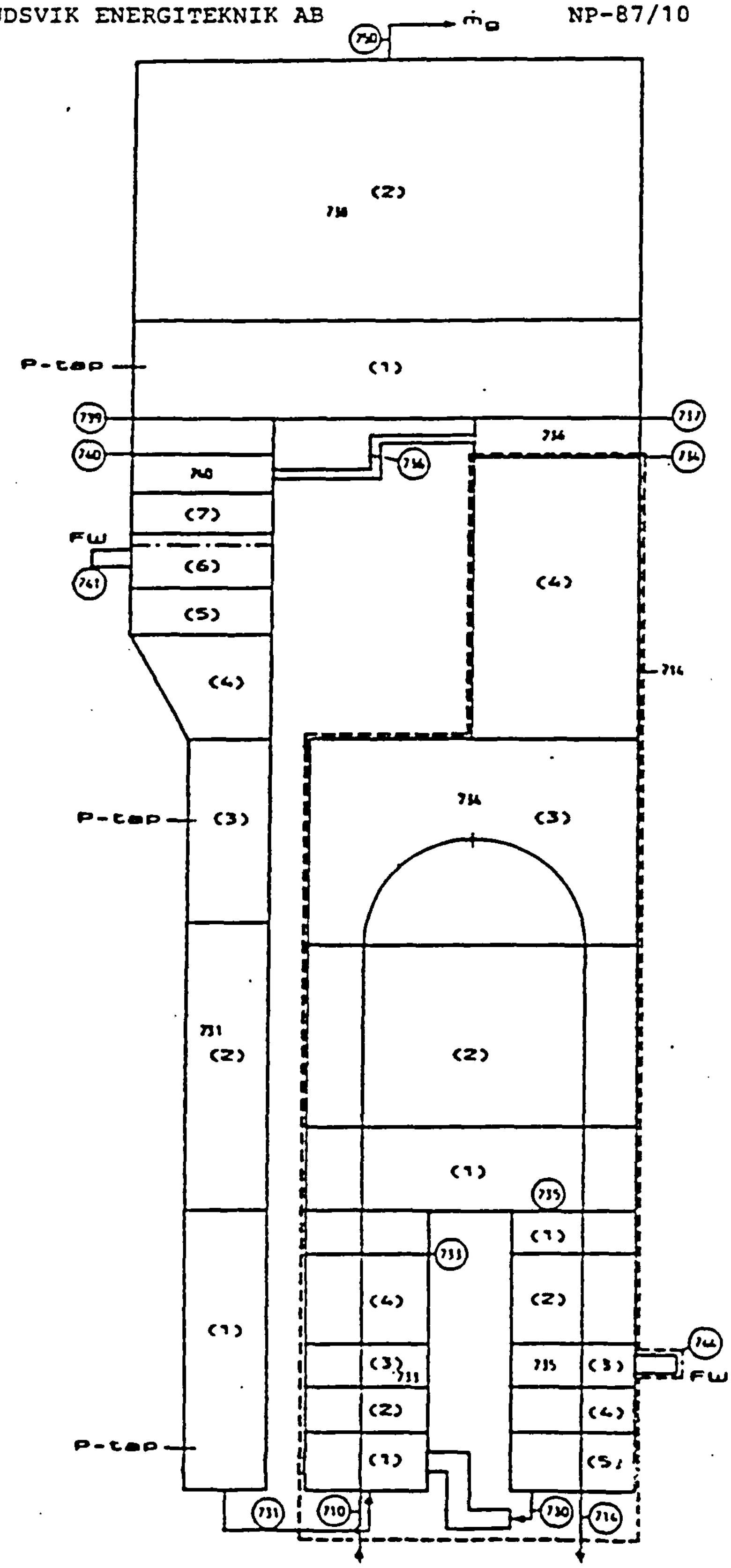

Figure 2. Steam Generator Nodalization

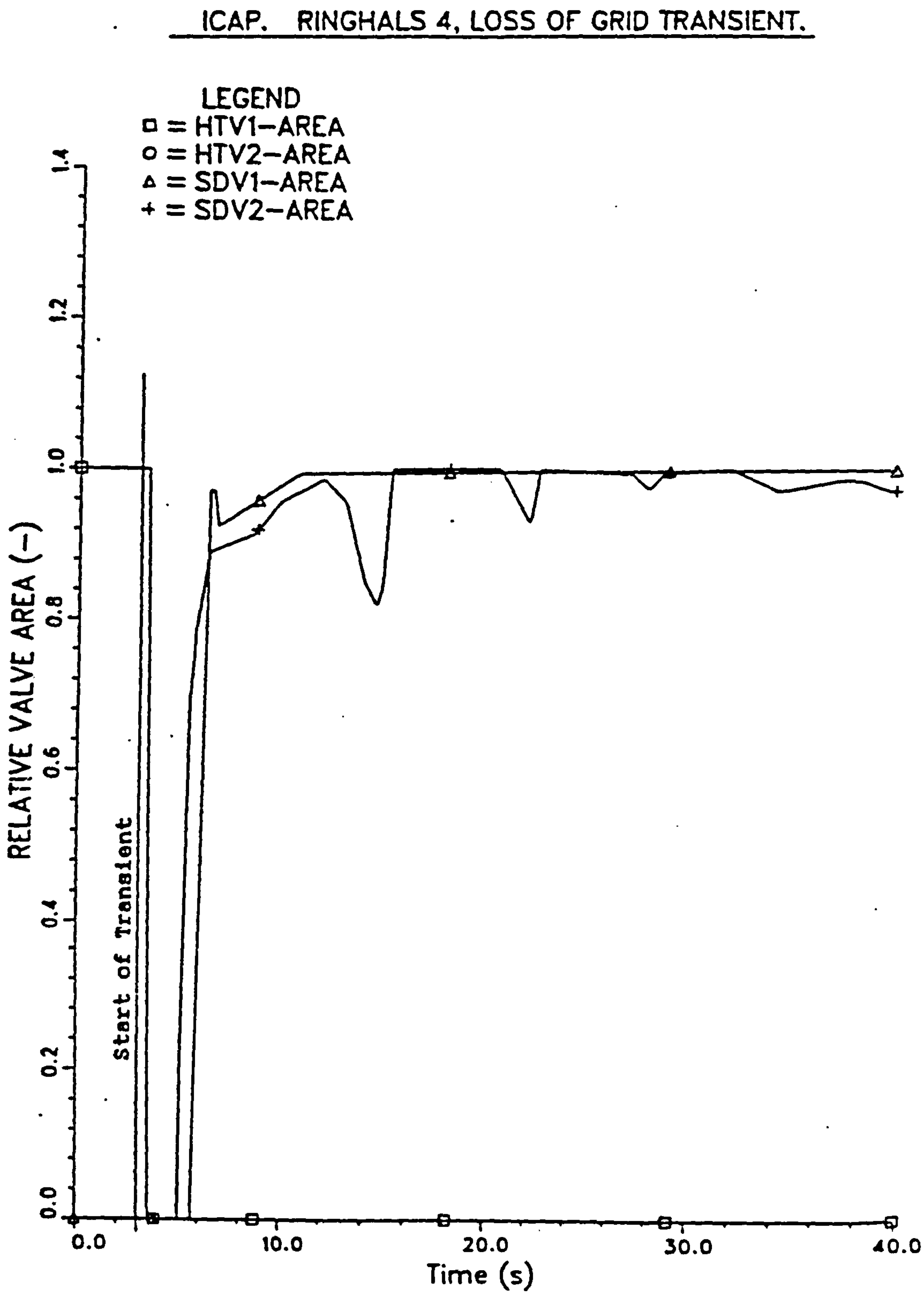

Figure 3. Valve Area Boundary Condition

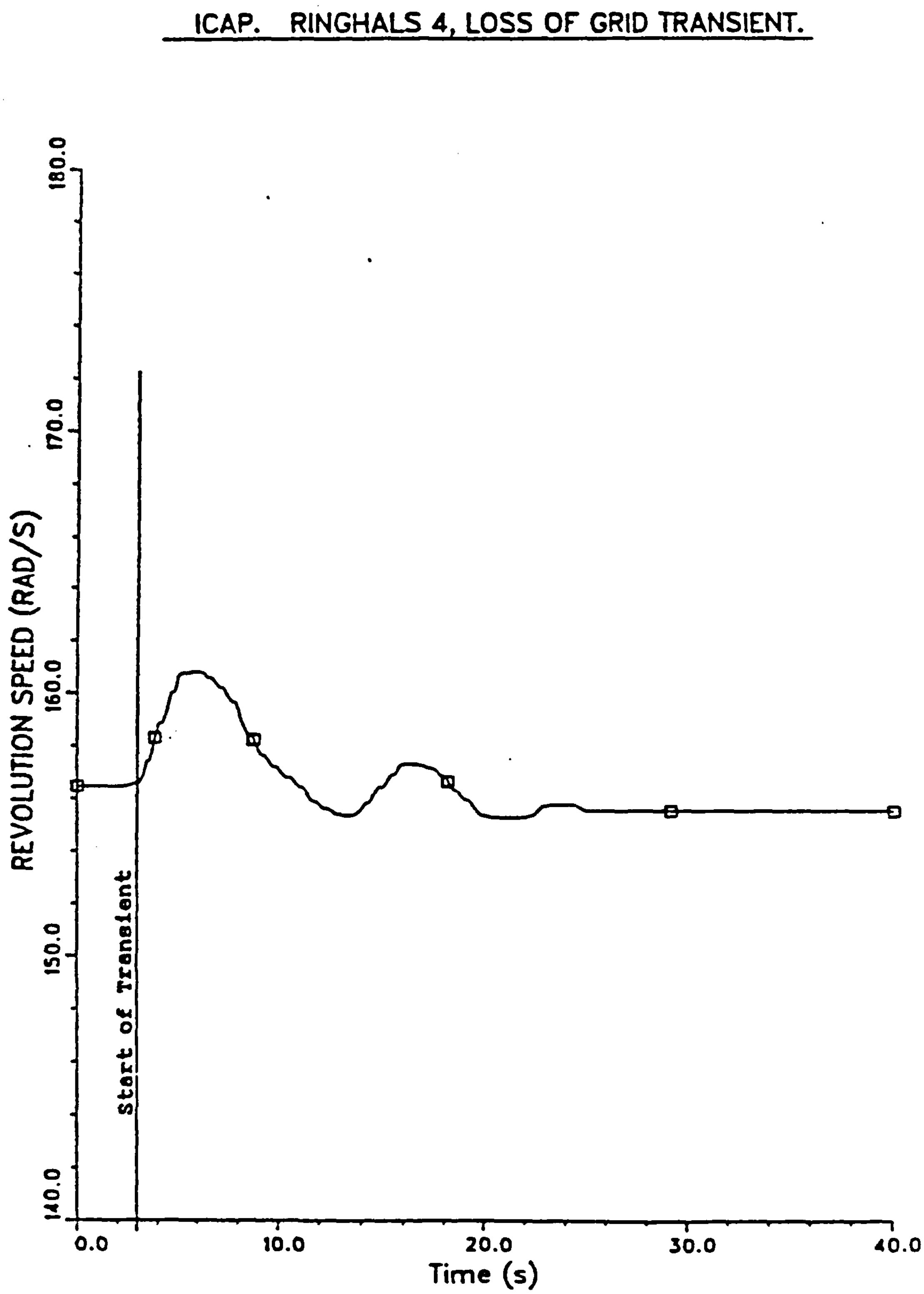

