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ROBERT C. MECREDY
Vice President
Nuclear Operations

March 14, 2000

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

Subject: Main Steam Check Valves
R. E. Ginna Nuclear Power Plant
Docket No. 50-244

Dear Mr. Vissing:

In our letter to you of September 24, 1999, we responded to several questions arising from the NRC's Inspection Report 99-05. Within those responses, RG&E referenced a Duke Engineering and Services Report RG0007-T14-001, "Assessment of Main Steam Non-Return Check Valve Closure Analysis", Rev. 0.

Attached for your review is a copy of that referenced report.

Very truly yours,

Robert C. Mecredy

Attachment
gjw\548

xc: Mr. Guy S. Vissing (Mail Stop 8C2)
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Division of Licensing Project Management
Office of Nuclear Reactor Regulation
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1000112

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Document No. RG0007-T14-001

Assessment of Main Steam Non-Return Check Valve Closure Analysis

Table Of Contents

1.0 Objective

The objective of this report is to perform the following assessments:

1) Evaluate the methodology and conservatism used to calculate the closing moments in the analysis, and assess the need to perform more sophisticated flow analysis, such as 3D Computational Fluid Dynamics (CFD) flow modeling.

2) Evaluate the analysis methodology of treating the back of the check valve disc as a flat circular disc, and assess the need for the analysis to address flow around the disc.

2.0 Discussion

Typically this type of check valve is set up with minimal breakaway loads such that the combined moment of the disc assembly overcomes the counter weight and frictional resistance such that the valve still begins to close under a no flow condition (free swing closed). These check valves have a rotating horizontal shaft with two valve body penetrations to accommodate a dual counterweight arrangement. The pressure boundary is maintained to be leak tight along the shaft by use of two packing stuffing box arrangements. Typically it is the translating or rising stem packing arrangements that require higher packing loads, not the rotating stem packing. The amount of packing resistance (breakaway moment) necessary to close the valve from the full open position with no flow (900 ft-lb) seems extremely high. With recent valve rework and change to the new wedge type packing arrangement, it may no longer be necessary to tighten the packing as much to achieve a tight seal. DE&S and Duke Power have not encountered this type of valve with this high a frictional load. DE&S and Duke Power have not performed any testing to date to substantiate fluidynamic torque or moment effects on this type of valve.

3.0 Evaluation and Assessment

3.1 Current Calculation Method:

The reference 3 analysis is a developed methodology that tries to calculate a result where there is no clear industry data or normally accepted technique. As such, there are many areas where it would be easy for any reviewer to take issue. While this method would not have been the selected approach by DE&S (based on experience with our valve testing and development of valve fluidynamic response,) it remains a viable and reasonable approach to the problem. Given this method of approach, the calculation appears reasonable and complete. Our selected methods would (and are) based primarily on results of other valve type tests of closed conduit flow models and not strictly on free stream aerodynamic or fluidynamic data. Without any validating data, any single approach including a CFD model, will require a large operating margin for greater assurance.

The reference 3 calculation applied a conservative flow rate (603.3 lbm/sec) to obtain a limiting value for the packing friction. It would seem more appropriate to calculate the expected fluidynamic torque at more realistic flow rates and later review or apply margins based upon the good engineering judgement and estimates of the accuracy. The reference 8 design analysis identifies reverse flow begins at 8.0 seconds and quickly ramps to a peak reverse flow of 881.6 lbm/sec at time 8.2 seconds. (This is a ramp speed of 4,408 lbm/sec^2.) The reverse flow rates are identified as equal to 817.9 lbm/sec at 0.6 seconds later (8.6 seconds) and 774.8 lbm/sec at a full second later (9.0 seconds). These flows are more appropriate for use as the valve is expected to be closed within this (one second) interval.

The methodology for calculation of fluid moment (torque) used by Reference 3 uses drag force and pressure force. The use of drag coefficient for flat circular plate in a free flow stream is likely underestimated since the backside of disc is not flat and closed conduit flow tends to increase this drag. The addition of the central disc hub and disc arm adds significantly to the fluidynamic drag of the disc structure at a location below the hinge pin; thus adding to the closing moment calculated. Therefore, the use of the Bernoulli equation to account for the pressure drag is oversimplified and likely underestimates the fluid moment. Also, both the lift and the drag forces combine to generate the torque. (Note: At the angle of approach for the disc, the lift force is a downward closing force.)

In general, a body moving through a fluid experiences a drag force, which is usually divided into two components: friction drag and pressure drag. Frictional drag comes from friction between the fluid and the surface over which it is flowing. This friction is associated with the development of boundary layers. Pressure drag comes from the eddying motions that are set up in the fluid by the passage of the body. This drag is associated with the formation of a wake, which is similar to that seen behind a passing boat. Formally, both types of drag are due to viscosity, but the distinction is useful because the two types of drag are due to different flow phenomena. Frictional drag is important for attached flow and it is related to the surface area exposed to the flow. Pressure drag is important for separated flows, and it is related to the cross-sectional area of the body. In the current closure analysis, the pressure drag is controlling.

