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SUBJECT: Forwards more detailed response re main steam check valve performance questions arising from NRC Insp 99-05 following completion of assessment being performed by Duke Engineering & Services.

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September 24, 1999

U.S. Nuclear Regulatory Commission
Document Control Desk
Attn: Guy S. Vissing
Project Directorate I
Washington, D.C. 20555

Subject: Response to Questions Related to Main Steam Check Valve Performance per NRC Inspection 99-05

Reference: August 23, 1999 letter from Robert C. Mecredy to USNRC, "NRC #40500 Team Inspection 50-244/99-05, dated 8/6/99"

Dear Mr. Vissing:

On August 23, 1999, RG&E provided a summary of the discussions between RG&E and NRC personnel regarding main steam check valve performance questions arising from NRC Inspection 99-05 (see Reference). At that time we stated we would provide more detailed responses following the completion of an independent assessment being performed by Duke Engineering and Services. The attached responses include the results of that assessment.

We have concluded that the main steam check valves are operable, in that they would perform their safety function for the limiting steam line break, with substantial margin. We have also decided to initiate engineering activities to optimize packing of the valves so as to provide the minimum amount of friction needed for a leak-tight packing configuration. Recommendations from these engineering activities would be implemented during the year 2000 refueling outage.

Very truly yours,

Robert C. Mecredy

Attachment

IE01

xc: Mr. Guy S. Vissing (Mail Stop 8C2)
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RESPONSE TO NRC QUESTIONS

QUESTION 1:

What analysis method will be used? It is felt that a 3-dimensional computational fluid dynamics code is needed to properly model the complex flow conditions. How will the analysis performed be validated for this type of application?

RESPONSE:

Based upon the introductory discussion to the eight NRC Questions, it appears that the major concern related to the RG&E Main Steam (MS) check valve analysis performed in Design Analysis DA-ME-92-147, Rev.2 (Reference 1) revolves around the fact that a fundamental change was made to the check valve without a comprehensive safety evaluation. This change resulted in a small difference between the calculated available torque due to reverse flow and required torque to initiate valve closure. Since the difference in available and maximum measured As-Found torque reported in Reference 1 was only approximately 1% (912 ft-lb vs 900 ft-lb), a concern exists with the uncertainty associated with the simplified methodology used in Reference 1 to quantify a complicated flow condition. Specifically, it has been stated that the licensee must demonstrate that the uncertainty in the Reference 1 calculation must be less than the available margin of torque needed to close the valve.

RG&E concurs with the NRC assessment that the flow pattern around the MS check valve disk under reverse flow conditions represents a complicated flow geometry; however, RG&E believes that sufficient conservatisms exist in the Reference 1 approach to bound these uncertainties. Specifically, the following areas of conservatism exist with the RG&E methodology that causes its calculated torque value to be significantly below the true torque that would be generated during a reverse flow condition for the limiting MS break size:

- Conservative reverse mass flow rate
- Conservative static pressure under the valve disk
- Conservative break size selection

The conservatism resulting from these three areas are discussed below.

<u>Mass Flow Rate</u>

The major source of conservatism in the Reference 1 analysis involves the reverse mass flow rate that was assumed. The mass flow rate used in Reference 1 to assess closing torque was

obtained after reviewing the LOFTRAN analysis performed in Reference 2. In lieu of using transient reverse flow and MS pressure data from the LOFTRAN analysis, the check valve analysis used a single mass flow rate and MS pressure that bounded the LOFTRAN transient data. If the transient reverse flow data had been used to generate a time dependent torque curve, the Reference 1 methodology would have calculated significantly higher transient torque values than what was calculated in Reference 1.

To demonstrate this condition, a revised LOFTRAN analysis was performed in Reference 3 that more closely modeled the reverse flow transient for mass flow from the intact SG to the break location. The Reference 3 analysis also used a more conservative assessment of MS piping hydraulic resistance to minimize reverse flow from the intact SG. The resulting transient SG pressures for the limiting 0.86 ft^2 steam line break is shown in Figure 1. The resulting transient mass flow rate contribution to the total break flow from each SG is shown on Figure 2.

The Figure 2 results indicate that no reverse flow exists through the MS check valve until after the turbine stop valves have closed. Prior to the closure of the turbine stop valves, all of the break flow is supplied from the faulted SG. After the turbine stop valves have closed, the break area represents the only flow path available for both the faulted and intact SGs. Consequently, reverse flow is initiated from the intact SG to the break location immediately following the isolation of flow to the turbine. The transient flow distribution to the break from the two SGs is a function of the individual SG pressures and the hydraulic resistances for the two flow paths.

Due to the more rapid de-pressurization of the faulted SG prior to the turbine stop valve closure, the intact SG pressure is higher than the faulted SG. This pressure difference causes a surge of flow from the intact SG to the break immediately following the stop valve closure as the difference in SG pressures adjust to the new flow network represented by the closed stop valves and the break. If the initial surge did not close the check valve, the differences in SG pressure would decrease and the flow from the faulted SG would increases while the flow from the intact SG decreases until a new quasi-equilibrium condition exists as shown in Figures 1 and 2.

The maximum reverse check valve flow shown in Figure 2 is approximately 46 % higher than the flow rate used in the Reference 1 analysis. This higher flow indicates that appreciable margin exists with the Reference 1 calculated torque. Using the

transient reverse flow and SG pressure data from Figures 1 and 2, the transient torque developed across the check valve disk with the Reference 1 methodology has been calculated in Reference 4. These results are shown in Figure 3. The Figure 3 results indicate that immediately following the turbine stop valve closure, the initiation of reverse flow from the intact SG to the break results in a calculated torque value that would exceed 2000 ft-lb. This initial torque is more than a factor of two higher than the value calculated in Reference 1. Therefore, the Figure 3 results demonstrate that the mass flow rate and MS pressure used in the Reference 1 calculation were chosen in a conservative manner.

