An Independent Verification of Byron/Braidwood D4 SG Tube Support Plate Differential Pressures during MSLB

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Statement of Disclaimer

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Abstract

The purpose of this calculation is to perform and document an independent assessment of the Westinghouse calculations generated to provide structural loadings on the steam generator tube support plates during limiting transient conditions. The Main steam line break (MSLB) event from hot zero power was determined by the vendor to yield the highest differential pressures across the support plates. The vendor utilized the TRANFLO code for the initial work, and validated their results using the MULTIFLEX computer code. This assessment develops and utilizes methods based primarily on first principles physics to determine bounding differential pressures seen at the most highly loaded TSP. This provides a realistic assessment of the margin inherent in the vendor methods.

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1. Introduction

During a main steam line break event, the rapid blowdown of the faulted steam generator can lead to significant loads on the tube support plates. Westinghouse has performed transient thermal hydraulic calculations on the Byron 1/Braidwood 1 Model D4 steam generators in support of structural calculations regarding the extent of tube support plate deformation. Independent assessment with other computer codes has been performed, although some questions remain, particularly with respect to the margin of safety and the allowances for calculational uncertainties. Therefore, a method of characterizing the loads on the upper support plates based on first principles physics, independent of computer codes, was developed. This report documents the methods created for this purpose and details the results obtained.

2. Methodology/Model Description and Assumptions

2.1 Description of the Problem

The limiting case has been previously determined to be a break of the steam line directly outside the steam generator nozzle, with the generator at initial conditions of hot zero power and normal water level. The D4 steam generator is shown in Figure 1. What is desired is the differential pressure vs. time that exists at the upper support plate during this event. To calculate this differential pressure, one must determine the dynamics of the fluid motion in the tube region following the initiation of the break.

Calculation of the dynamic response of the tube region fluid requires that a number of related issues be addressed. These include characterization of the break flow and transient pressure response of the steam space, acoustic effects both prior to and following initiation of fluid motion, and determination of the differential pressure operating on the bulk fluid in the tube region.

2.2 Time Sequence

An understanding of the time sequence of events following initiation of the break is important to understanding the relationships between the key physical phenomena. Figure 2 provides a depiction of the key events and their relative temporal location for this event. As can be seen, this event can be thought of as consisting of three major regions, each dominated by different physical effects.

The initial phase is the acoustic region, characterized by the establishment of critical flow at the nozzle and initiation of depressurization of the steam regions of the generator, but prior to the initiation of bulk fluid motion. A key occurrence in this region is that a decompression wave traverses the generator, initially at high speed through the contiguous single phase regions. The effect of this decompression wave is to initiate voiding in the fluid, drastically reducing the acoustic velocity, which then determines the pressure response times in the subsequent phases.

The next phase is the bulk fluid motion phase. Given the reduced acoustic velocity of the two phase mixture and the continuing decompression of the steam regions, a differential pressure across the liquid region will occur, causing bulk motion of the fluid. This motion is dominated by momentum effects and pressure losses at the grids and other structures. The fluid will accelerate to maximum velocities early in this phase and then decelerate as viscous effects involve more of the upper structures of the steam generator. Additionally, the decompression rate decreases as time goes on, due to pressure reduction as well as increasing liquid content in the break effluent.

The last phase is the long term behavior. This phase can be thought of as a quasi-steady state condition dominated by mass balance effects. The fluid remaining in the tube regions will flow at a rate comparable to the break flow rate. The velocities at this point are low and decrease with time as the blowdown progresses to completion.

2.3 Initial Conditions and Geometry

The vendor calculations indicate that the limiting case occurs at hot zero power conditions with water levels at normal values. The water level is at 487" , just below the swirl vanes in the separators. The temperature of the water and steam are uniform at 557 F, and saturation conditions are assumed. Key geometric parameters have been derived based on TRANFLO input descriptions and are presented in the table below:

Table 1 Key Geometric parameters of D4 Steam Generator

Parameter	Value
Initial Steam Space Volume	2556.52 ft3
Steam space Path Length	27.745 ft
Liquid Region Path Length	40.583 ft
Tube Bundle flow area	56.45 ft2
TSP flow area	17 ft2
Entrance area of separators	22.01 ft2
TSP loss coefficient	1.08
Separator Entrance loss coeff	13.9
Break Area (restricting Nozzle)	1.388 ft2

2.4 Discussion of Acoustic Phenomena

The break is assumed to occur over a time interval of 1 msec. Since this time interval is too short to assume equilibrium conditions (about 1/100 second or greater), a decompression wave will travel through the steam generator at high speeds. (about 3500 fps in the liquid and 1500 fps in the steam. This will require approximately 40 milliseconds. The result of the passage of this wave will be the generation of voids, requiring about 10 milliseconds to occur. Therefore 50 milliseconds into the event, the initial decompression wave will have traversed the generator and initiated voiding in the liquid regions. This is significant in that once the voiding occurs, the acoustic velocity decreases dramatically. Reference 1 provides a value of 157.5 fps for the speed of a decompression wave in equilibrium saturated water. This speed then dictates the rate at which pressure differentials can develop between the decompressing steam space and the bottom of the fluid regions, since the pressure disturbance propagates at the acoustic speed. Therefore the maximum differential pressure operating on the fluid can be determined by estimating the rate of change of pressure in the steam space and employing the acoustic propagation length of the fluid to determine the time and therefore pressure lag at the bottom of the steam generator.

2.5 Determination of Steam Space Pressure Response

In the initial phases of the blowdown, the steam region pressure response can be readily characterized by treating the steam as a perfect gas and employing formulas for adiabatic blowdown (isentropic expansion) or isothermal blowdown of a pressure vessel (Reference 1). These in fact, give relatively good results in the period of time initially after the break initiates prior to the decompression wave reaching the fluid surface. Once, the fluid surface becomes involved however, the flashing rate leads to significantly lower pressure decay than would be predicted by the simple isentropic formulas. Therefore, alternate methods must be utilized to obtain the steam space pressure response.

A review of methods for determining the vessel dome pressure response indicates that this is generally accomplished via detailed numerical methods. Some textbooks provide plots of vessel pressure ratios, calculated using detailed methods, with dimensional time scales to provide an approximate method to assess the pressure response. Use of this type of approach for this problem yields depressurization rates of approximately 124 psi/sec. The figure with tangent lines drawn from Reference 3 used to establish this depressurization rate is enclosed in the Appendix. The generalized time axis value was based on the break area (1.388 ft2) divided by the initial liquid mass (145,256 lbm). The initial depressurization ratio estimated above, 124 psi/sec, compares favorably to the value 132 psi/sec calculated by the TRANFLO code for the first .57 seconds of the event.

Therefore the maximum dynamic differential pressure that could exist in the steam generator prior to motion of the fluid is:

$$\Delta P = \frac{dP}{dt} * (\Delta t_l + \Delta t_v)$$

where

Δt_l, Δt_v = acoustic transport times for the liquid and vapor regions

dP/dt = rate of pressure decay in the steam region

2.6 Determination of Bulk Fluid Motion

Once the pressure response of the steam space has been determined and a pressure differential across the fluid region defined, the bulk motion of the fluid can be characterized. For the purposes of this calculation, the pressure drop determined above will be applied across a control volume extending from the second highest support plate (N TSP) to the entrance to the separators. Figure 3 provides a diagram of the control volume. Using the one-dimensional Bernoulli integral approach (Reference 2), the following equation can be written:

$$\left(\frac{L}{A}\right)_T \frac{dM}{dt} + \Delta P + \rho g(z_2 - z_1) + \frac{M^2}{2\rho}\left(\frac{1}{A_2^2} - \frac{1}{A_1^2} + \sum \frac{K}{A^2}\right) = 0$$

where

$(L/A)_T$ = Total path inertia (length/area)

M= Mass flow rate

ΔP= differential pressure

z_1, z_2 = elevations at beginning and end of control volume

ρ = fluid density

A_1, A_2 = entrance and exit areas

$\Sigma(K/A^2)$ = friction factor/area representing viscous pressure loss terms at obstructions

This equation can then be directly integrated to achieve a solution of the mass flow rate of fluid vs. time. The solution has the form:

$$M(t) = \frac{1}{C}\left[\frac{e^{2C\Delta P\left(\frac{A}{L}\right)_T} - 1}{e^{2C\Delta P\left(\frac{A}{L}\right)_T} + 1}\right]$$

where

$$C^2 = \frac{1}{2\rho\Delta P}\left[\frac{1}{A_2^2} - \frac{1}{A_1^2} + \sum \frac{K}{A^2}\right]$$

This equation can then be solved for the bulk fluid motion. The pressure drop at the upper TSP can then be readily determined. It should be noted that this formulation ignores the effects of wall friction for conservatism.