Figure 4. Reactor Coolant Pump Speed Boundary Condition

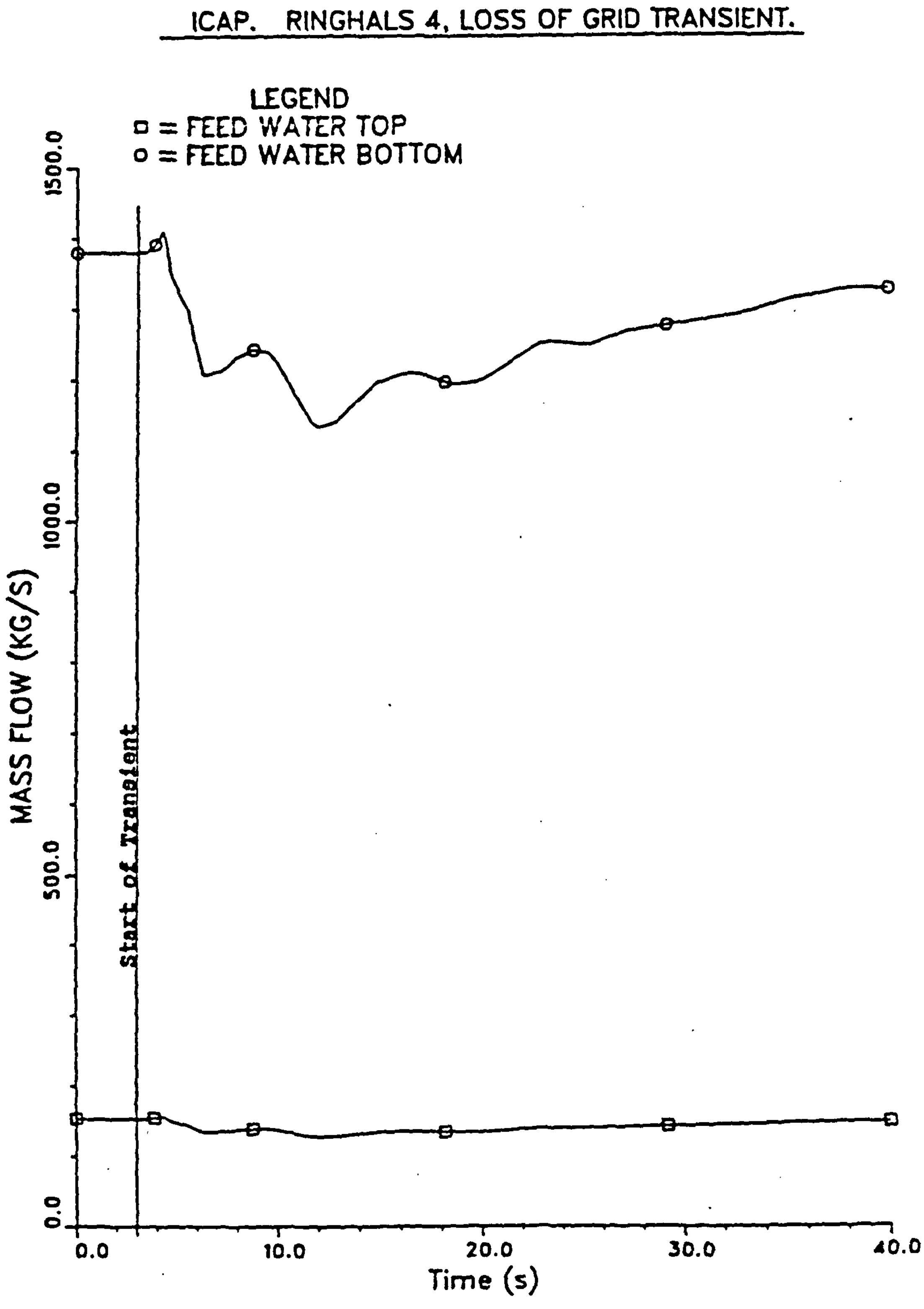

Figure 5. Steam Generator Feed Water Flow Boundary Condition

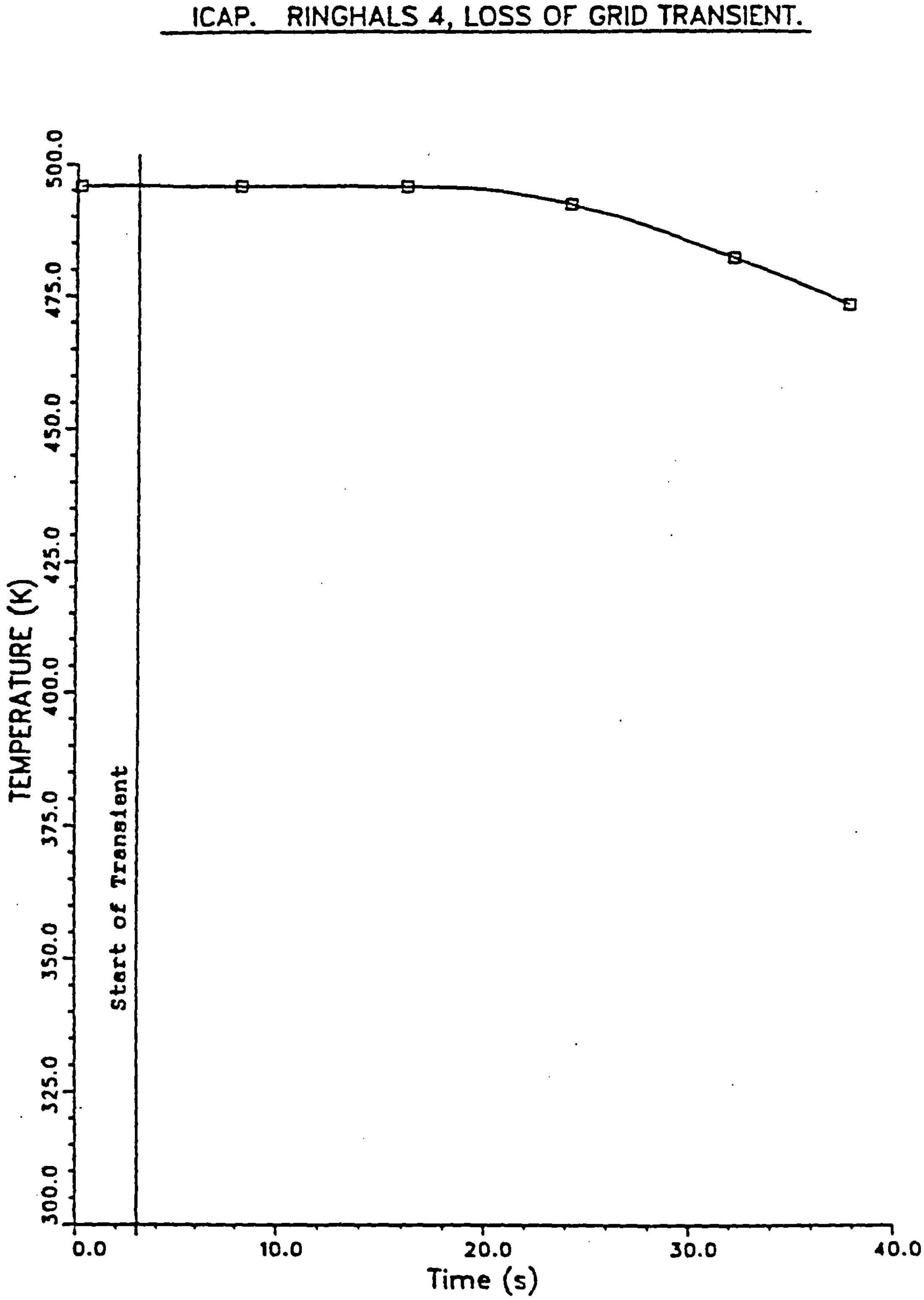