Valve Static Pressure

A second major conservatism in the Reference 1 methodology is the static pressure assumed under the valve disk. The major contributor in the Reference 1 analysis to the valve closing torque is the differential pressure assumed across the valve disk. The differential pressure term used in Reference 1 was approximately 1.3 psi; and, this pressure difference generated approximately 95 % of the total torque calculated by Reference 1.

The differential pressure across the disk is the difference between the static pressure on the top side of the disk and the static pressure on the bottom side of the disk. For the static pressure on the top side of the disk, Reference 1 assumed the top side of the disk would represent a low flow region. Therefore, the static pressure on the top side of the disk was assumed to approach the fluid stagnation pressure. Since any flow through the top side of the disk results in a static pressure that is less than the fluid stagnation pressure, this assumption is inherently non-conservative. However, the difference between static and stagnation pressure on the top side of the disk was judged to be small and was more than compensated by the conservative assessment of static pressure under the disk made by Reference 1.

Reference 1 assumed that the fluid static pressure under the valve disk was equal to the fluid static pressure in the piping upstream of the MS check valve. The fluid static pressure in the upstream piping was calculated based on the piping cross sectional area of 593 in^2 based on the 27.5" pipe ID. As the steam flows into the MS valve body and flows underneath the valve disk, the valve flow area decreases; thereby causing the steam velocity to increase and its static pressure to decrease. Due to the orientation of the valve disk when it is up against its stop and due to the valve design (as shown on Reference 5), the flow

area continues to decrease as it travels from the leading edge of the disk to the point just upstream of the valve seat area. As the steam enters the valve seat area and clears the back end of the disk, the steam flow area increases. The flow area at the valve seat location is approximately 452 in^2 based upon the seat ID of 24" specified by Reference 5.

Although the 24" seat ID is not the minimum under disk flow area, it can be used to estimate the magnitude of the change in static pressure from the valve inlet through to the seat area location. By conservatively ignoring frictional losses associated with the check valve flow, the decrease in static pressure between the valve inlet and the valve seat area can be assumed to equal the increase in the velocity head between these two locations. The velocity head in turn is proportional to the square of the flow velocity(or inversely proportional to the square of the flow area). Consequently, for a 23.8 % decrease in flow area between the valve inlet and the valve seat location, the velocity head term would increase by 53.2 %. At the valve inlet, the velocity head term as calculated in Reference 1 was approximately 1.3 psi. Therefore, at the valve seat area, the velocity head term would be approximately 2 psi. This would result in a decrease in the static pressure under the disk of approximately 0.7 psi(2.0 psi - 1.3 psi).

Since the majority of the check valve flow will occur under the valve disk, the 0.7 psi magnitude decrease in static pressure on the underside of the valve disk is more than sufficient to compensate for any non-conservatisms introduced into the Reference 1 analysis due to the stagnation assumption for the valve area above the disk. This magnitude change in static pressure for the underside disk area would ensure that for the flow conditions analyzed in Reference 1 that the actual static pressure differential across the valve disk would be greater than the approximately 1.3 psi value used to calculate closing torque.

Break Area

A third area of conservatism in the Reference 1 analysis relates to the break size assumed for the limiting MSLB where operation of the check valve under reverse flow conditions is assumed to occur. Reference 2 evaluated both the containment pressure response and the RCS response to a 0.86 ft^2 main steam line break (MSLB). For this break size with no closure of the MS check valve both the peak containment pressure and the RCS core response were within the design basis conditions for Ginna Station. The peak containment pressure calculated for this MSLB was approximately 59 psig. Since this is below the containment design pressure of

60 psig, this break size was chosen as the threshold break size for evaluating valve closure torque in Reference 1.

Since the peak calculated containment pressure for the 0.86 ft^2 break is below the containment design pressure of 60 psig, it represents a conservative choice for the threshold break size. If additional iterations on peak calculated pressure as a function of break size had been performed, it would have been possible to justify a somewhat larger break size that would have still kept containment pressure below its 60 psig design value. The larger break size would have resulted in higher break flow rates; and, correspondingly, higher check valve reverse flow rates and disk torques. Although the increase in total reverse flow that would have occurred is expected to be small, it does represent an additional conservatism in the choice of mass flow rate used by RG&E in Reference 1 to analyze valve closing torque as a result of reverse flow.

RG&E Alternate Calculation

To perform a check on the adequacy of the Reference 1 methodology for determining valve torque, RG&E in Reference 4 also evaluated the valve closure torque that would result solely as a function of frictional differential pressure across the valve disk. Since most of the frictional pressure drop is expected to be due to losses associated with the valve disk, the overall check valve hydraulic resistance can be used as a means for checking the adequacy of the Reference 1 methodology.

From the original Bill of Material for the MS check valves the design differential pressure at 100 % power conditions with forward flow is 2.72 psi. Using this differential pressure and the 100 % power MS conditions for flow rate and pressure, Reference 4 calculated the hydraulic resistance for the valve for forward flow. For reverse flow condition, the check valve hydraulic resistance would be larger than that observed under forward flow conditions. The increase in hydraulic resistance for the valve would result primarily from the leading edge effect associated with the valve disk sitting on its stop. Since the leading edge of the valve disk under reverse flow protrudes approximately 2" into the flow stream, the disk would create increased turbulence and corresponding frictional losses under reverse flow.