Figure 1 Diagram of D4 Steam Generator

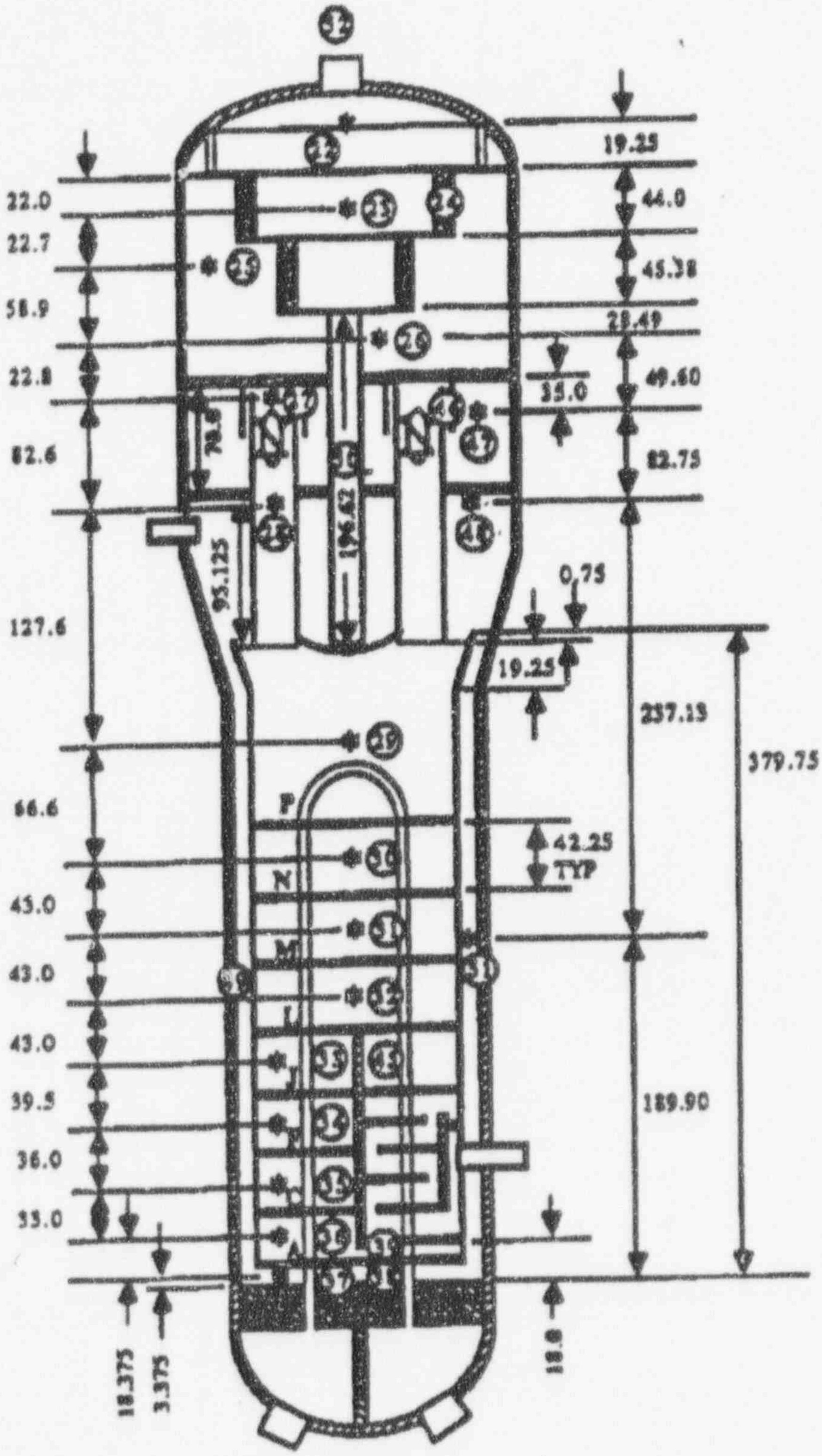


Figure 2 Time Sequence for MSLB

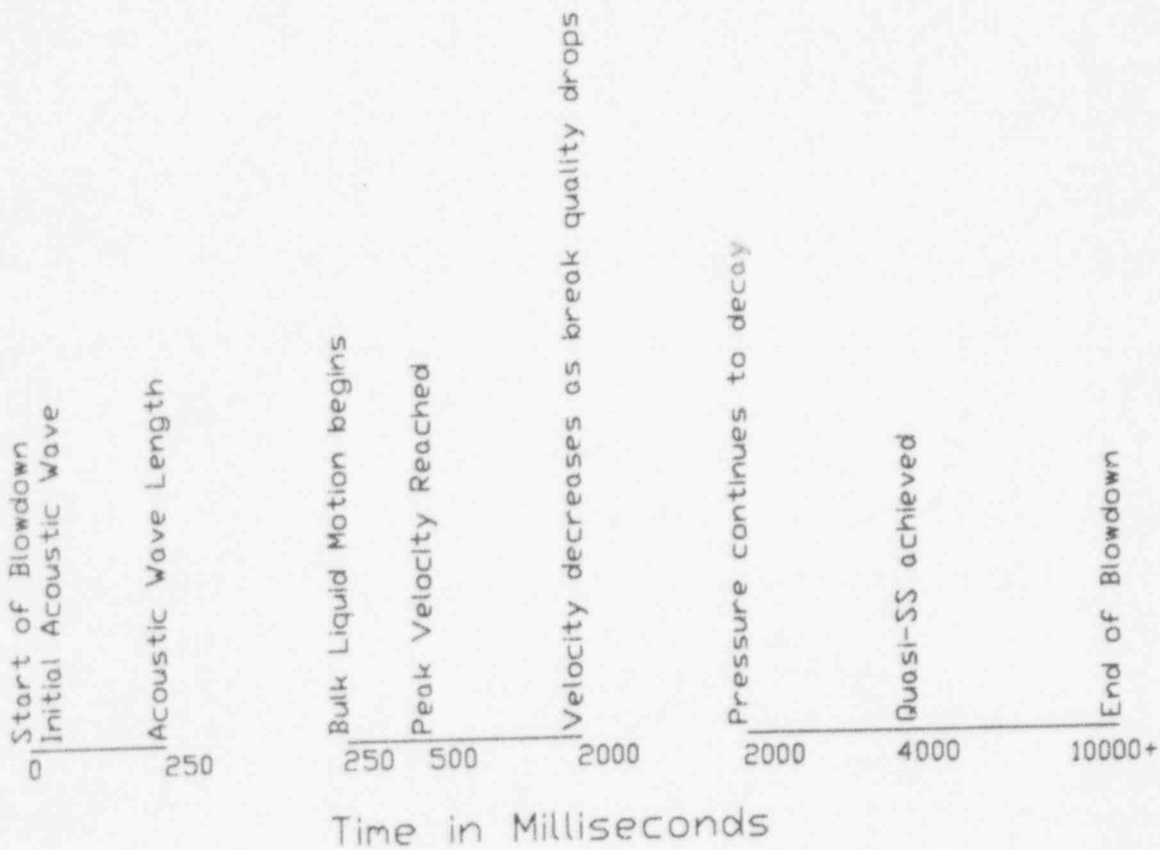
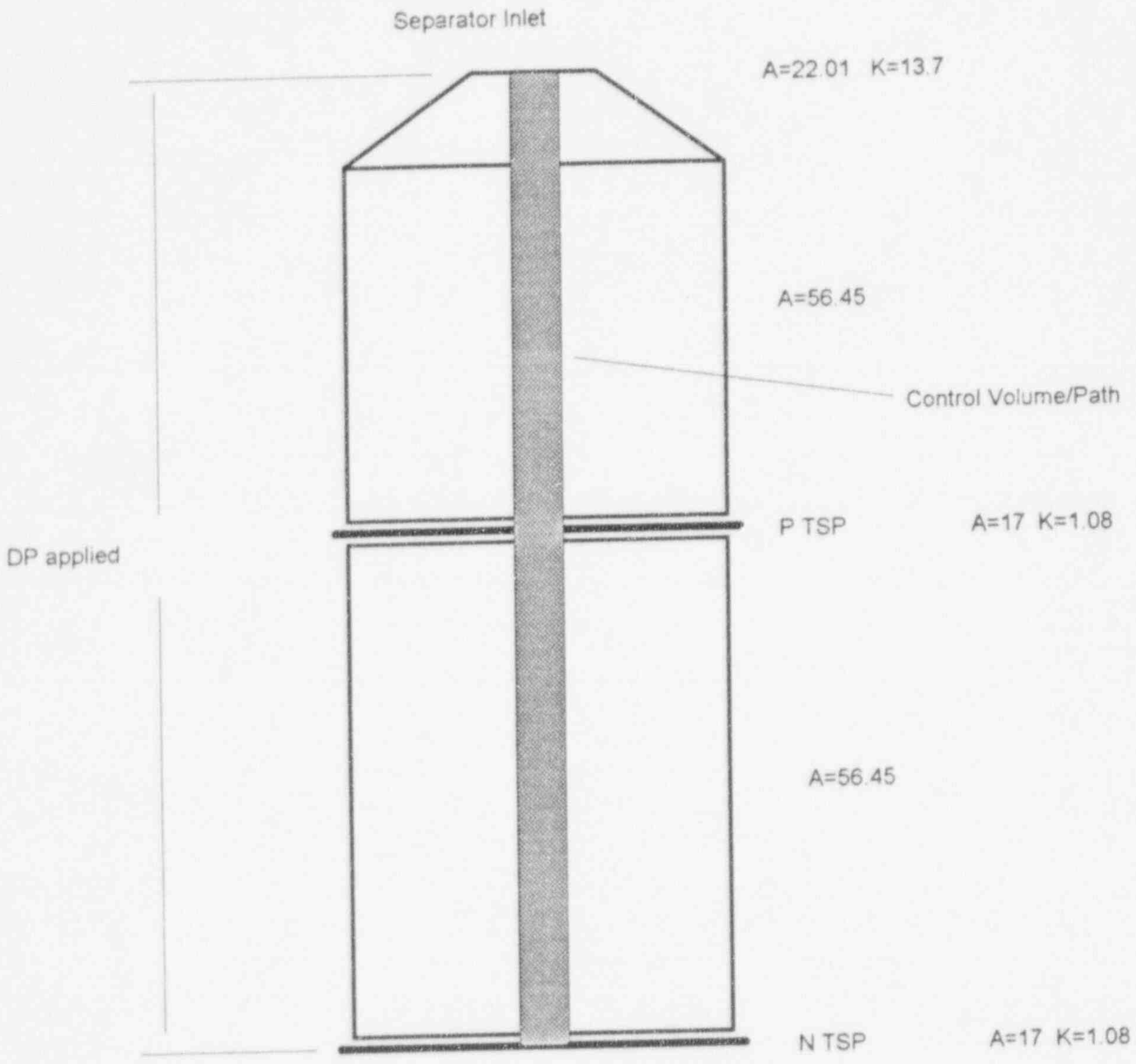