Figure 6. Steam Generator Feed Water Temperature Boundary Condition

ICAP. RINGHALS 4, LOSS OF GRID TRANSIENT.

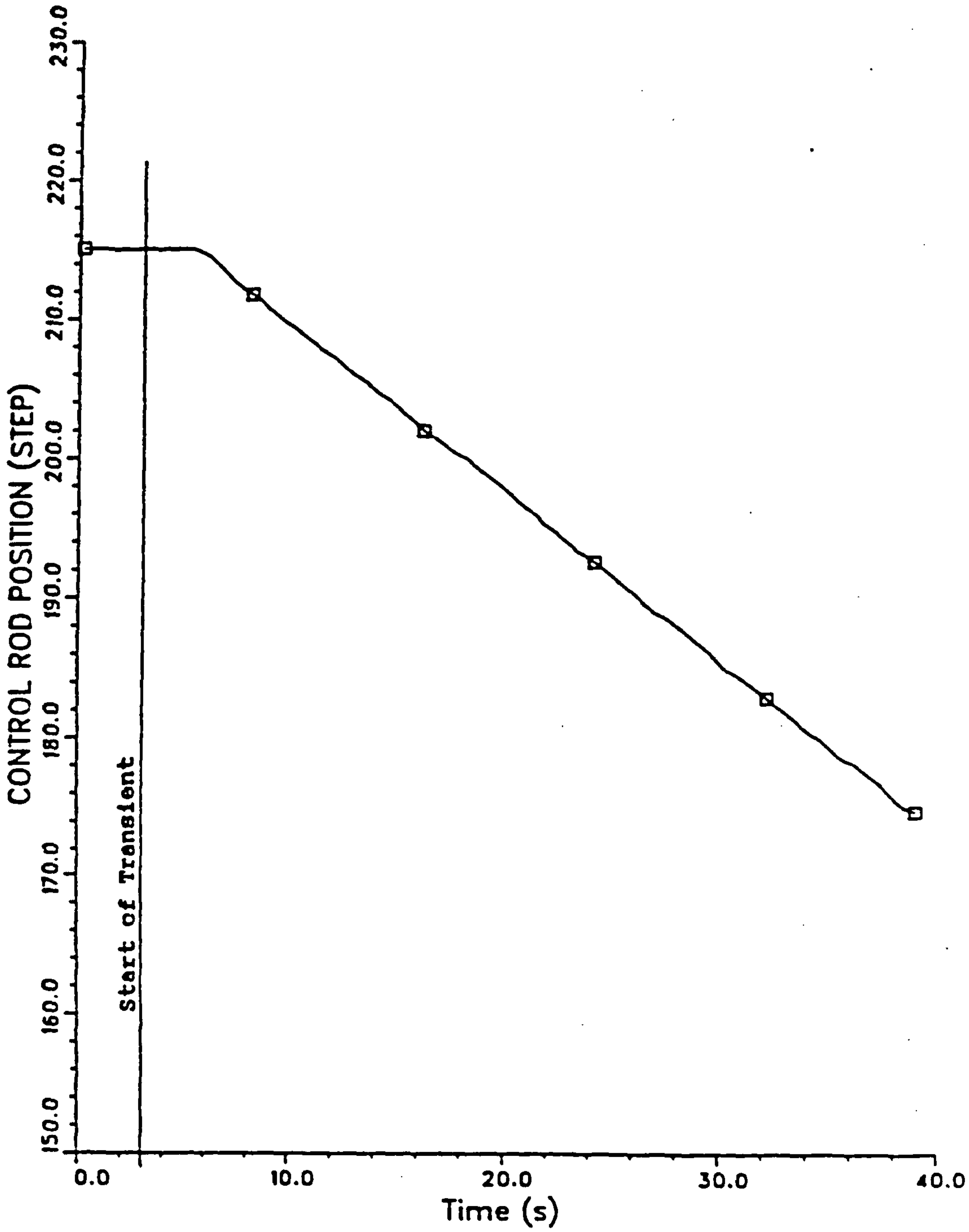

Figure 7. Control Rod Position

ICAP. RINGHALS 4, LOSS OF GRID TRANSIENT.