Reference 4 conservatively used the forward flow hydraulic resistance for calculating frictional differential pressure across the valve disk under reverse flow conditions. The resulting differential pressure as a function of time based upon

the Figure 1 and Figure 2 SG pressure and check valve flow rates was then calculated. This differential pressure was then used to calculated the net load on the valve disk and the corresponding closing torque. The results of this calculation are shown on Figure 4, where it is compared to the transient Reference 1 methodology results previously discussed.

The alternate methodology based on frictional pressure drop shows a transient profile that is similar to the Reference 4 transient methodology results. The calculated torque values are approximately 18 % lower than the Reference 4 transient methodology; however, its results are still significantly higher than the torque value used in the Reference 1 static analysis. The difference with the Reference 4 transient results is attributed primarily to the following two conservatisms associated with the alternate methodology:

1. Use of forward flow hydraulic resistance for reverse flow.
2. Neglecting difference in static pressure differences between the top and bottom side of the valve disk

Therefore, although the closing torques calculated by the alternate methodologies are lower than those obtained with the Reference 1 methodology; they also demonstrated that at the beginning of reverse flow conditions the closing torque on the valve disk is appreciably higher than the 900 ft-lb required to initiate valve closure.

On-Going Activities

In addition to the information provided above RG&E has a number of on-going activities related to this issue. These activities include:

1. Independent Third Party Review of Valve Torque
2. Assessment of Means to Reduce Closure Torque

Third Party Review

As a result of the concern identified with the adequacy of the RG&E Reference 1 method for determining valve closure torque, RG&E has requested that Duke Engineering & Services (DE&S) perform an independent third party review of this issue. The results of the DE&S Independent Review are documented in Reference 6 and are summarized below.

Although DE&S performed a literature search for experimental data on swing check valve closing torque; no relevant information was found. Consequently, DE&S analytically assessed check valve closure torque based upon two alternate methodologies. One method used information for assessing torque on closure of tilting disk check valves; whereas, the second methodology used information for closure of butterfly valve disks. Valve closure torques were calculated for reverse flow rates that varied from the 603 lb/sec value used in Reference 1 to the maximum flow rate shown on Figure 2. For both methods conservative valve characteristics were chosen. When compared to the original RG&E method used in Reference 1, the two methods calculated torques that were respectively 10 % and 33 % less than the RG&E method.

Although the DE&S alternate methods calculated lower torques at 603 lb/sec than was used by RG&E in Reference 1; DE&S identified that the torque developed by the actual transient flow shown in Figure 2 resulted in maximum torques well in excess of the 912 ft-lb value calculated by Reference 1. Actual torque margins based upon transient flows ranged from 43 % to 91 % for the two alternate methods. Additionally, DE&S stated that the rapid increase in reverse flow experienced by the check valve would result in a transient impact loading on the valve packing that would cause valve movement at a lower torque than would be developed during normal valve torque testing. Based upon the large flow margin available between the flow used by RG&E in Reference 1 and the actual transient flow shown in Figure 2, DE&S concluded that the fluidynamic forces experienced by the check valve would be sufficient to close the check valves when experiencing the transient flow rates shown on Figure 2.

Finally, although the fluid flow may be sufficient to cause valve closure; DE&S stated that the present 600 ft-lb torque value used by RG&E to establishing packing compression is excessive based upon their experience with swing check valves. Consequently, DE&S recommended that the valve and packing configuration be reviewed and reworked as necessary so as to lower the packing torque used to set up the valve.

Reduction of Closure Torque

As a result of the on-going discussion and questions related to this issue between RG&E and the NRC, RG&E believes that it is prudent to reduce the torque required to initiate valve closure in order to return the valve to a condition more representative of the original design intent (i.e. gravity closing). In order to accomplish this, RG&E has relocated the check valve counterweights to their fully retracted position. This decreased

the moment arm associated with the counterweights by approximately six inches.

Since the two 150 lb counterweights act to prevent valve closure, their relocation has decreased the amount of torque required by reverse flow to initiate valve closure by approximately 150 ft-lb. For the nominal 600 ft-lb set-up torque used for establishing valve packing friction coming out of the 1999 Refueling Outage, this results in a 25 % reduction of the required flow induced torque to 450 ft-lb. For the largest As-Found measured torque of 900 ft-lb, the 150 ft-lb reduction decreases the required torque due to reverse flow by approximately 17 % to 750 ft-lb.

In addition to this short term action, RG&E is reviewing other long term actions that would be implemented in the 2000 Refueling Outage to decrease the torque required to initiate valve closure under reverse flow conditions. These actions include the complete removal of the counterweights as well as changes in the method used to pack the valve.

Conclusion

Based upon the conservatisms discussed above and the results of the independent assessment performed of valve closure torque, RG&E concludes that sufficient margin exists between calculated torque and the maximum As-Found measured torque to ensure that closure of the Main Steam check valves would occur under reverse flow for the most limiting Main Steam Line Break. The most conservative analytical assessment discussed above provides greater than 40 % margin to the maximum As-Found measured torque of 900 ft-lb. Due to this large amount of margin, RG&E concludes that a three dimensional computational fluid dynamics analysis of the check valve is not warranted.