Figure 3 Control Volume Diagram



3. Calculations

3.1 Steam Region Depressurization Rate

The steam region depressurization rate of 124 psi/sec was determined using the method presented in Section 2.5. By way of comparison, the TRANFLO code produces a depressurization rate of approximately 132 psi/sec during the first 500 milliseconds of the event.

3.2 Determination of Applied Pressure Gradient

Given the differential pressure rate calculated above, the maximum pressure that could be applied across the fluid region can then be determined. Using a value of 130 psi/sec, the pressure rate occurring just after the initial acoustic effects, the differential pressure acting on the fluid becomes:

$$DP = 124 psi/\sec x(40.583 ft/157.5 ft/\sec + 27.75 ft/1476.4 ft/\sec)$$

$$DP = 34.28 psi$$

3.3 Bulk Fluid Motion Calculations

3.3.1 Single Phase Case -Small Control Volume

Using the formulation discussed in section 2.6, the maximum velocity of the fluid at the tube support plate and then the pressure loss (load) on the support plate can be calculated. The velocity at the P TSP is shown in Figure 4. The pressure drop that would result from this velocity of single phase fluid is shown in Figure 5. The pressure drop is calculated using the relationship:

$$\Delta P = \frac{K\rho V^2}{2x144xg}$$

where

K=local loss coefficient

ρ=density lbm/sec

V= velocity ft/sec

3.3.2 Single Phase Case - Extended Control Volume

This case was performed to provide a more realistic estimate of the maximum velocity of the fluid. This case extends the control volume to the bottom of the steam generator and accounts for the additional losses in the lower tube support plates. The areas were assumed to be continuous to the bottom, and the same loss coefficient was utilized for all support plates. This is conservative given that higher loss coefficients and slightly reduced areas exist in the preheater and boiler sections in the lower portions of the generator. The velocity at the P TSP is shown in Figure 6. The pressure drop that would result is shown in Figure 7.

3.3.3 Two Phase Case - Extended Control Volume

This case was performed to provide an indication of the effects of two phase fluid flow in the tube regions. Since the initial decompression wave will cause void formation, some increase in fluid friction can be expected. The extended control volume model was modified to include a HEM multiplier on the local loss factors used. This approach is consistent with a "liquid only" based calculation per Reference 2, page 487. A two phase friction multiplier was selected assuming 1% mass quality, which bounds the amount of voids calculated by TRANFLO in the initial phase of the event. The velocity at the P TSP is shown in Figure 8. The pressure drop that would result is shown in Figure 9.