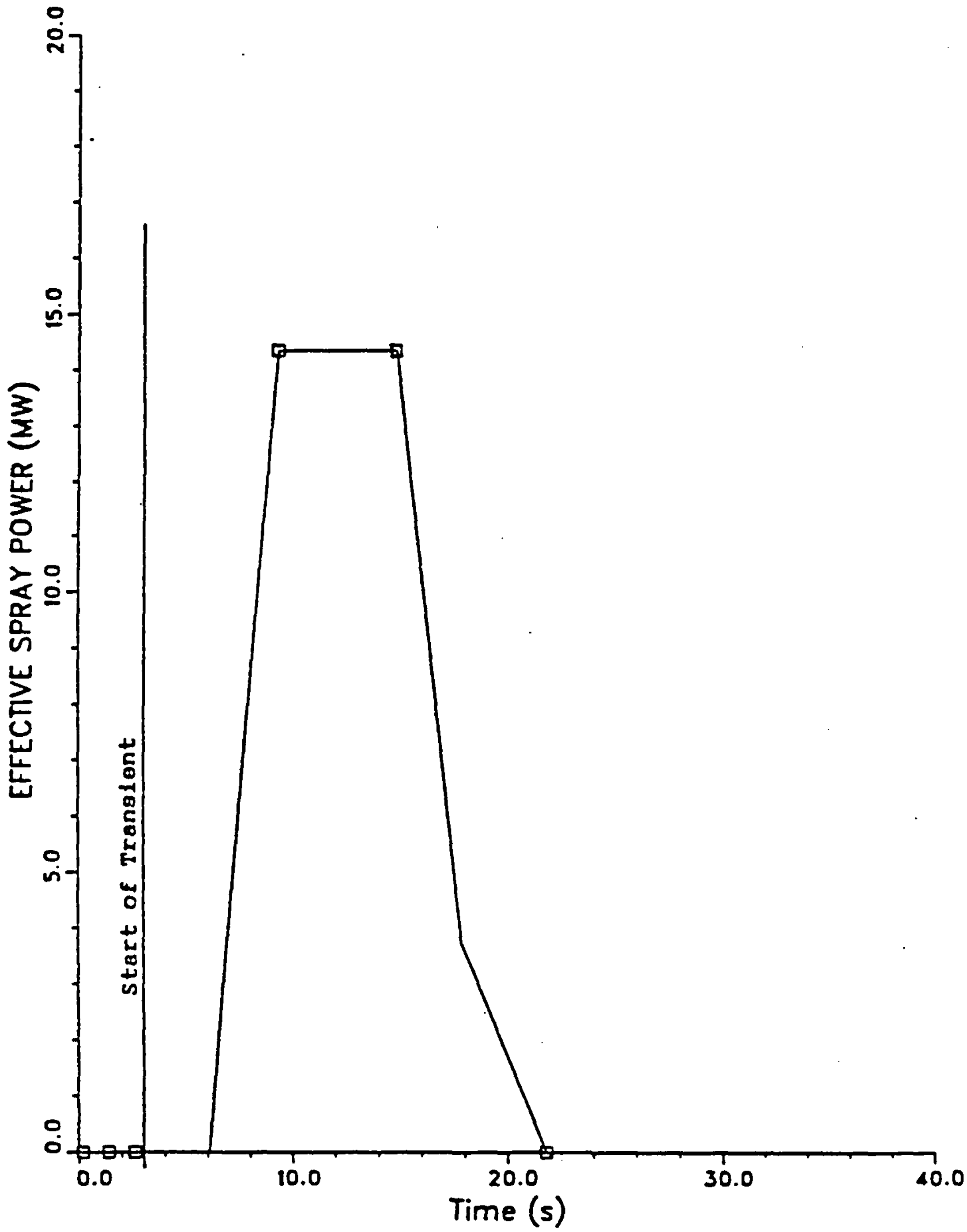

Figure 8. Pressurizer Spray Power Boundary Condition

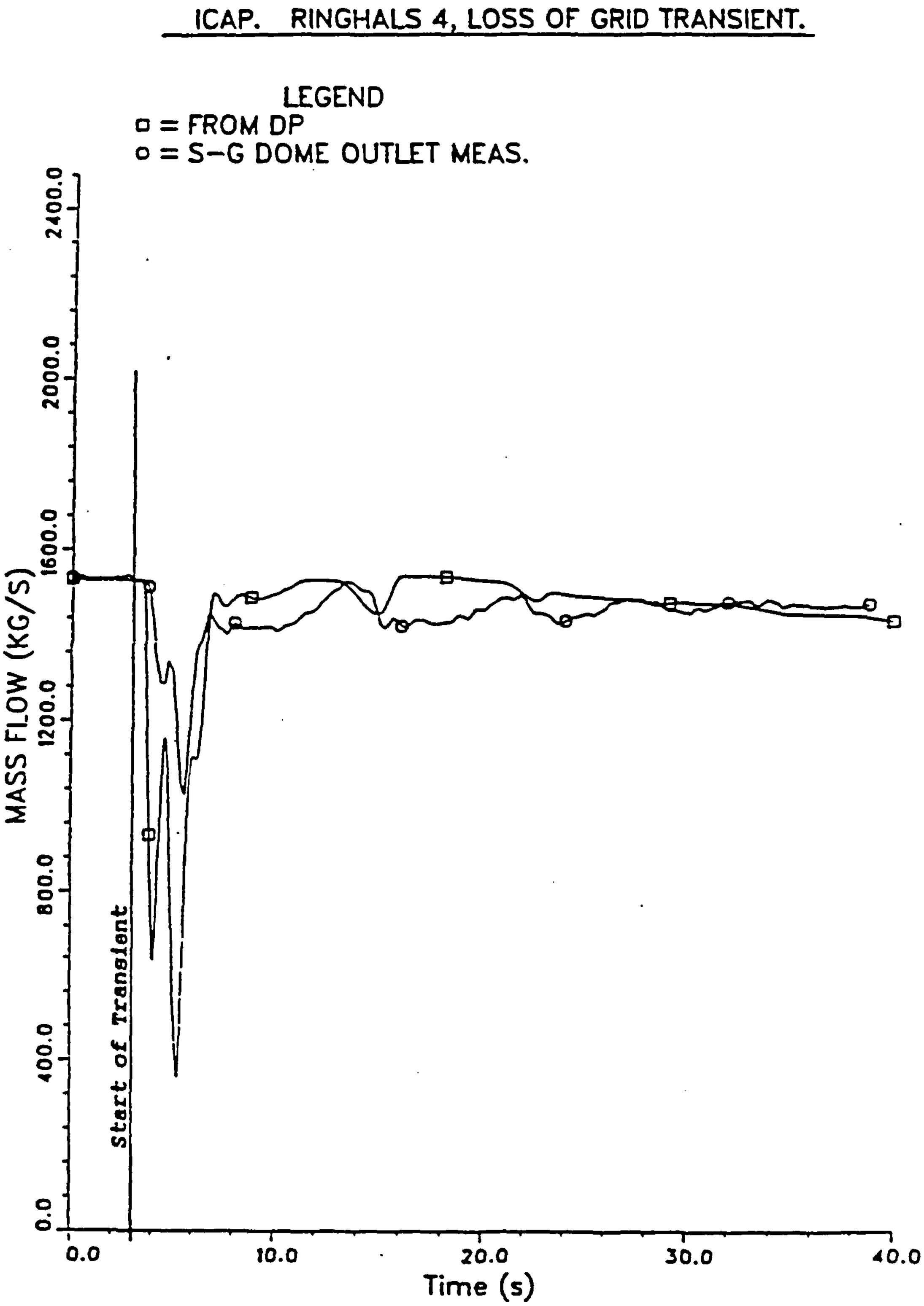

Figure 9. Steam Line Mass Flow

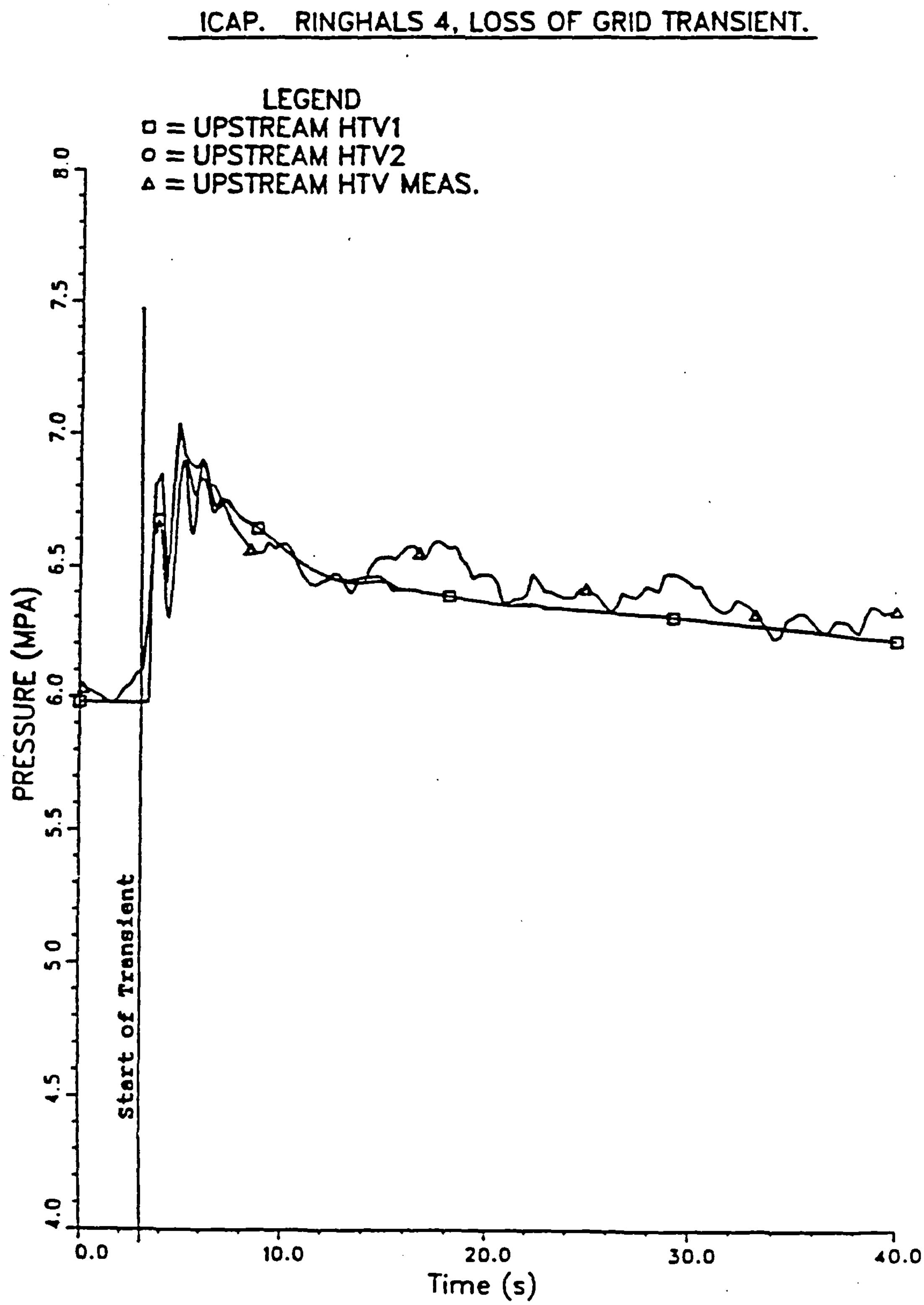