To provide additional margin for present and future plant operation, RG&E has initiated actions to reduce the actual breakaway torque that would be needed for check valve closure. For present plant operation RG&E has re-positioned the valve counterweights to minimize the torque that acts to prevent valve closure. This action has decreased the torque needed to intitate valve closure by approximately 150 ft-lb. For the present operating cycle this activitity has provided, as a minimum, an addition 17 % of torque margin. Finally, for future operating cycle RG&E has initiated engineering activities to optimize packing of the valve so as to provide the minimum amount of packing friction needed for a leak tight packing configuration. Recommendations from these engineering activities would be implemented during the year 2000 Refueling outage.

FIGURE 1 - SG PRESSURE

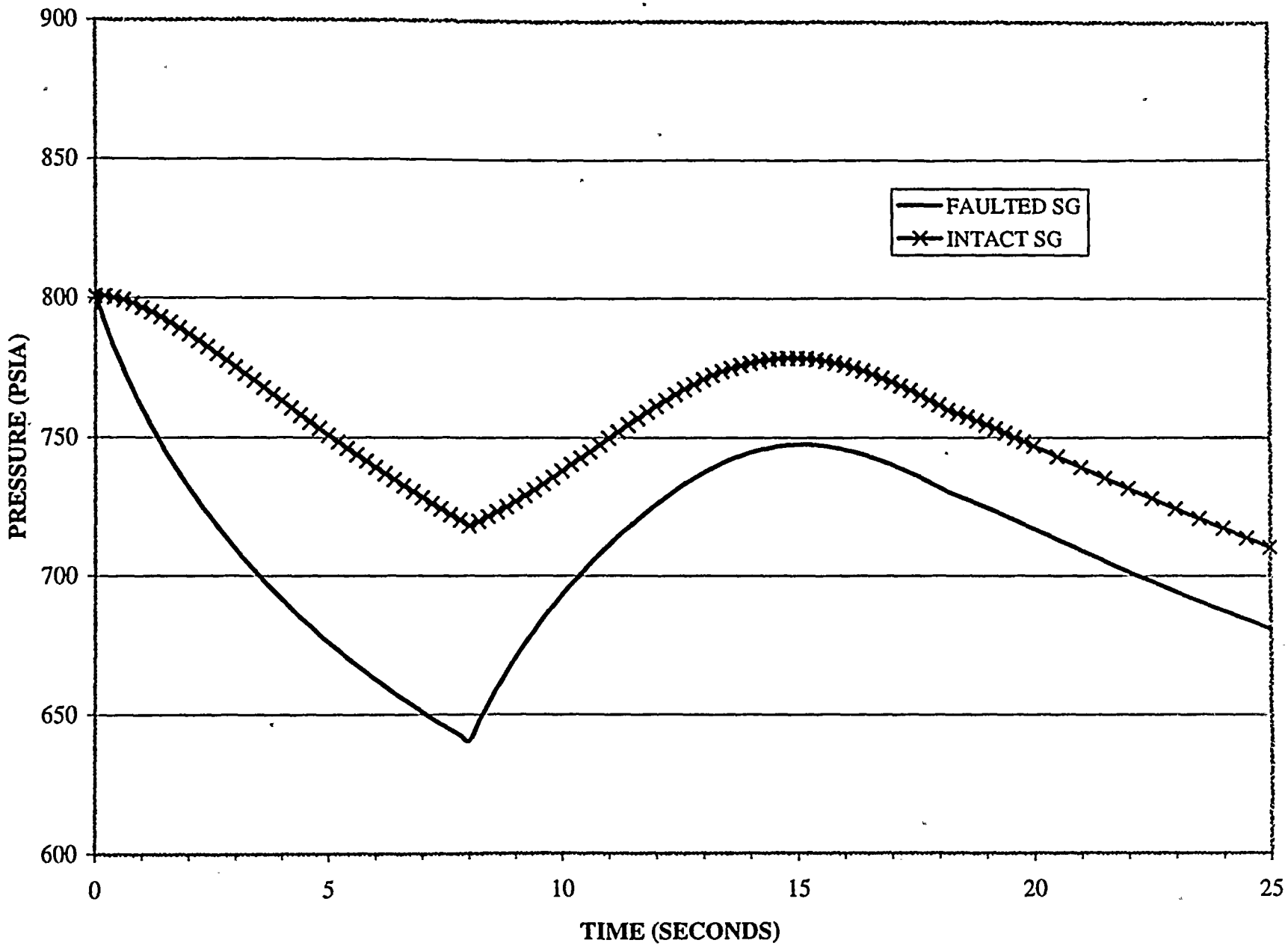


FIGURE 2 - BREAK FLOW DISTRIBUTION

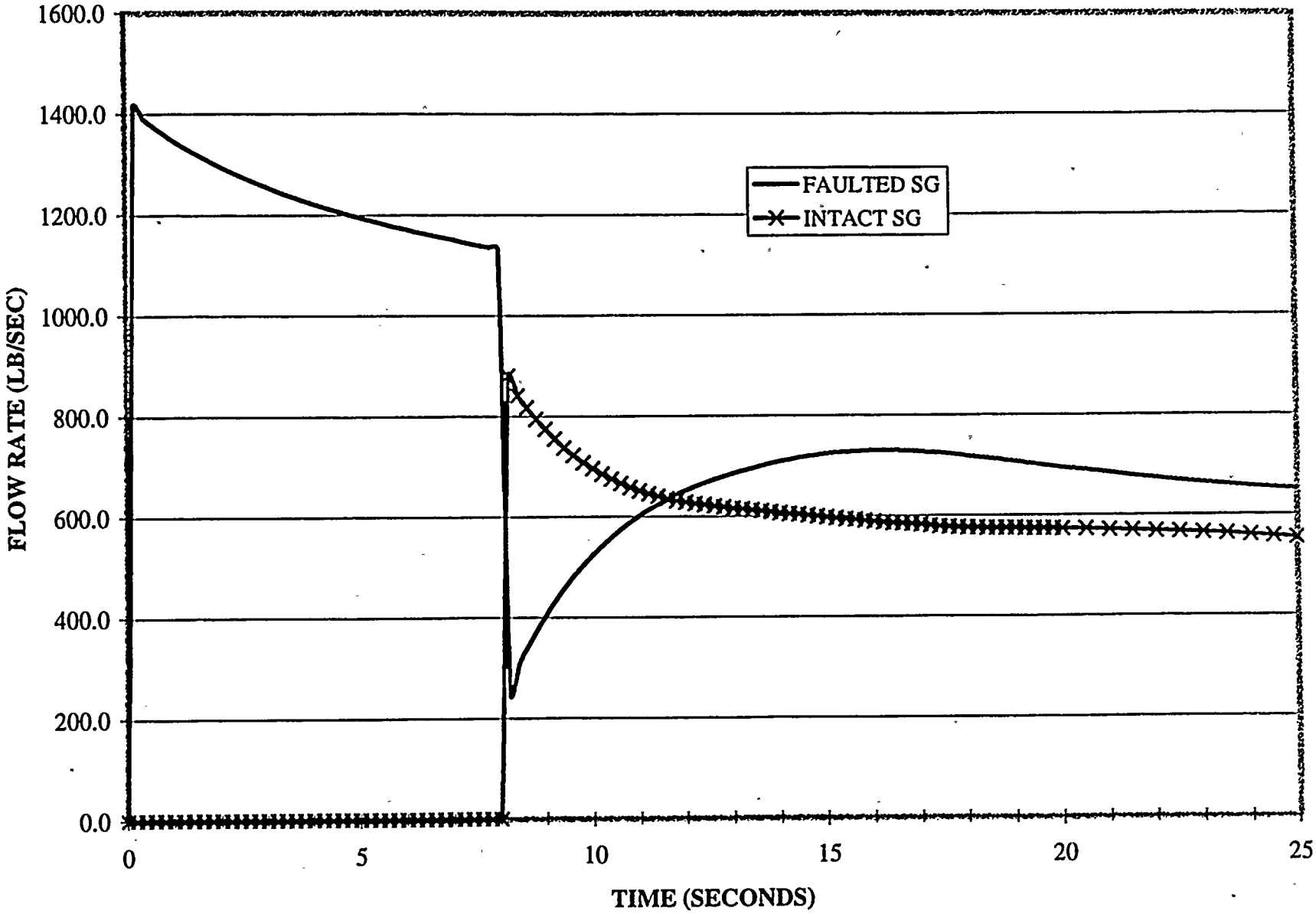


FIGURE 3 - CHECK VALVE TORQUE

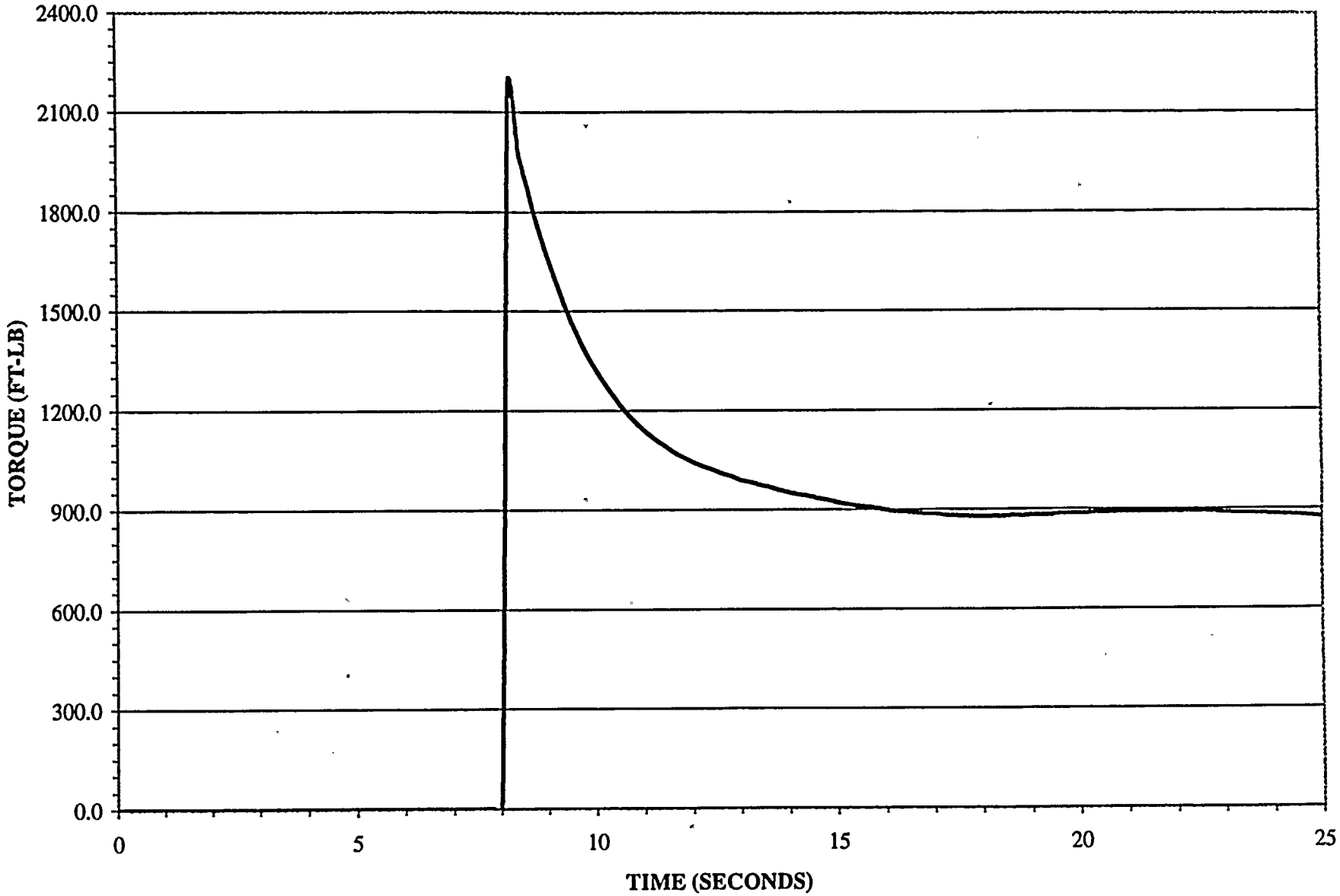
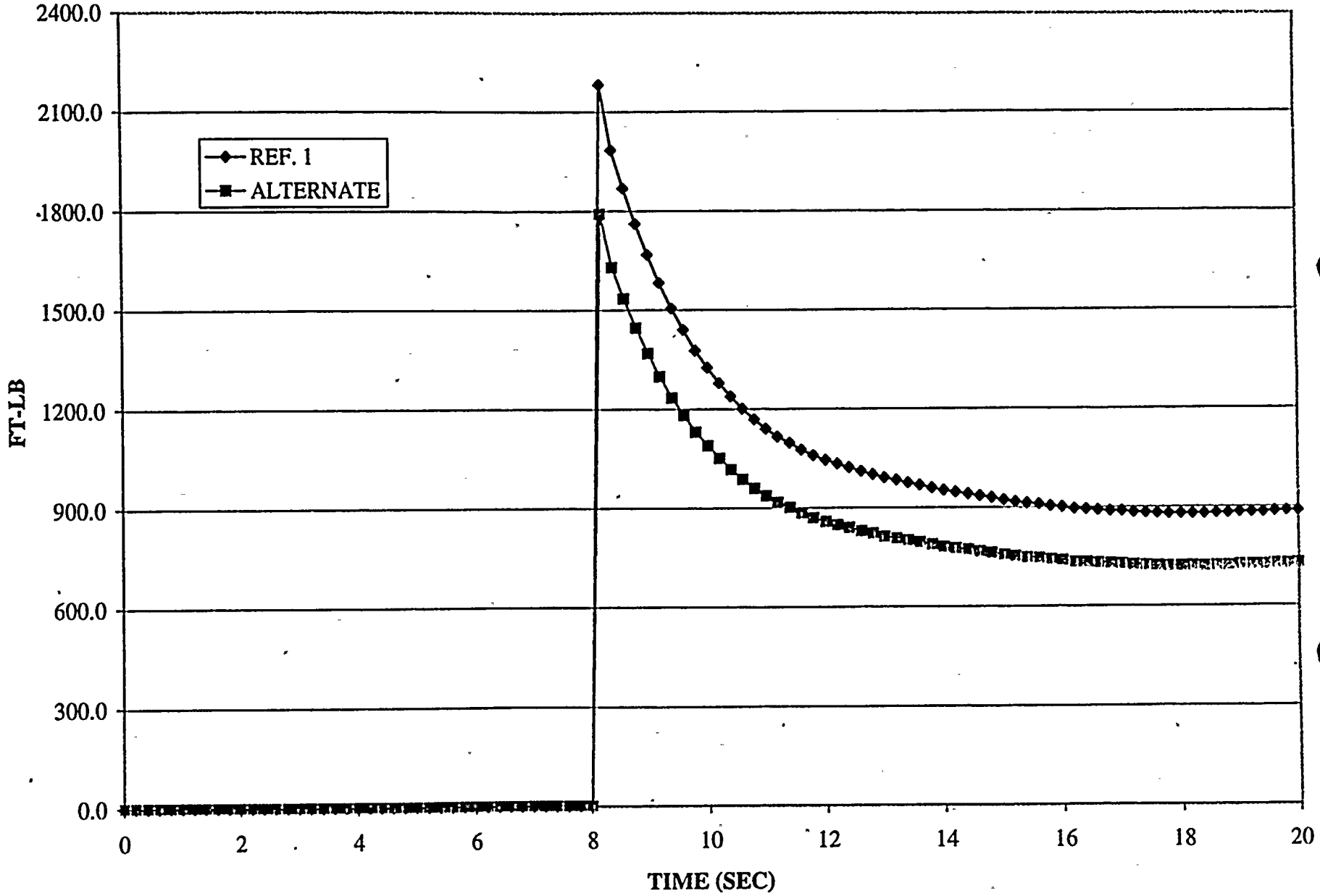


FIGURE 4 - CHECK VALVE TORQUE COMPARISON



RESPONSE TO NRC QUESTIONS

QUESTION 2:

What is the basis for assuming that the steam flow is non compressible?