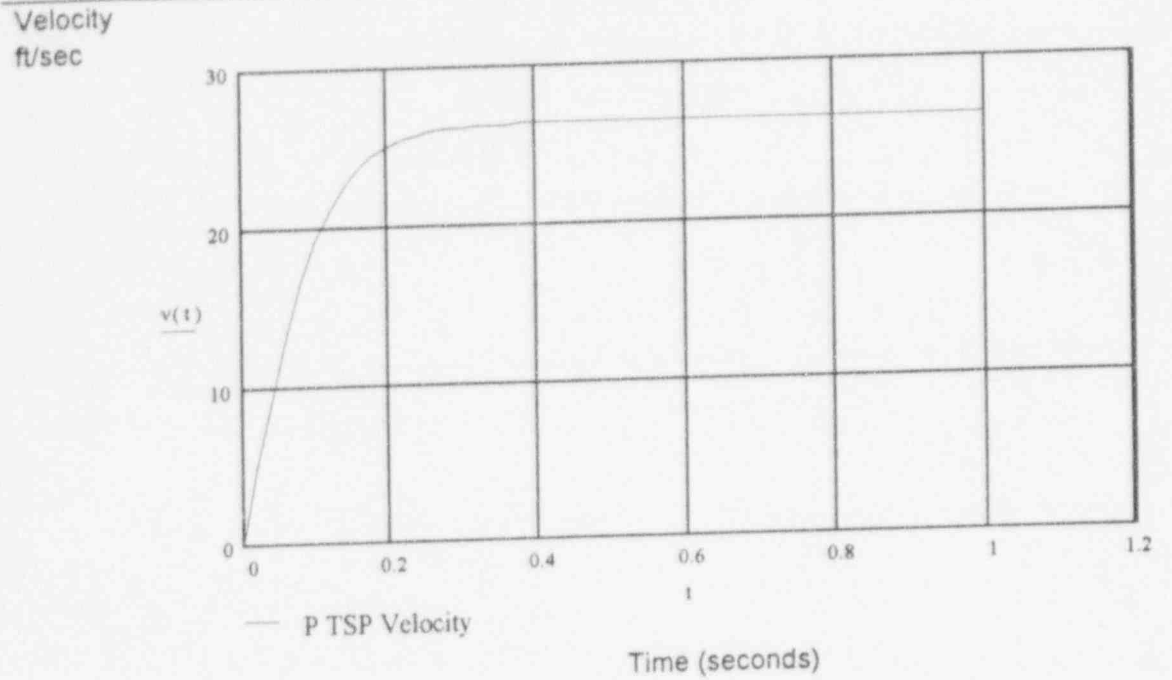


Figure 4 Velocity at P-TSP Single Phase Case

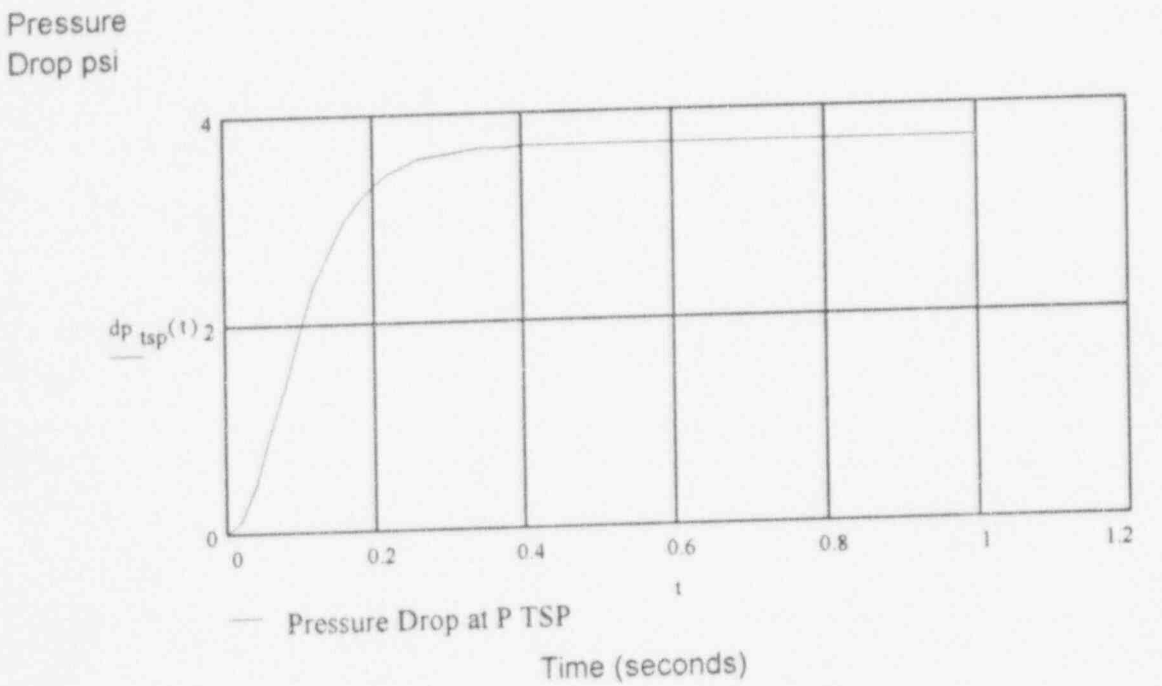
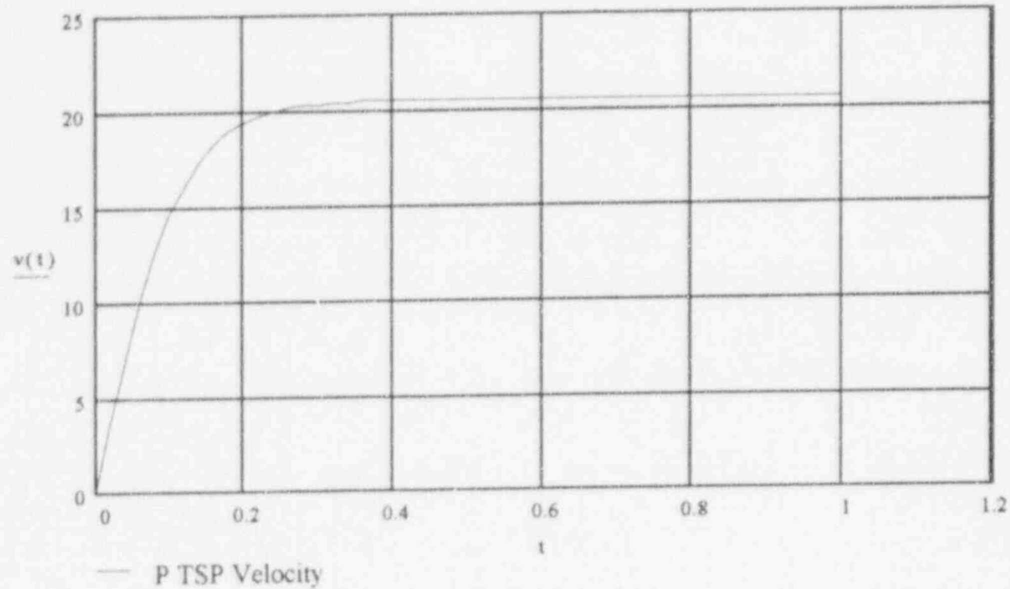


Figure 5 Pressure Drop at P TSP Single Phase Case

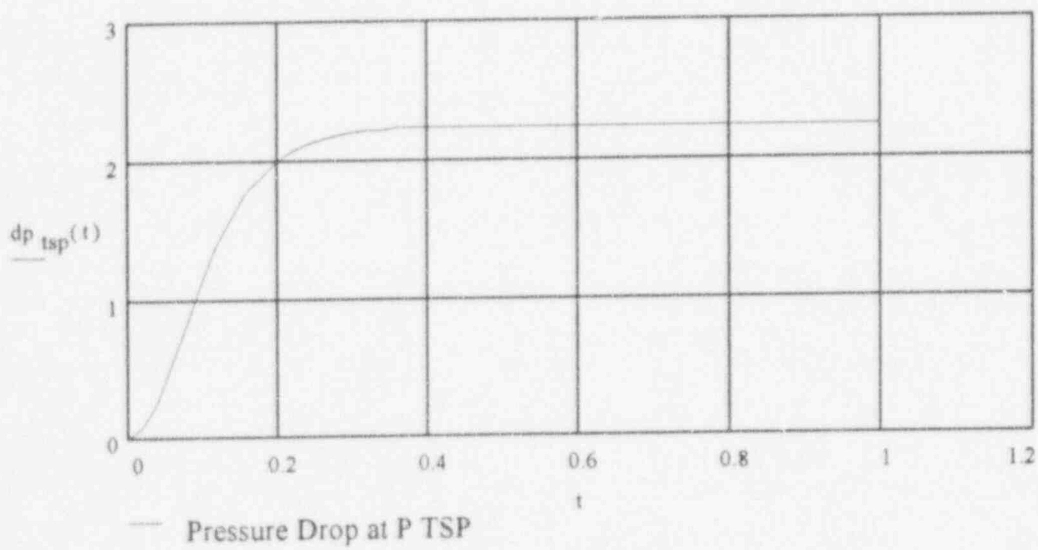
Velocity
ft/sec



Time (seconds)

Figure 6 Fluid velocity at P TSP - Extended CV case

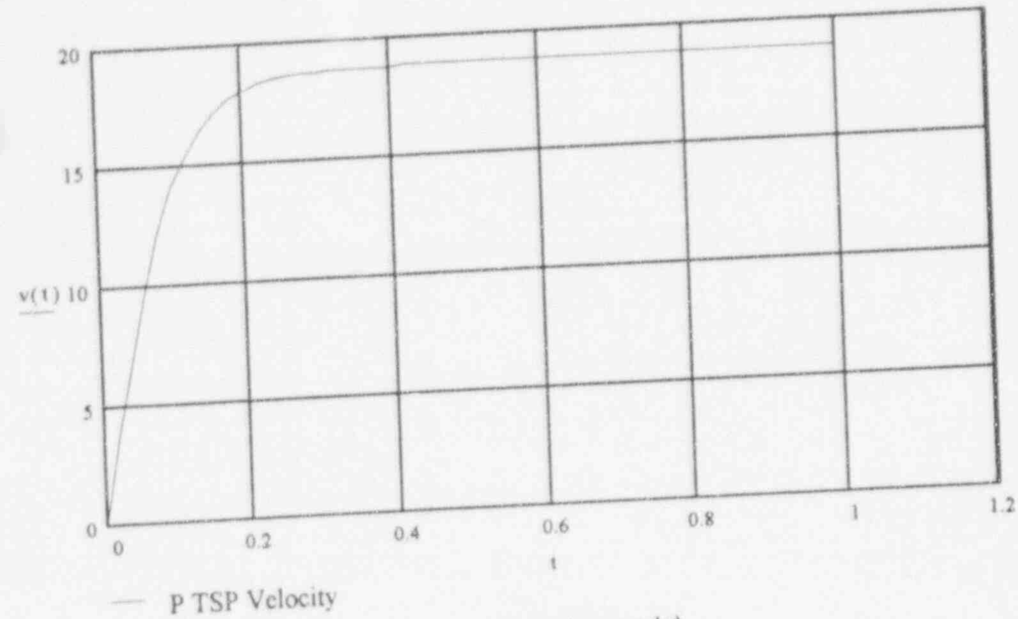
Pressure
Drop psi



Time (seconds)

Figure 7 Pressure Drop at P-TSP Extended CV Case

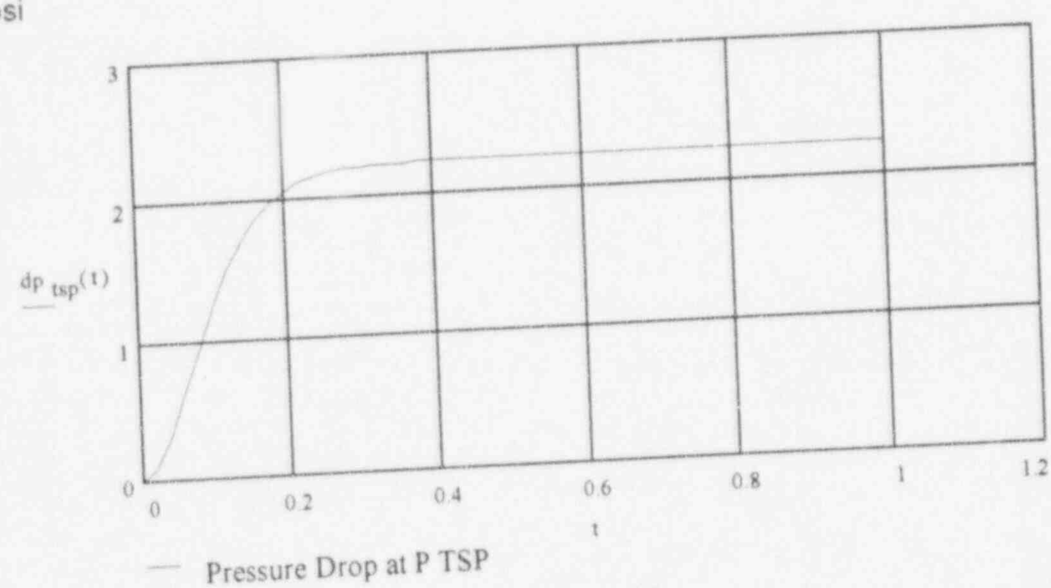
Velocity
ft/sec



Time (seconds)

Figure 8 Velocity at P TSP -Extended CV two phase case

Pressure
Drop psi



Time (seconds)

Figure 9 Pressure Drop at P TSP - Extended CV two phase case

4. Results

The results obtained from these calculations are presented in Table 2. The base case HZP/NWL TRANFLO results are provided for comparison. As can be seen, the limiting CV case produces very conservative results. This is expected since the entire pressure drop occurring in the steam generator is being applied to a small section of the upper tube bundle. This case is believed to be limiting, and demonstrates the conservatism inherent in the factor of two applied to the base TRANFLO results used to generate structural loads. The extended CV cases provide a more physically realistic treatment of the total pressure drops in the generator, and support the results obtained with TRANFLO. The two phase case provides an estimate of the effects that would be seen if HEM multipliers are applied to the pressure drop determination. The increased pressure drop of the two phase flow is nearly compensated by a decrease in predicted velocity, with the net result being a minor variation in pressure drop.