Figure 10. Steam Line Pressure Upstream Turbine Governor Valves

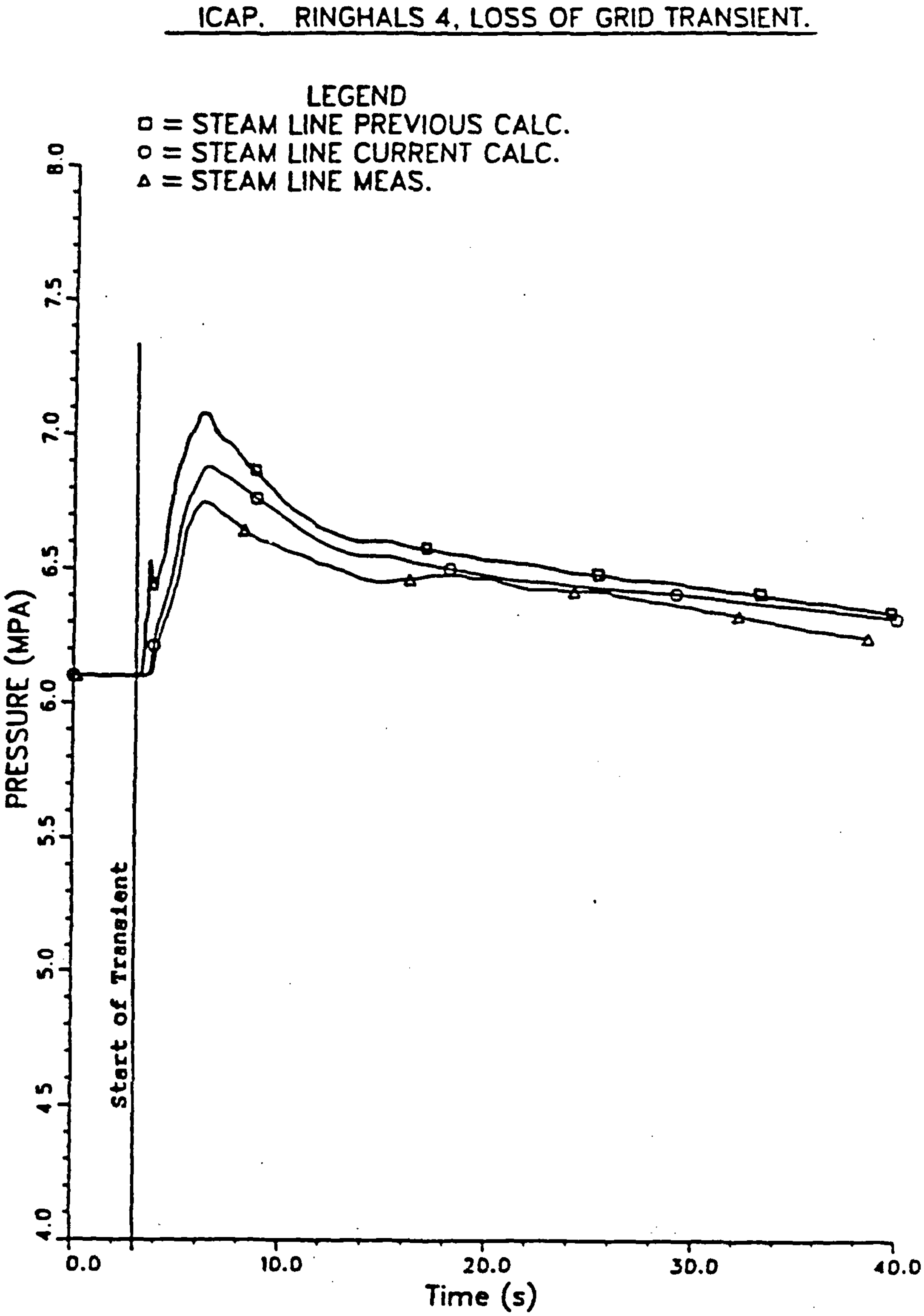

Figure 11. Steam Line Pressure Downstream the Steam Generator

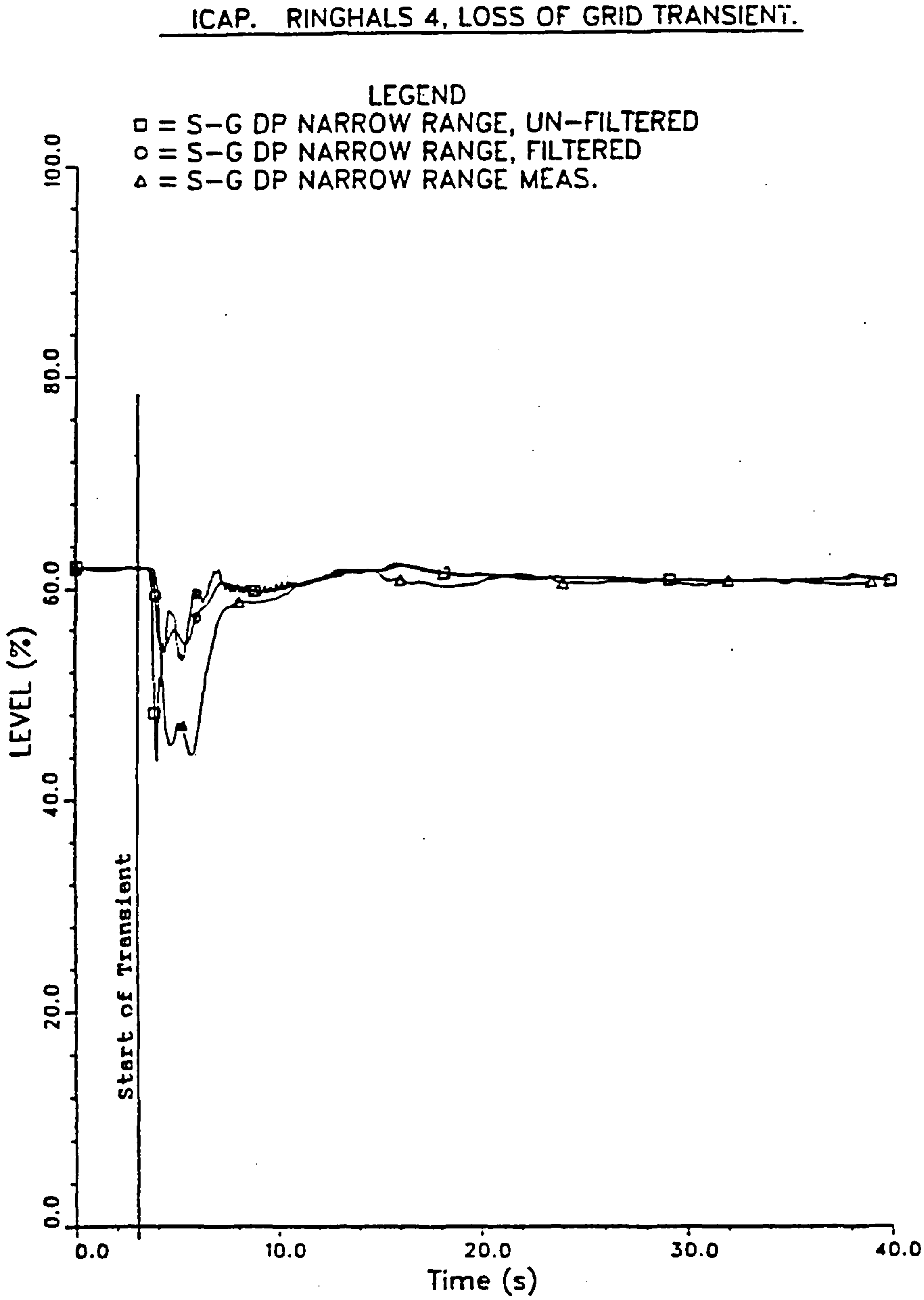