As there is a void in the industry knowledge, alternate approaches to the same result should be developed and the results compared. This review will, therefore, look at alternate methods for the extrapolation of this phenomenon.

Since the governing equation of motion is rotation along the valve shaft axis (angular motion), a one dimensional closure analysis is sufficient to describe the motion of the swing check valve. This would indicate that a three-dimensional CFD model is an excessive analysis method to answer this question based upon it's high cost and lack of any better certainty in accuracy. Without a benchmark, our experience has concluded that CFD is no more accurate until verified and conformed to known results or alternate analysis.

Two alternate methods of evaluating the Rochester Gas and Electric Corporation swing check valve fluidynamic torque (reference 3) are herein performed to determine relative validity and conservatism of the reference 3 calculation. All the methods used in the reference 3 calculation, these two assessments and any CFD model would require verification in order to apply with great certainty. However, if separate approaches yield similar results; uncertainty is decreased.

As presented in the following sections, the moment coefficient or torque coefficient based on test data is essential to estimate the fluid moment or fluidynamic torque on swing check and butterfly valves. These methods, while not strictly applicable to the subject swing check valve, are validated and generally accepted engineering practice. In any case, when the valve is looked at as a control volume, any energy lost within this volume must end up somewhere. In most valve designs and calculation methods, it is generally assumed that the valve disc or closure member absorbs the majority of this energy loss.

3.2 Alternative Method 1:

This section presents the formula to estimate the fluid moment applied to the tilting disc check valve under reverse flow steady state condition. This formula is based on References 1 and 2.

In accordance with Reference 2, the fluid moment on a disc in steady state condition is:

$$M_L = bA_d \frac{\gamma V^2}{2g_c K_f^{\ 2}}$$

Where:

Symbol		Description	Units
M_L	=	Fluid moment on check valve disc	ft-lbf
b	=	Distance from hinge pin to centerline of disc	ft
A_d	=	Circular disc area	ft^2
γ	=	Steam density	lbm/ft^3
V	=	Steam velocity based on pipe	ft/sec
g_c	=	Conversion factor, gravitational acceleration = 32.17 lbm-ft/lbf-sec^2	lbm-ft/lbf-sec^2
K_f	=	Moment coefficient	dimensionless

Based on Reference 3,

Symbol		Description	Units
b	=	Distance from hinge pin to centerline of disc, = 15.5 in (reference 3, page 8 of 10).	in
D	=	Diameter of disc = 25.5 in (page 6 of 10).	in
A_d	=	Circular disc area = $\pi/4 * D^2 = \pi/4 * 25.5^2 = 510.7\ in^2 = 3.55\ ft^2$	ft^2
γ	=	Steam density =1.75 lbm/ft^3 (page 7 of 10)	lbm/ft^3
V	=	Steam velocity based on pipe area	fps
A_{PIPE}	=	Area of pipe (593.9 in^2 or4.12 ft^2)	ft^2
W	=	Weight flow of steam	lbm/sec

for V calculations: $V = w / \gamma / A_{PIPE}$

w, lbm/sec	V, fps
881.6	122.3
816.9	113.3
774.8	107.5
603.3	83.7

The moment coefficient, K_f for reverse flow is normally obtained from experimental data. After an intensive literature survey, this coefficient for swing check valve could not be found. However, Reference 2 conducted extensive experimental tests on a 16-inch diameter tilting disc check valve. Both steady state flow coefficients (defined as 1/(square root of resistance coefficient)) and moment coefficients were determined for forward and reverse flow. It showed that the steady state flow coefficients and moment coefficients were very similar in magnitude. The reason for this is that the majority of the pressure drop across the valve is created by the disc structure itself. This characteristic for tilting disc check valves is assumed to be applicable to the swing check valve since both behave similarly as a check valve and the disc remains the major source of flow obstruction in the reverse direction.