Case	Depressurization rate psi/sec	Peak Velocity at P-TSP ft/sec	Max. Pressure drop at P-TSP psi
Base- small CV	124	26.37	3.68
Extended CV 1Φ	124	20.56	2.24
Extended CV 2Φ	124	18.81	2.23
TRANFLO	132	≈17	1.6 (3.2 used in structural evaluation)

Table 2 Summary of Results

5. Conclusions/Discussion

A methodology to determine the peak loads on the upper tube support plate that would result from a design basis MSLB event has been developed and exercised. This methodology is based solely on first principles and has no reliance on computer codes. The results obtained compare favorably with those obtained via computer simulation, and provide a basis to assess the margin of safety utilized in the analyses of TSP loads. It can be concluded that the factor of two used in the structural assessment results in a physically bounding pressure drop, even allowing for typical uncertainties in two phase pressure drop prediction.

6. References

1) "Introduction to Unsteady Thermofluid Mechanics", F. J. Moody, 1990.

2) "Nuclear Systems I", N. E. Todreas and M. S. Kazimi, 1990.

3) "The Thermal Hydraulics of a Boiling Water Nuclear Reactor", R. T. Lahey Jr. and F. J. Moody, 1977.

Appendix A - Mathcad Cases

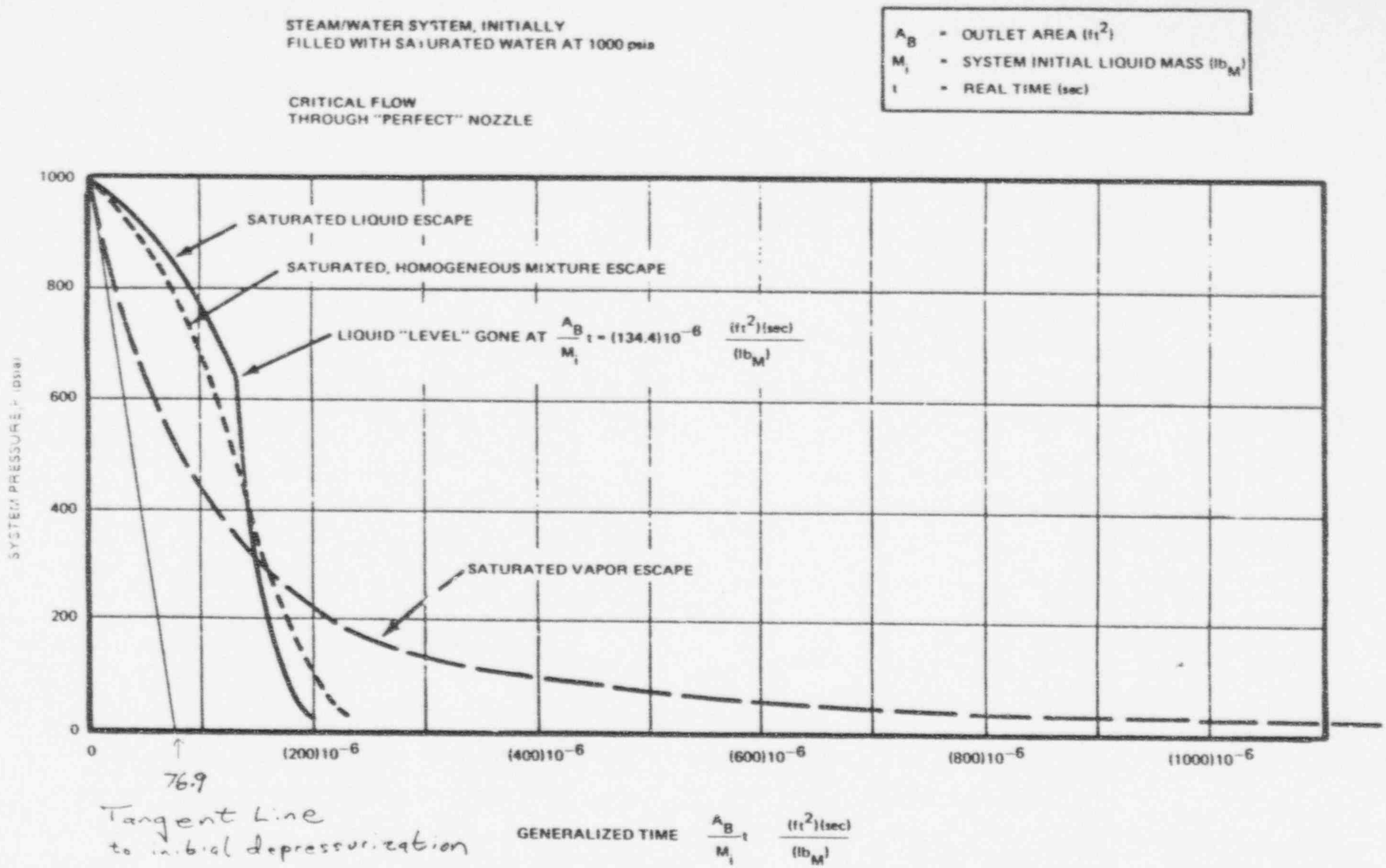


Fig. 9-17a. Pressure transients—blowdowns from 1000-psia reference system.

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A Simplified Approach to Assessing TSP Loads

Introduction

A simple physical model to describe the fluid behavior at the upper TSP can be developed based on the Bernoulli integral equation, as described in Kazimi's "Nuclear Systems I" text. In the initial part of the transient, the fluid in the tube area adjacent to the upper support plate is single phase liquid. Following the break, this liquid is subjected to decompression and acceleration forces. Blowdown calculations have been performed to estimate the driving pressure. By drawing a control volume around the upper support plate, one can solve the Bernoulli integral equation for the flow rate of the fluid vs time, accounting for inertial and viscous effects. This is a reasonable approximation to the initial behavior of the fluid, since only minor void generation occurs initially.

Geometrical Input

$A_i := 17 \cdot ft^2$ Flow Area of N TSP, entrance to control volume

$A_o := 22.01 \cdot ft^2$ Flow area of Separator inlet, exit of control volume

$A_{tube} := 56.45 \cdot ft^2$ Flow area of tube region

$dp_1 := 34.28 \cdot 32.2 \cdot 144 \cdot \frac{lb}{ft \cdot sec^2}$ Differential Pressure (dynamic component)

$A_{tsp} := 17 \cdot ft^2$ Area of TSP

$K_{tsp} := 1.08 \cdot 2$ Loss Coefficient of TSPs (P and N)

$A_{sep} := 22.01 \cdot ft^2$

$K_{sep} := 13.9$

The inertia of the path can be determined by the path lengths divided by the respective areas

$$I := \frac{8.1666 \cdot ft}{A_{tube}} + \frac{3.5733 \cdot ft}{A_{tube}} + \frac{14.1567 \cdot ft}{A_{sep}}$$

$\rho := 45.5 \cdot \frac{lb}{ft^3}$ Fluid Density

Neglect Gravity Effects, since applied load is only dynamic component

$$P_{grav} := \rho \cdot 32.2 \cdot lb \cdot \frac{ft}{sec^2 \cdot lb} \cdot ((8.1666 + 3.5733) \cdot ft)$$

$dp := dp_1$

General Solution

Kazimi derives a solution with a constant C^2 of the form indicated below:

$$C_0 := \frac{1}{2\cdot\rho\cdot dp}\cdot\left[\left(\frac{1}{A_o^{\,2}} - \frac{1}{A_i^{\,2}}\right) + \frac{K_{tsp}}{A_{tsp}^{\,2}} + \frac{K_{sep}}{A_{sep}^{\,2}}\right]$$