Figure 12. Steam Generator Downcomer Level

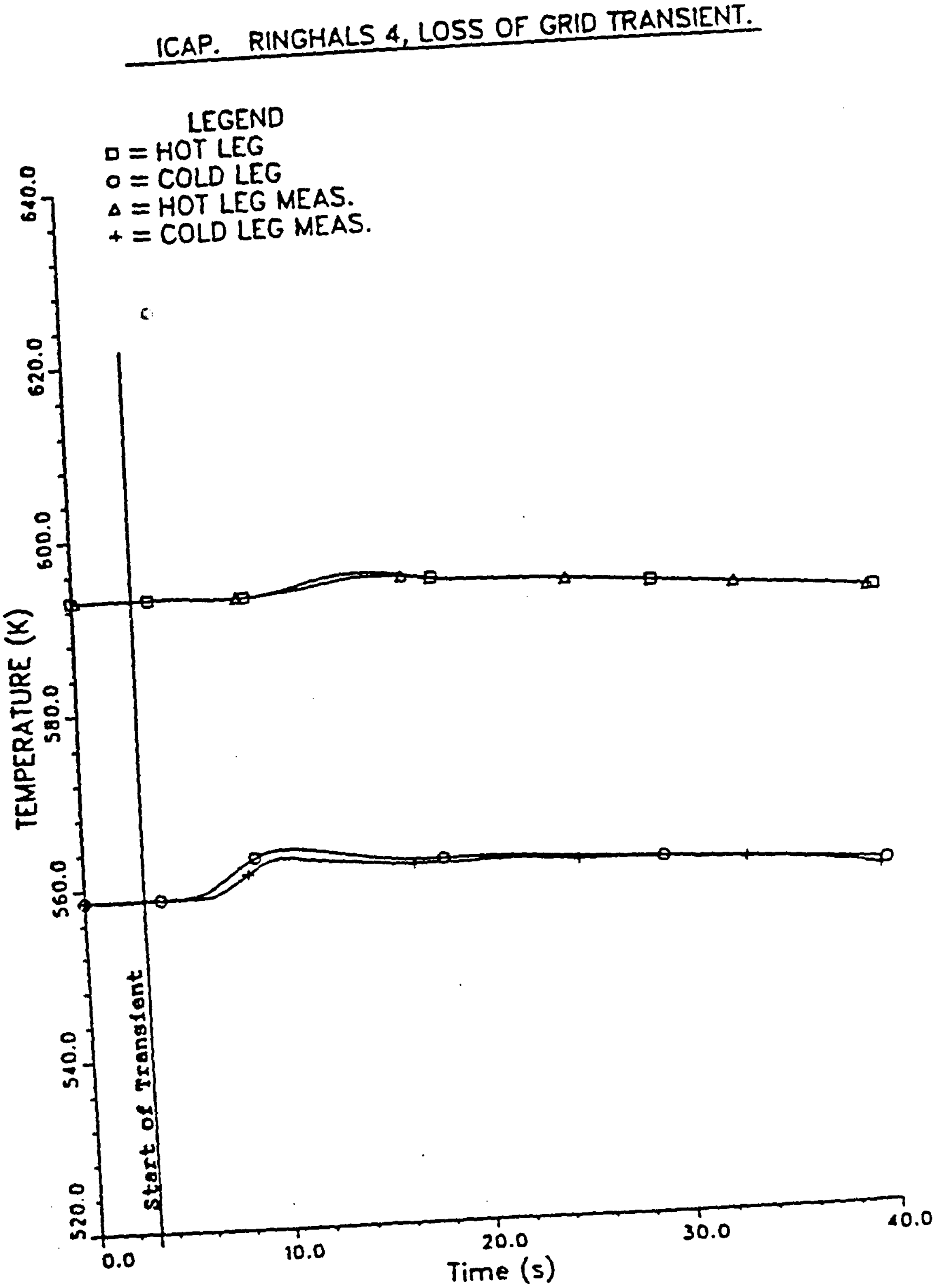

Figure 13. Hot and Cold Leg Temperature

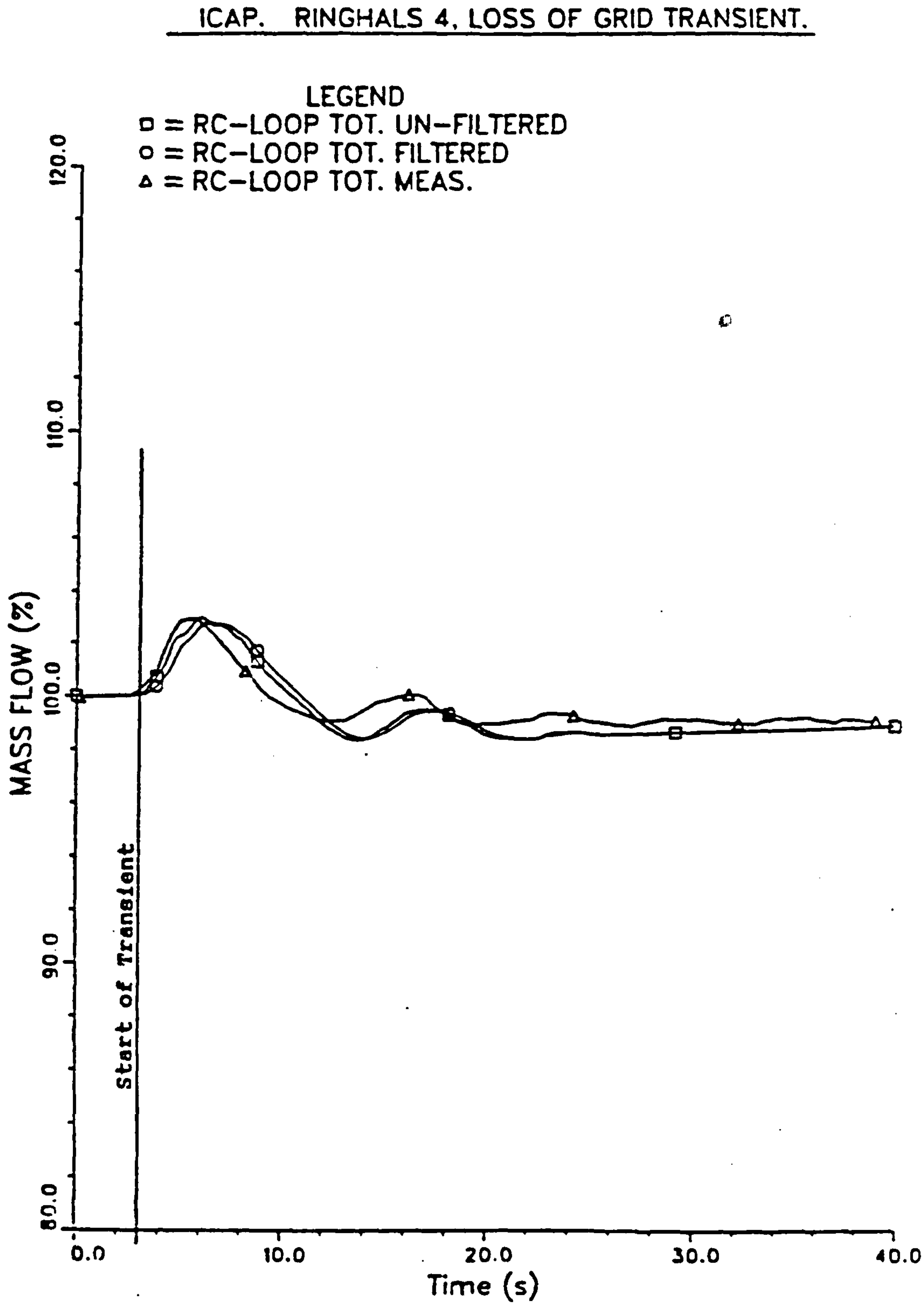

Figure 14. Primary Side Mass Flow