RESPONSE:

In Section 5.1 of RG&E Design Analysis DA-ME-92-147, Revision 2; it is stated that the saturated steam flow through the check valve is assumed to be non-compressible since the pressure is relatively constant across the valve. The Design Analysis evaluates the closing torque generated at a specific point in time where the reverse flow rate through the check valve is 603.3 lbm/sec and the steam pressure at the valve is approximately 800 psia. The Design Analysis then calculates the pressure head associated with the steam velocity at the inlet to the check valve. Since the steam velocity pressure head term is small (e.g. approximately 1.3 psi), the difference between the fluid static pressure and stagnation pressure is approximately 0.15 % (1.3 psi / 800 psia). For steam velocities with Mach numbers less than 0.1 (e.g. velocity less than approximately 150 ft/sec), isentropic flow relationships for an ideal gas indicates that the difference between static and stagnation densities are less than 0.5 %. Therefore, the density change for a compressible fluid for the fluid conditions that would exist in the valve body are small and can be neglected. This represents the basis for the non-compressible assumption made in Section 5.1 of the Design Analysis.

It should be noted that Attachment 1 of Design Analysis DA-ME-92-147, Revision 2 includes a plot of torque versus steam flow for main steam line pressures of 700 psia, 800 psia and 900 psia respectively. The non-compressible assumption was not used to develop the torque valves calculated for these three steam pressures. For each steam pressure (e.g. 700 psia, 800 psia and 900 pisa) the corresponding saturated steam density was used to determine torque as a function of steam flow rate. The non-compressible assumption was only used for the calculation of the velocity pressure head for each flow condition and steam pressure.

QUESTION 3:

Since flow in the line is changing in mass flow rate and reversing direction, what is the basis for assuming constant pressure (during normal operation the flow past the check valve is about 914 lbm/sec; then, subsequent to the line break the flow at the check valve reverses and decreases to 603.3 lbm/sec)?

RESPONSE:

Design Analysis DA-ME-92-147, Rev.2 used a check valve flow and steam pressure at one point in time to calculate the corresponding closure torque developed by the flow and pressure conditions. This pressure and flow were chosen to bound the transient data. If the transient flow and pressure data were used, a transient torque curve could have been generated that would take into account the time dependent nature of the flow and pressure experienced by the check valve. This transient torque data has been provided in response to Question 1 and, it demonstrates the inherent conservatism in choosing a single bounding point.

Additionally, once sufficient torque is developed to overcome the valve packing friction, the resulting movement of the check valve disk into the flow stream would result in an increase in drag force across the valve disk which would ensure that the valve would go closed. Therefore, DA-ME-92-147, Rev. 2 did not need to evaluated flow and pressure conditions that would exist subsequent to the initiation of valve closure.

QUESTION 4:

Is the mass flow rate of 603.3 lbm/sec in the calculation based on choked flow at the exit?

RESPONSE:

The LOFTRAN computer program was used in DA-NS-99-054, Rev. 0 (Reference 2)to calculate the blowdown of the Steam Generators due to a steam line break. The LOFTRAN program calculated the transient flow at the break location as well as the transient flow supplied to the break by both Steam Generators. The break flow rate represents the summation of the two flow paths that feed the break. The actual total break flow is determined by use of a choked flow correlation for saturated steam. The choked break flow is primarily a function of both the break area size and the main steam line pressure at the break location.

QUESTION 5:

How was the mass flow rate coming from the "line break" SG considered in the calculation of the 603.3 lbm/sec coming from the "operational" Steam Generator?

RESPONSE:

The flow rate out of the break at any point in time is determined based upon choked flow, the break size and the local steam line pressure at the break location. The break flow is fed by flow that reaches the break from both SGs after the turbine stop valves have closed. Consequently, the break flow represents the summation of the two individual flow paths. The transient flow rates for the two flow paths that supply the break are shown in Figure 2.

Prior to closure of the turbine stop valves, the flow rate exiting the faulted SG exceeds the break flow (i.e. a portion of the steam flows to the turbine). Therefore, all of the flow out of the break is supplied by the faulted SG up to the time that the turbine stop valves close. No flow from the intact SG reaches the break until after the turbine stop valves close. The 603.3 lb/sec value used in the determination of valve closure torque under reverse flow conditions was chosen to be a conservative assessment of the reverse flow conditions that would exist for the check valve.

QUESTION 6:

What is the basis for assuming the check valve closes in one (1) second?

RESPONSE:

Section 7.1 of Design Analysis DA-ME-92-147, Rev. 2 states that the mass flow and pressure conditions at t = 1 second were used since this is the check valve closure time assumed. The one second time represents the typical UFSAR Chapter 15 accident analysis time for Main Steam check valve closure following initiation of reverse flow from a design basis double ended guillotine rupture. Consequently, the mass flow rate of 603.3 lbm/sec and 800 psia represent the LOFTRAN calculated values for flow from the "intact" SG at the one second time in the main steam line break transient as calculated by DA-NS-99-054, Rev. 0. As stated in Section 5.3 of DA-NS-99-054, Rev. 0; the use of the flow and pressure at 1 second into the transient is conservative since the actual flows and pressures that would exist following the Turbine Stop valve closure generate higher valve closure torques. This has been demonstrated by the transient torque curve provided in response to Question 1.

In reality for this smaller steam line break, reverse flow through the check valve from the "intact" SG would not occur until after the Turbine Stop Valves have closed terminating flow from the two SGs to the Turbine. This would occur after 1 second in time. This has been demonstrated by the transient flow and pressure results provided for the response to Question 1 (Figures 1 & 2).