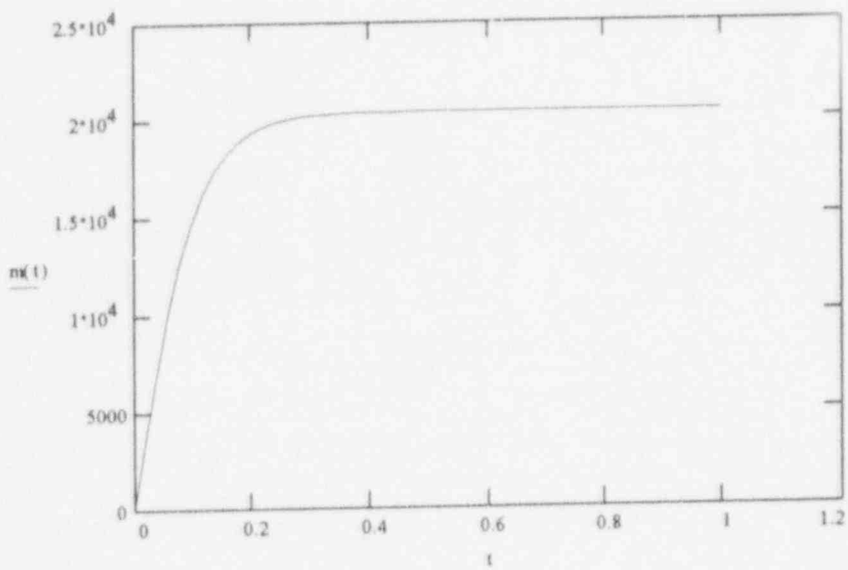
$$C := \sqrt{C_0}$$

$$t := 0\cdot sec, .02\cdot sec .. 1\cdot sec$$

The time dependent solution is of the form

$$m(t) := \frac{1}{C}\cdot\left[\frac{e^{\left(\frac{2\cdot C\cdot dp}{I}\cdot t\right)} - 1}{e^{\frac{2\cdot C\cdot dp}{I}\cdot t} + 1}\right]$$

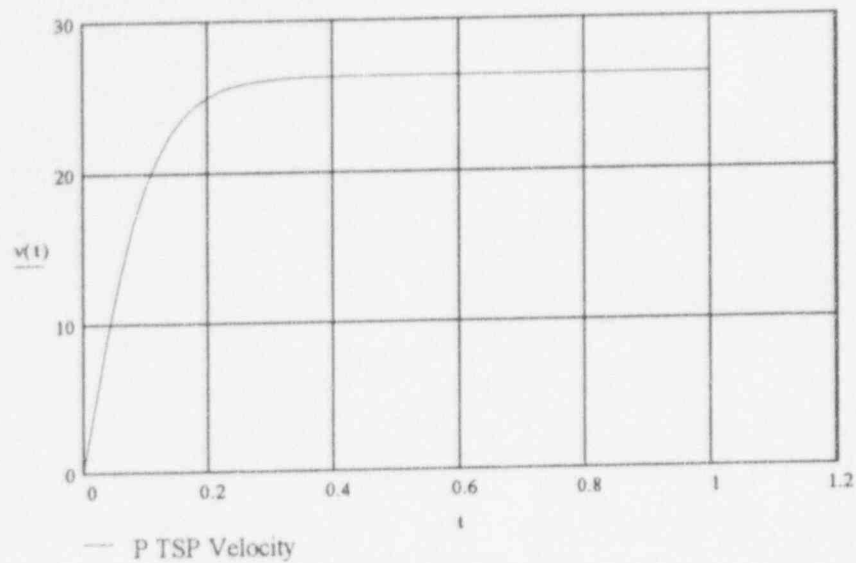
The results are shown graphically below



The velocity at the tube support plate is shown below

$$v(t) := \frac{m(t)}{\rho \cdot A_{tsp}}$$

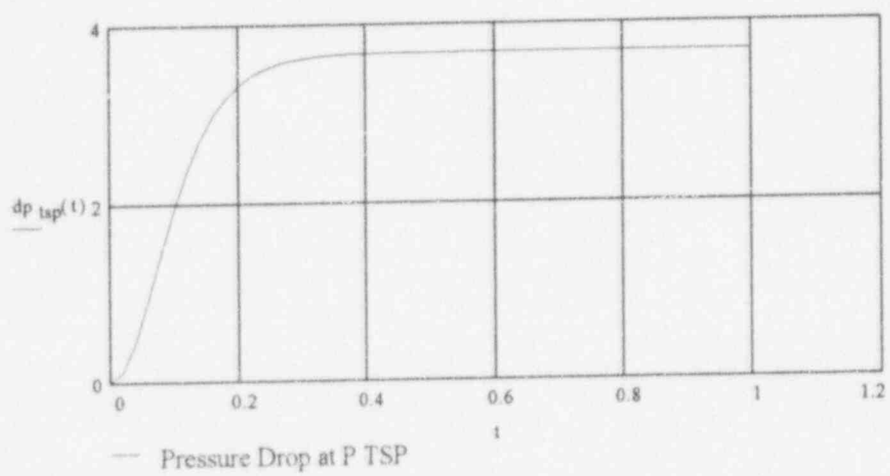
Velocity
ft/sec



Time (seconds)

$$dp_{tsp}(t) := \frac{1.08 \cdot \rho \cdot v(t)^2}{2 \cdot 144 \cdot 32.2}$$

Pressure
Drop psi



Time (seconds)

A Simplified Approach to Assessing TSP Loads-Full Tube Bundle Case

Introduction

A simple physical model to describe the fluid behavior at the upper TSP can be developed based on the Bernoulli integral equation, as described in Kazimi's "Nuclear Systems I" text. In the initial part of the transient, the fluid in the tube area adjacent to the upper support plate is single phase liquid. Following the break, this liquid is subjected to decompression and acceleration forces. The depressurization rate of the steam region can be estimated with textbook blowdown methods and a driving pressure across the fluid region can be inferred. By drawing a control volume around the fluid regions, one can solve the Bernoulli integral equation for the flow rate of the fluid vs time, accounting for inertial and viscous effects.

In this case the same basic approach is followed, but with the control volume extended to the bottom of the tube region.

Geometrical Input

$A_i := 17 \cdot ft^2$ Flow Area of N TSP, entrance to control volume

$A_o := 22.01 \cdot ft^2$ Flow area of Separator inlet, exit of control volume

$A_{tube} := 56.45 \cdot ft^2$ Flow area of tube region

$dp_1 := 34.28 \cdot 32.2 \cdot 144 \cdot \frac{lb}{ft \cdot sec^2}$ Differential Pressure

$A_{tsp} := 17 \cdot ft^2$ Area of TSP

The actual areas are smaller and the losses larger in the lower regions. For simplicity, it will be conservatively assumed that the lower tube region can be modeled identically to the upper regions. This will underpredict the losses and inertias in the lower region.

$K_{tsp} := 1.08 \cdot 8$ Loss Coefficient of all TSPs (P to A)

$A_{sep} := 22.01 \cdot ft^2$ $K_{sep} := 13.9$

The inertia of the path can be determined by the path lengths divided by the respective areas

$$I := \frac{8.1666 \cdot ft}{A_{tube}} + \frac{3.5733 \cdot ft}{A_{tube}} + \frac{14.1567 \cdot ft}{A_{sep}} + \frac{3.0 \cdot ft}{A_{tube}} \cdot 2 + \frac{2.5 \cdot ft}{A_{tube}} + \frac{3.5733 \cdot ft}{A_{tube}} \cdot 2$$

$\rho := 45.5 \cdot \frac{lb}{ft^3}$ Fluid Density

Gravity Effects are ignored since the elevation head is not added to the dynamic load:

$P_{grav} := 0 \cdot \frac{lb}{ft \cdot sec^2}$

$dp := dp_1 - P_{grav}$

General Solution

Kazimi derives a solution with a constant C^2 of the form indicated below:

$$C_0 := \frac{1}{2\cdot\rho\cdot dp}\cdot\left[\left(\frac{1}{A_o^{\,2}}-\frac{1}{A_i^{\,2}}\right)+\frac{K_{tsp}}{A_{tsp}^{\,2}}+\frac{K_{sep}}{A_{sep}^{\,2}}\right]$$

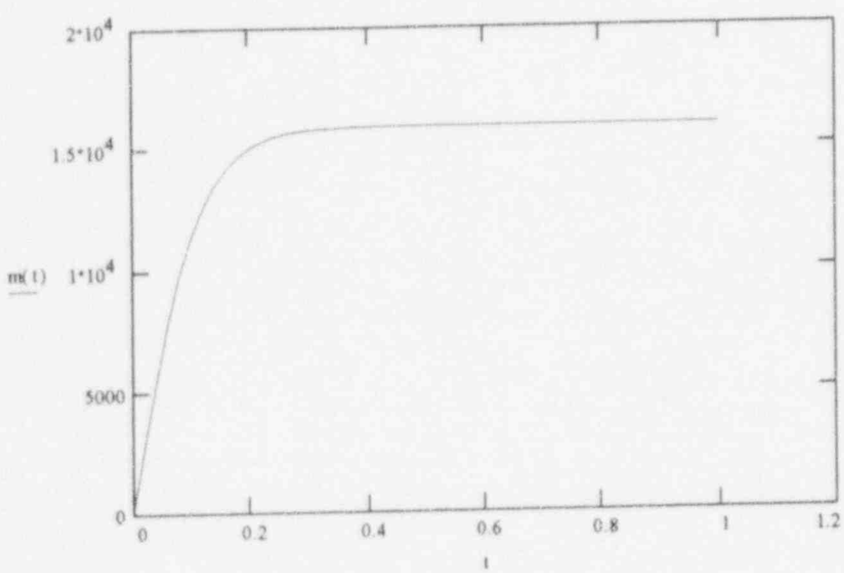
$$C := \sqrt{C_0}$$

$$t := 0\cdot sec, .02\cdot sec .. 1\cdot sec$$

The time dependent solution is of the form

$$m(t) := \frac{1}{C}\cdot\left[\frac{e^{\left(\frac{2\cdot C\cdot dp}{l}\cdot t\right)}-1}{e^{\frac{2\cdot C\cdot dp}{l}\cdot t}+1}\right]$$