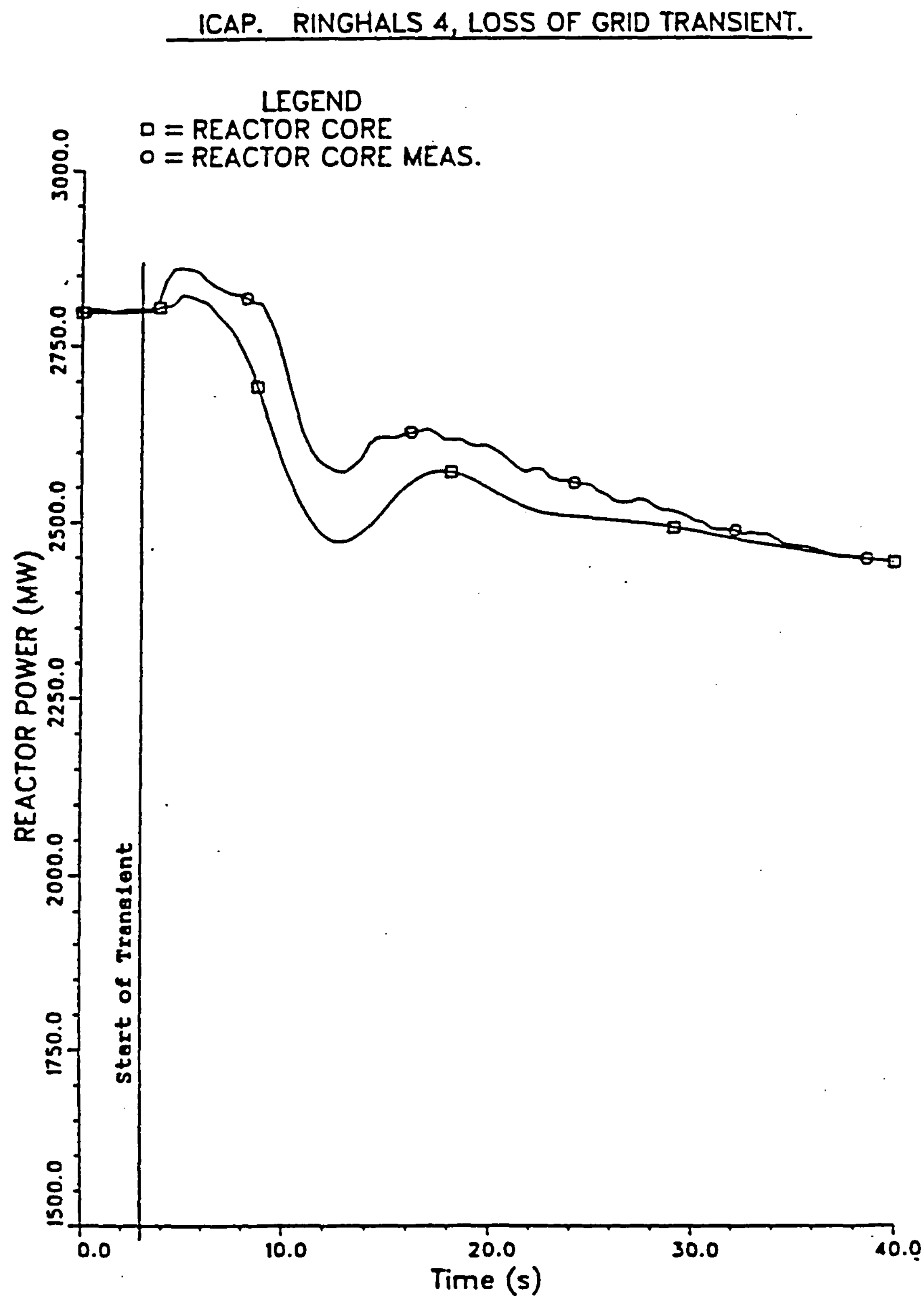

Figure 15. Reactor Core Power

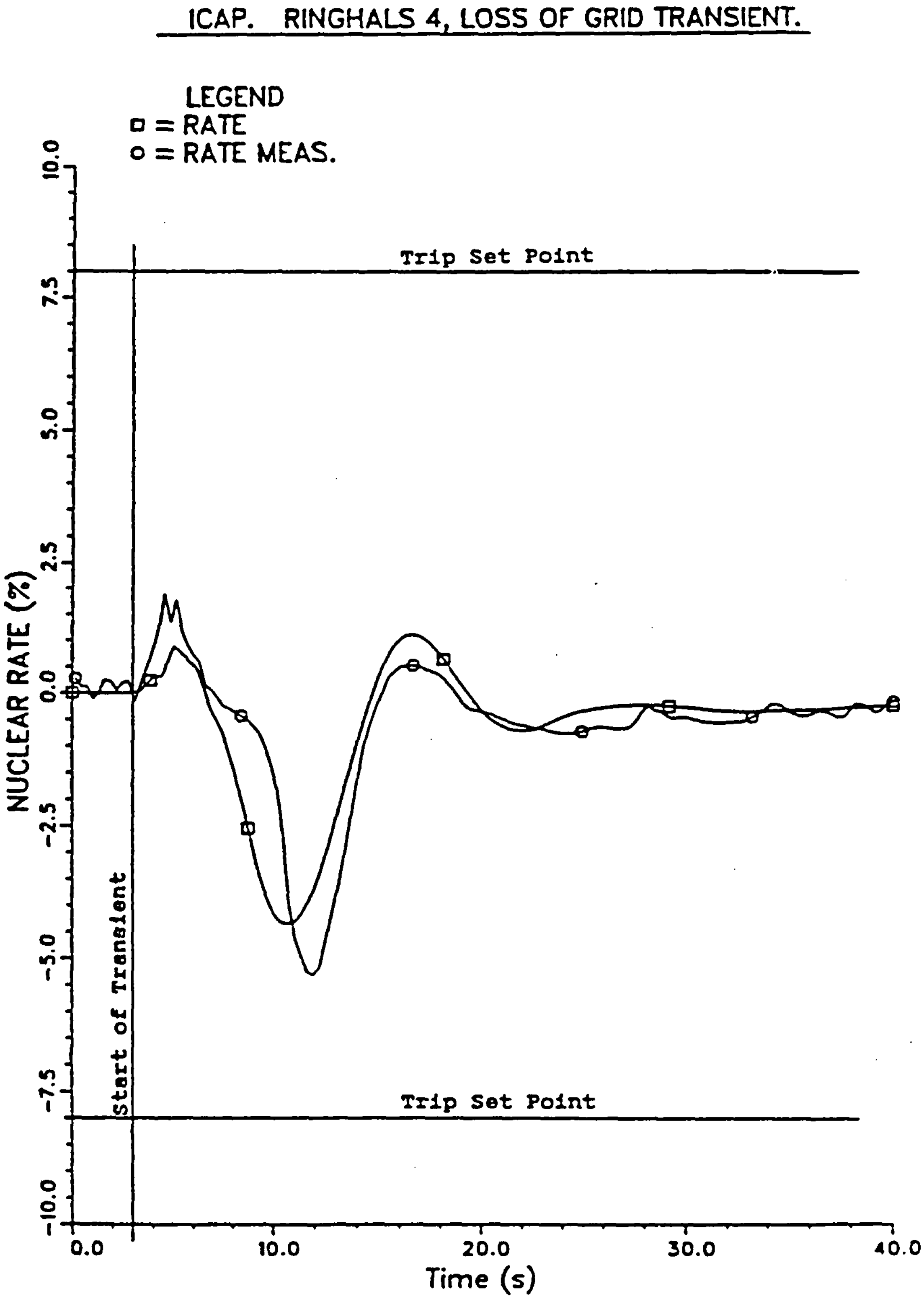

Figure 16. Reactor Core Nuclear Rate

ICAP. RINGHALS 4, LOSS OF GRID TRANSIENT.