The actual value of one second has no significant impact on the DA-ME-92-147, Revision 2 analysis.

QUESTION 7:

What is the basis for the check valve disc being treated as a flat circular disc? Won't there be flow on both sides of the disc since the disc is round with gaps between the disc and the valve body?

RESPONSE:

The assumption made in Reference 1 of treating the check valve as a flat disk was used to determine an appropriate drag co-efficient for steam flow over the valve disk. The drag co-efficient was then used to calculate an appropriate drag force and corresponding moment. The flat disk was used since the front edge of the valve disk that sits in the flow stream under reverse flow is a circular disk with a thickness of approximately 3.75" as shown on Reference 5. The bottom side of the disk is flat over its entire surface. The top side of the disk is flat over approximately the first half of the disk.

At the top side center of the disk the disk hinge arm is attached to the disk by a hex nut. Any flow above the disk over the back half of the disk will also experience interaction due to the presence of the disk arm. Since the disk arm and the arm hex nut connection at the center of the disk provide a flow obstruction for flow on the top side of the disk, their presence would contribute to increased drag on the disk. Therefore, it was judged that ignoring their presence and treating the disk as a flat circular disk was conservative for assessing an appropriate drag co-efficient for the valve disk. Additionally, with the valve disk full open up against it stop, the valve disk presents a 15° negative angle of attack (angle to flow stream below horizontal orientation) under reverse flow conditions. Due to this negative angle of attack, the valve disk would generate a drag load that would act on the valve body in a direction that would cause it to go closed.

Since the leading edge of the flow disk protrudes approximately 2" below the top of the valve body ID, the portion of the valve flow above the valve disk would be expected to be scooped into the valve body area above the valve disk. The flow area above the disk is large in relationship to the valve flow area that would push flow above the valve disk. This large flow area would cause the steam velocity above the disk to be significantly below its value in the valve body. This reduction in velocity would cause the static pressure of the fluid above the valve disk to approach the fluid stagnation pressure.

It should also be noted that Reference 1 conservatively neglected the projected area for the back half of the valve disk when assessing the drag force acting on the valve disk body. This decreased the total drag force calculated in Reference 1 by 50 %. Additionally, with the Reference 1 methodology the moment calculated for the drag force represents only a small percentage of the total calculated moment. Only approximately 5 % of the total moment calculated by Reference 1 results from the drag force calculation. Consequently, the impact on the flat disk drag co-efficient associated with the presence of the disk arm and hex nut attachment is expected to have a minimal impact on the torque calculation performed by Reference 1. Therefore, any uncertainty associated with the flat disk assumption is expected to be negligible; and, would be bounded by the conservatisms discussed in the response to Question 1.

QUESTION 8:

What are the area and dimensions of clearance between the open disc circumference and the valve body? This information is needed to determine the area that is available for steam flow to exit the space above the open disc. And please provide, if readily available in conjunction with your analysis, the cross-sectional area:

- for steam flow to enter the area above the open disk,
- inside the inlet pipe to the valve,
- at the most flow restrictive point inside the open check valve (e.g., the minimum throat area),
- inside the outlet pipe from the valve,

and, the volume:

- above the disk,
- in the valve body upstream of the minimum throat area,
- in the valve body downstream of the minimum throat area.

RESPONSE:

RG&E presently has no quantitative information from either the vendor (Atwood-Morrill) or past on-site examinations on the clearances between the valve disk and the valve body. Based upon the vendor drawing (Reference 5) and the full open orientation of the valve disk, it is expected that the clearance varies along its entire circumference. The maximum gap dimension is expected to occur at the leading edge of the valve disk. The minimum clearance would be expected to occur at the hinge pin location.

With regard to the specific areas and volumes requested by the NRC, no quantitative information on volumes is presently available from the vendor drawing (Reference 5); however, since Reference 5 is a scaled drawing it may be possible to approximate the requested volumes be using scaled dimensions from the drawing. The quantitative cross sectional area information requested by the NRC based upon Reference 5 is listed below:

1.	Steam flow to enter the area above the open disk	Not Specified
2.	Inside the inlet pipe to the valve	594 in^2
3.	The most flow restrictive point inside the open check valve	Not Specified
3A.	Flow area at the valve seat location	452 in^2
4.	Inside the outlet pipe from the valve	594 in^2

RESPONSE TO NRC QUESTIONS

The results for items 2 and 4 are based upon the nominal piping inside diameter of 27.5" for the 30" Main Steam piping attached to the valve body. The weld prep details on Reference 5 support this dimension. The valve areas for flow to enter the valve top and for the minimum restriction location under the valve disk are not specified; however, the flow area for the valve seat location has been provided based upon the seat ID listed on Reference 5.

RG&E has discussed with Atwood-Morrill the availability of the information on valve disk clearances, valve areas and valve volumes as requested by the NRC. Presently Atwood-Morrill has stated that this information is unavailable. RG&E is pursuing with Atwood-Morrill the possibility of obtaining this information; however, its future availability is uncertain at this time.

REFERENCES:

1. RG&E Design Analysis DA-ME-92-147, Rev. 2
2. RG&E Design Analysis DA-NS-99-054, Rev. 0
3. RG&E Design Analysis DA-NS-99-054, Rev. 1
4. RG&E Design Analysis DA-ME-99-070, Rev. 0
5. Atwood-Morrill Dwg 20729-H, Rev. 3B
6. Duke Engineering & Services Report RG0007-T14-001, "Assessment of Main Steam Non-Return Check Valve Closure Analysis", Rev. 0