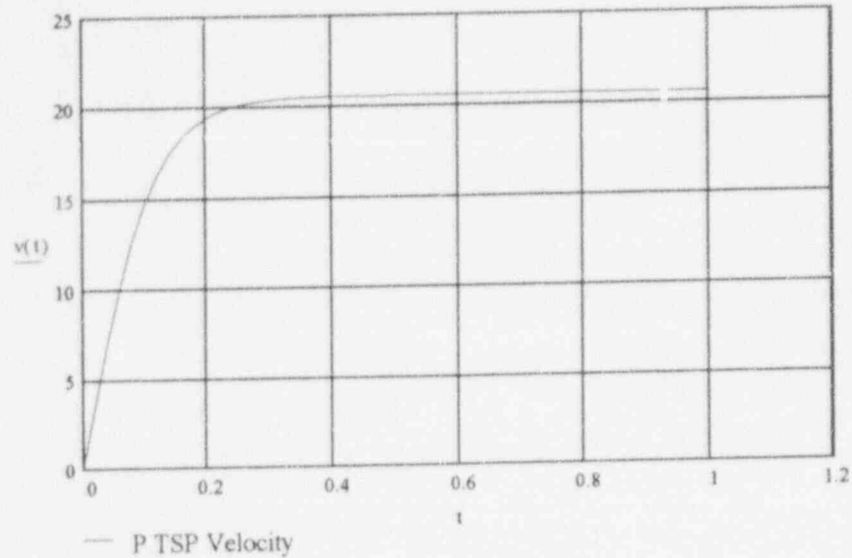
The results are shown graphically below



The velocity at the tube support plate is shown below

$$v(t) := \frac{m(t)}{\rho \cdot A_{tsp}}$$

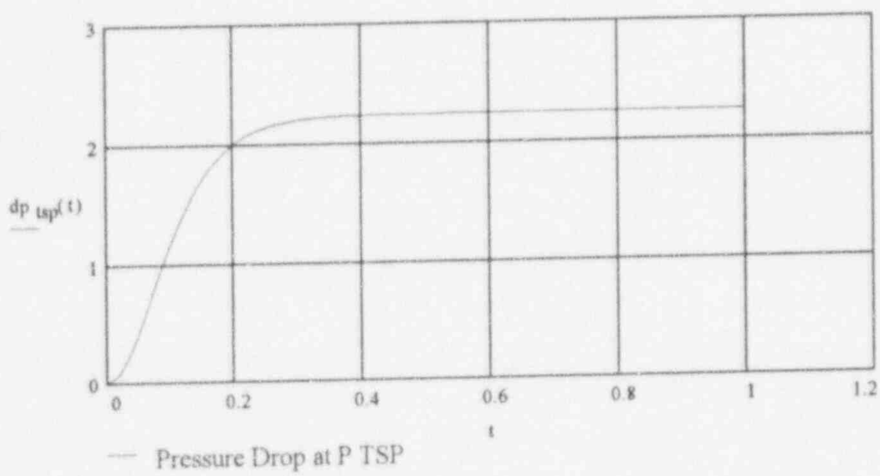
Velocity
ft/sec



Time (seconds)

$$dp_{tsp}(t) := \frac{1.08 \cdot \rho \cdot v(t)^2}{2 \cdot 144 \cdot 32.2}$$

Pressure
Drop psi



Time (seconds)

A Simplified Approach to Assessing TSP Loads-Extended CV/2 phase

Introduction

A simple physical model to describe the fluid behavior at the upper TSP can be developed based on the Bernoulli integral equation, as described in Kazimi's "Nuclear Systems I" text. In the initial part of the transient, the fluid in the tube area adjacent to the upper support plate is single phase liquid. Following the break, this liquid is subjected to decompression and acceleration forces. The depressurization rate of the steam region can be estimated by textbook blowdown methods and a driving pressure across the fluid region can be inferred. By drawing a control volume around the fluid regions, one can solve the Bernoulli integral equation for the flow rate of the fluid vs time, accounting for inertial and viscous effects.

In this case the same basic approach is followed, but with the control volume extended to the bottom of the tube region.

Geometrical Input

$A_i := 17 \cdot ft^2$ Flow Area of N TSP, entrance to control volume

$A_o := 22.01 \cdot ft^2$ Flow area of Separator inlet, exit of control volume

$A_{tube} := 56.45 \cdot ft^2$ Flow area of tube region

$dp_1 := 34.28 \cdot 32.2 \cdot 144 \cdot \frac{lb}{ft \cdot sec^2}$ Differential Pressure

$A_{tsp} := 17 \cdot ft^2$ Area of TSP

$\phi_{sq} := 1.19$ See attached table for HEM multiplier

The actual areas are smaller and the losses larger in the lower regions. For simplicity, it will be conservatively assumed that the lower tube region can be modeled identically to the upper regions. This will underpredict the losses and inertias in the lower region.

$K_{tsp} := 1.08 \cdot 8 \cdot \phi_{sq}$ Loss Coefficient of all TSPs (P to A)

$A_{sep} := 22.01 \cdot ft^2$ $K_{sep} := 13.9 \cdot \phi_{sq}$

The inertia of the path can be determined by the path lengths divided by the respective areas

$$I := \frac{8.1666 \cdot ft}{A_{tube}} + \frac{3.5733 \cdot ft}{A_{tube}} + \frac{14.1567 \cdot ft}{A_{sep}} + \frac{3.0 \cdot ft}{A_{tube}} \cdot 2 + \frac{2.5 \cdot ft}{A_{tube}} + \frac{3.5733 \cdot ft}{A_{tube}} \cdot 2$$

$\rho := 45.5 \cdot \frac{lb}{ft^3}$ Fluid Density

Gravity Effects will be ignored since only the dynamic load is applied

$P_{grav} := 0 \cdot \frac{lb}{ft \cdot sec^2}$

$dp := dp_1 - P_{grav}$

General Solution

Kazimi derives a solution with a constant C^2 of the form indicated below:

$$C_0 := \frac{1}{2 \cdot \rho \cdot dp} \cdot \left[\left(\frac{1}{A_o^2} - \frac{1}{A_i^2} \right) + \frac{K_{tsp}}{A_{tsp}^2} + \frac{K_{sep}}{A_{sep}^2} \right]$$

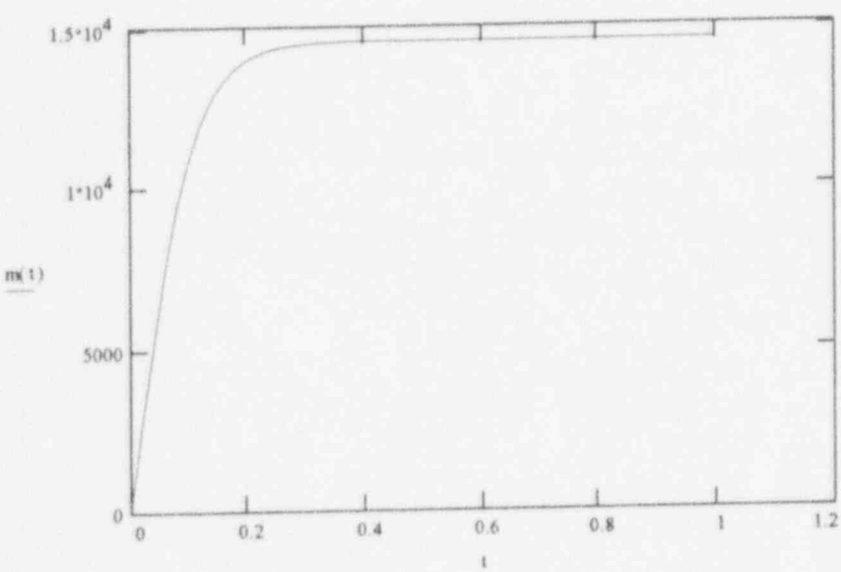
$$C := \sqrt{C_0}$$

$$t := 0 \cdot sec, .02 \cdot sec .. 1 \cdot sec$$

The time dependent solution is of the form

$$m(t) := \frac{1}{C} \cdot \left[\frac{e^{\left(\frac{2 \cdot C \cdot dp}{l} \cdot t\right)} - 1}{e^{\frac{2 \cdot C \cdot dp}{l} \cdot t} + 1} \right]$$