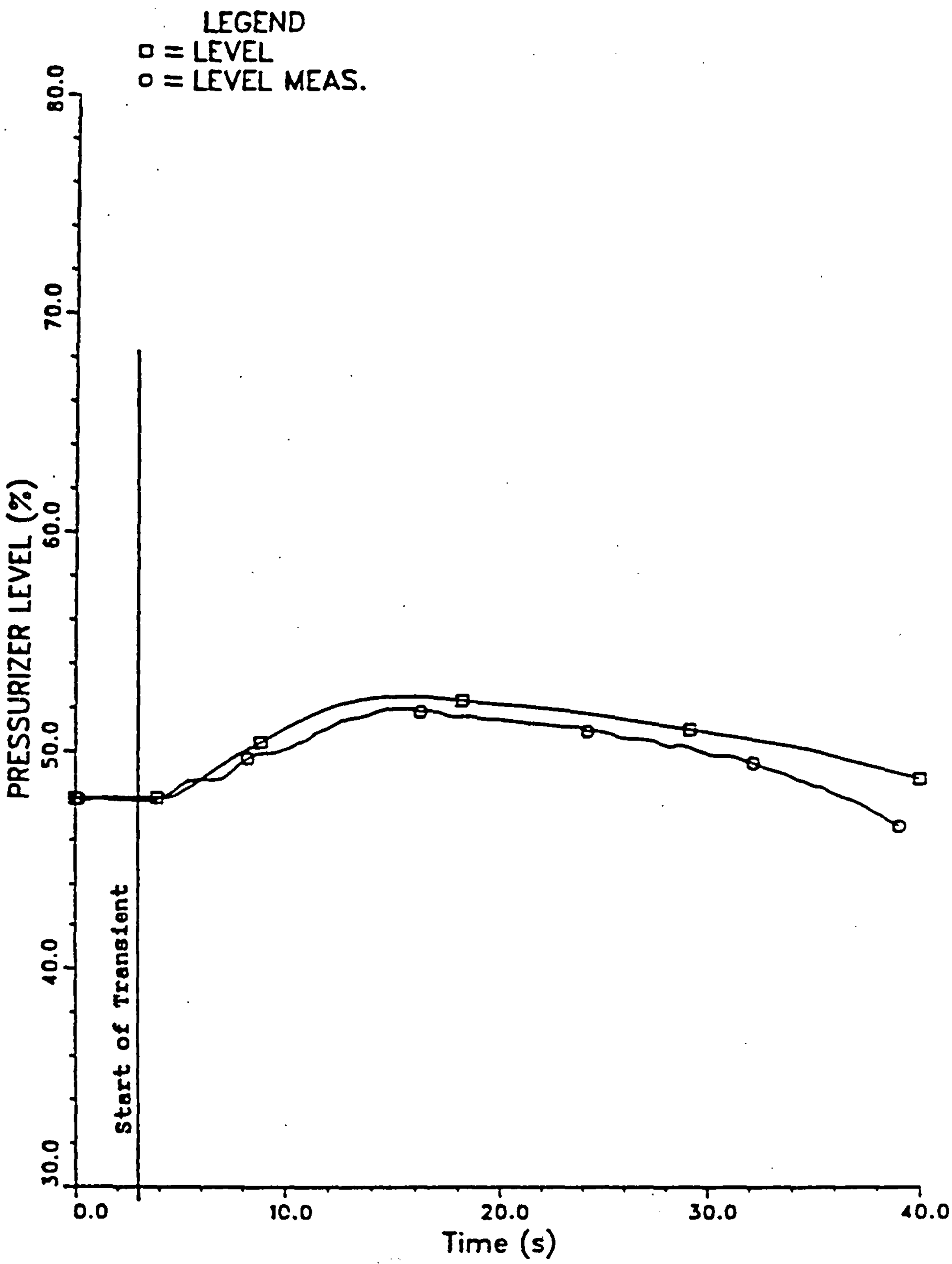

Figure 17. Pressurizer Level

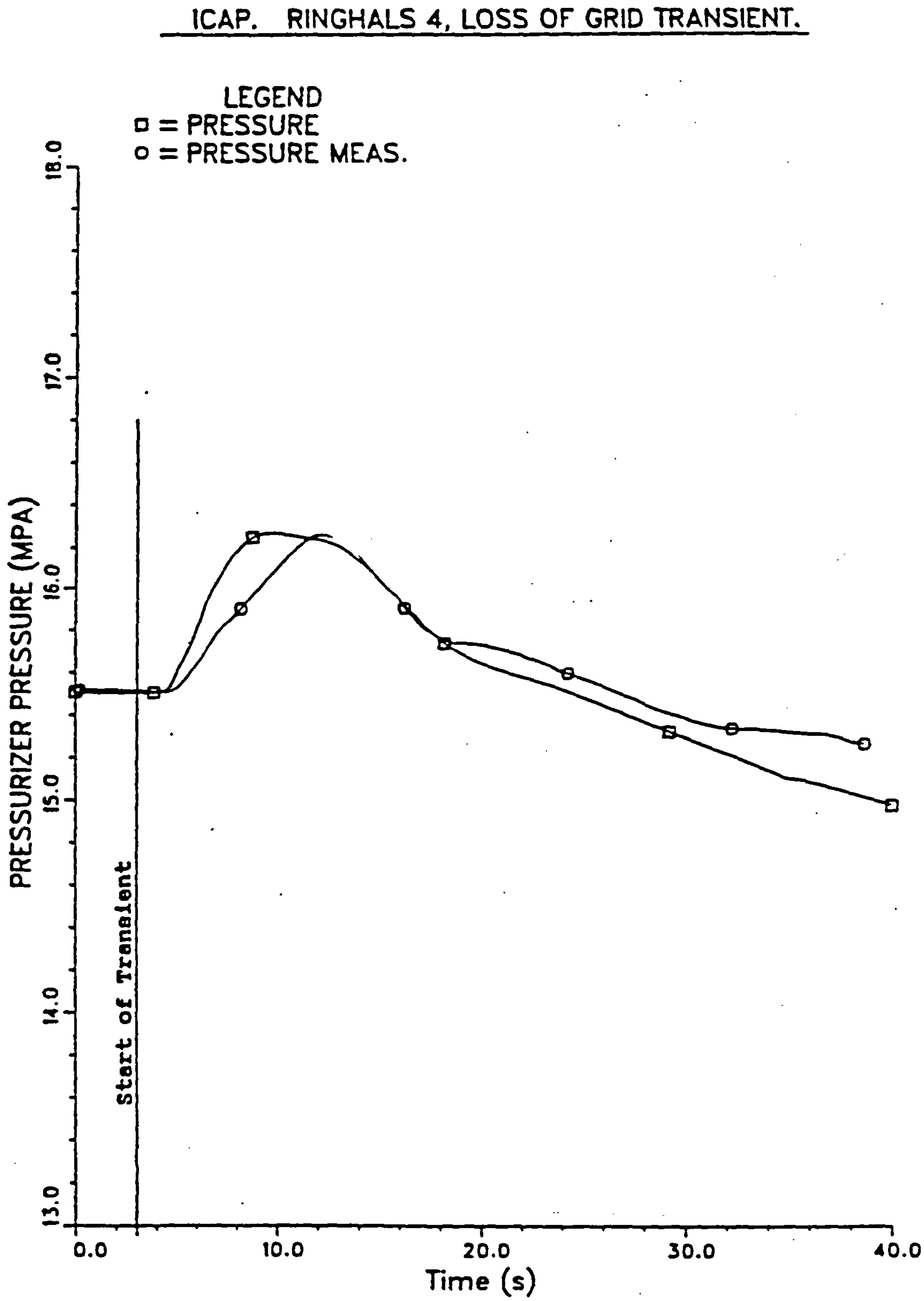

Figure 18. Pressurizer Pressure

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A loss of grid transient in a three loop Westinghouse PWR has been simulated with the frozen version of TRAC-PF1/MOD1 computer code. The results reveal the capability of the code to qualitatively predict the different pertinent phenomena and the data comparison was quite encouraging. Accurate predictions of the system response required careful determination of the boundary conditions simulating the turbine governor valves and steam dump valves behavior. An explicit modeling of the steam generator internals was also found to be important for the results. It was also revealed that the pressurizer system including spray and heaters and their operation should be modeled in some detail for proper response.

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