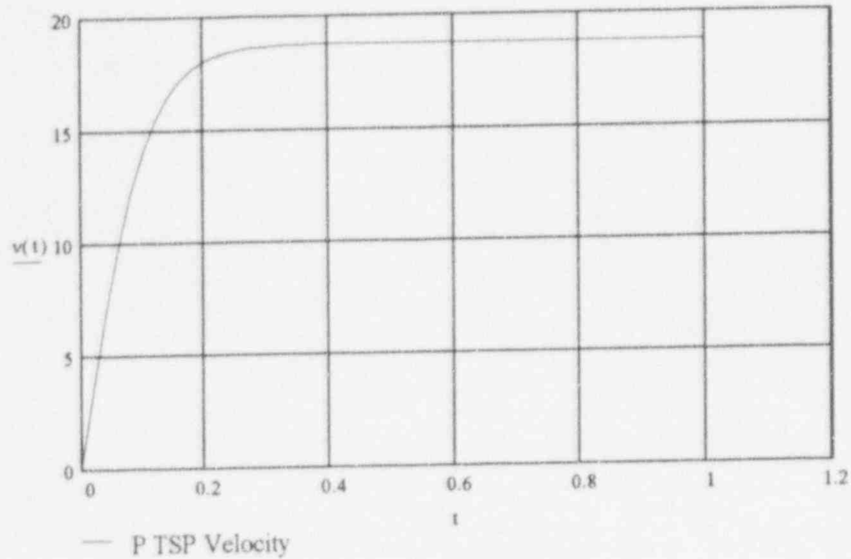
The results are shown graphically below



The velocity at the tube support plate is shown below

$$v(t) := \frac{m(t)}{\rho \cdot A_{tsp}}$$

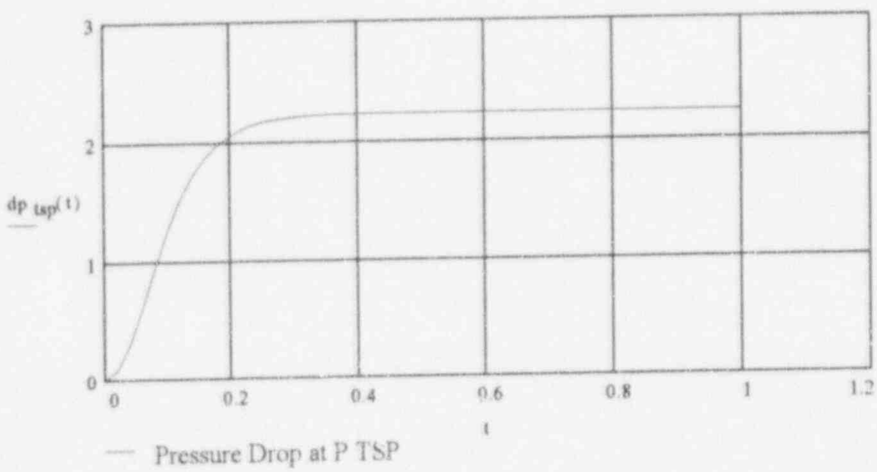
Velocity
ft/sec



Time (seconds)

$$dp_{tsp}(t) := \frac{1.08 \cdot \rho \cdot v(t)^2 \cdot \phi_{sq}}{2 \cdot 144 \cdot 32.2}$$

Pressure
Drop psi



Time (seconds)

Table 2.1 Values of the two-phase frictional multiplier ϕ_{lo}^2 for the homogeneous model steam–water system

$$\phi_{lo}^2 = \left[1 + x\left(\frac{v_{fg}}{v_f}\right)\right]\left[1 + x\left(\frac{\mu_{fg}}{\mu_g}\right)\right]^{-1/4}$$

Steam quality % by wt.	Pressure, bar (psia)								
	1·01	6·89	34·4	68·9	103	138	172	207	221·2
	(14·7)	(100)	(500)	(1000)	(1500)	(2000)	(2500)	(3000)	(3206)
1	16·21	3·40	1·44	1·19	1·10	1·05	1·04	1·01	1·0
5	67·6	12·18	3·12	1·89	1·49	1·28	1·16	1·06	1·0
10	121·2	21·8	5·06	2·73	1·95	1·56	1·30	1·13	1·0
20	212·2	38·7	7·8	4·27	2·81	2·08	1·60	1·25	1·0
30	292·8	53·5	11·74	5·71	3·60	2·57	1·87	1·36	1·0
40	366	67·3	14·7	7·03	4·36	3·04	2·14	1·48	1·0
50	435	80·2	17·45	8·30	5·08	3·48	2·41	1·60	1·0
60	500	92·4	20·14	9·50	5·76	3·91	2·67	1·71	1·0
70	563	104·2	22·7	10·70	6·44	4·33	2·89	1·82	1·0
80	623	115·7	25·1	11·81	7·08	4·74	3·14	1·93	1·0
90	682	127	27·5	12·90	7·75	5·21	3·37	2·04	1·0
100	738	137·4	29·8	13·98	8·32	5·52	3·60	2·14	1·0

Table 2.2 Values of the two-phase frictional multiplier ϕ_{lo}^2 for the Martinelli–Nelson model steam–water system

Steam quality % by wt.	Pressure, bar (psia)								
	1·01	6·89	34·4	68·9	103	138	172	207	221·2
	(14·7)	(100)	(500)	(1000)	(1500)	(2000)	(2500)	(3000)	(3206)
1	5·6	3·5	1·8	1·6	1·35	1·2	1·1	1·05	1·00
5	30	15	5·3	3·6	2·6	1·75	1·43	1·17	1·00
10	69	28	8·9	5·4	3·6	2·45	1·75	1·30	1·00
20	150	56	16·2	8·6	5·1	3·25	2·19	1·51	1·00
30	245	83	23·0	11·6	6·8	4·04	2·62	1·68	1·00
40	350	115	29·2	14·4	8·4	4·82	3·02	1·83	1·00
50	450	145	34·9	17·0	9·9	5·59	3·38	1·97	1·00
60	545	174	40·0	19·4	11·1	6·34	3·70	2·10	1·00
70	625	199	44·6	21·4	12·1	7·05	3·96	2·23	1·00
80	685	216	48·6	22·9	12·8	7·70	4·15	2·35	1·00
90	720	210	48·0	22·3	13·0	7·95	4·20	2·38	1·00
100	525	130	30·0	15·0	8·6	5·90	3·70	2·15	1·00

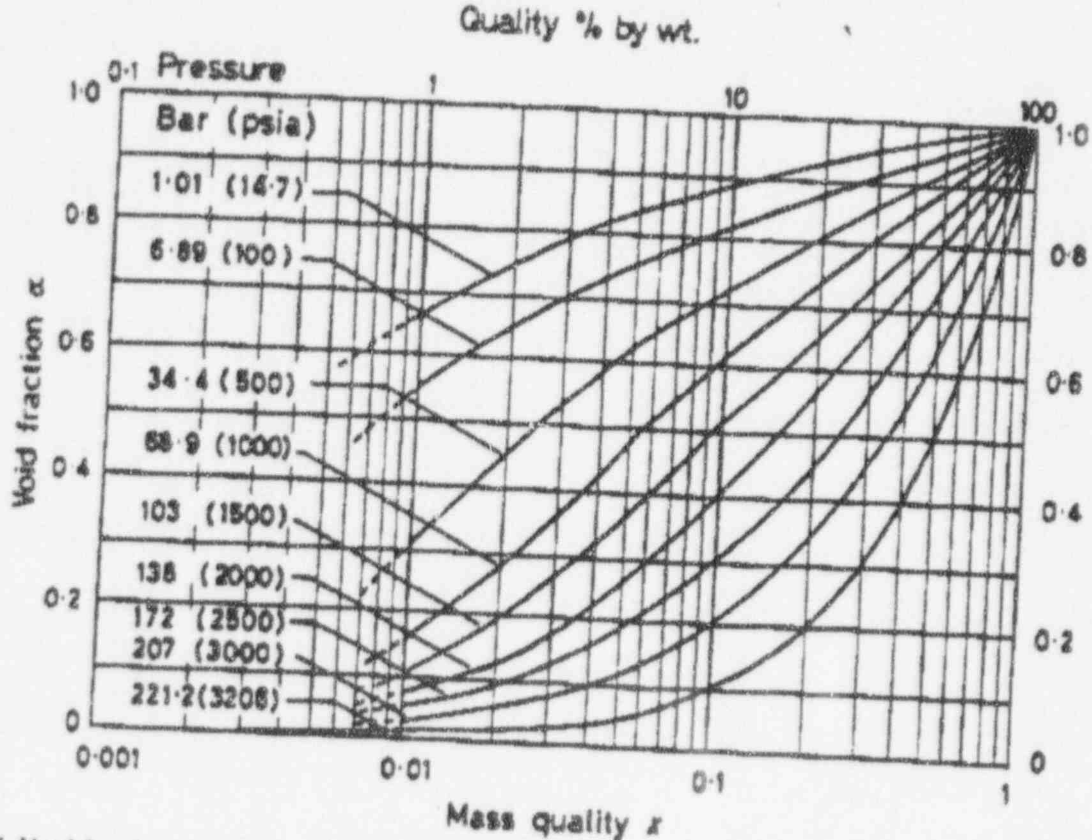


Fig. 2.6. Void fraction α as a function of quality and absolute pressure steam–water (Martinelli–Nelson[12])