The manufacturer of the subject swing check valve has been contacted. They provided the resistance coefficient of 0.8 in the reversed flow direction (See Attachment A). This leads to a moment coefficient of 1.118 (the inverse square root of 0.8) for this main steam swing check valve. For added conservatism, the flow resistance of the body will be subtracted from the vendor's value. For Reference 6 page A-26, the contraction and expansion losses are calculated as follows:

$$K_{CONTRACTION} = \frac{0.8 \times SIN\left(\frac{\Theta}{2}\right) \times \left(1-\beta^2\right)}{\beta^4} = 0.0514$$

$$K_{EXPANSION} = \frac{2.6 \times SIN\left(\frac{\Theta}{2}\right) \times \left(1-\beta^2\right)^2}{\beta^4} = 0.0464$$

$$K_{BODY} = K_{CONTRACTION} + K_{ENPANSION} = 0.0978$$

$$K_{DISC} = K_{TOTAL} - K_{BODY} = 0.8 - 0.0978 = 0.7022$$

$$K_f = \frac{1}{\sqrt{K_{DISC}}} = \frac{1}{\sqrt{0.7022}} = 1.1934$$

Where:

Symbol		Description	Units
K_{BODY}	=	Body resistance loss coefficient	none
K_{DISC}	=	Disc resistance loss coefficient	none
K_{TOTAL}	=	Total valve resistance loss coefficient, 0.8 per vendor, reference Attachment A	none
$K_{CONTRACTION}$	=	Contraction resistance loss coefficient	none
$K_{EXPANSION}$	=	Expansion resistance loss coefficient	none
$\Theta_{CONTRACTION}$	=	Contraction angle = 18° as measured from the 1/4 scale reference 7 vendor drawing	°
$\Theta_{EXPANSION}$	=	Expansion angle = 21° as measured from the 1/4 scale reference 7 vendor drawing	°
β	=	Beta ratio, 24 in port diameter / 27.5 in pipe inside diameter = 0.8727 from reference 3 and 7	none
K_f	=	Moment Coefficient	none

Hence the fluid moment is:

$M_L = (15.5/12) *(510.7/144) * 1.75*V^2 / (2*32.17*1.1934^2)$, ft-lbf

Results:

W, lbm/sec	V, fps	M_L, ft-lb	Margin at 912 ft-lb
881.6	122.3	1308.6	43%

W, lbm/sec	V, fps	M_L, ft-lb	Margin at 912 ft-lb
816.9	113.3	1123.0	23%
774.8	107.5	1011.0	11%
603.3	83.7	612.9	-33%

This fluid moment plus moment due to the disc gravitational torque are higher than the summation of moments due to friction (912 ft-lb) and counter weight gravitational torque when the above margin is positive.

This methodology also concludes that this swing check valve under reverse flow will close and closure will be initiated within the first sec of flow with a 43% margin.

3.3 Alternative Method 2:

This section presents the formulae to determine the fluidynamic torque applied to the butterfly valve disc under steady state conditions thereby initiating valve closure. These formulae are based on the methodology of Reference 4.

While the fluidynamic torque, or moment, response of swing check valves is not well known, the response of butterfly valves is well understood. Additionally, these data have the advantage that they are developed at many angles of approach velocity, including the 75° orientation of the subject swing check valve disc. The basis of these works and data is the following equation where the fluidynamic torque coefficient (C_T) is experimentally determined:

$$T_D = C_T \times D_{DISC}^3 \times \Delta P$$

where:

Symbol		Description	Units
T_D	=	Fluidynamic torque	in-lb
C_T	=	Fluidynamic torque coefficient	none
D_{DISC}	=	Disc diameter = 25.5 in from reference 3	in
ΔP	=	Differential pressure	psid

This is based on a first principle approach. The moment or torque is created by the differential pressure forces along all surfaces (on both sides of the disc) which acts about a center that is normally located forward of the physical center of the disc (in the upstream direction). This location is referred to as the aerodynamic center and generally occurs about a quarter of the chord length or disc diameter from the leading edge of the foil (reference 5). The basis of the above equation is from the following derivation that includes two fractional unknown components, f_1 and f_2. It is still true that for either the full open swing check valve with reverse flow or for the butterfly valve disc at 75° open that the majority of the pressure loss occurs across the disc and that little loss is associated with the valve body. For conservatism in the alternate 1 method, the body resistance was subtracted. However, in this analysis the most conservative result comes when f_1 is assumed as 1.0 to obtain the smallest lever arm length. This means that f_1 is approaching or approximately equal to 1.0. The following provides the basic derivation of the fluidynamic torque formula:

$$T_D = F_{FLUIDYNAMIC} \times L_{LEVER\ ARM}$$

$$T_D = [\, f_1 \times \Delta P \times A_{DISC} \,] \times [\, f_2 \times D_{DISC} \,]$$

$$T_D = \left[\frac{f_1 \times \Delta P \times \pi \times D_{DISC}^{\ 2}}{4} \right] \times [\, f_2 \times D_{DISC} \,]$$

$$T_D = \left[\frac{f_1 \times f_2 \times \pi}{4} \right] \times \Delta P \times D_{DISC}^{\ 3}$$

Therefore:

$$C_T = \left[\frac{f_1 \times f_2 \times \pi}{4} \right] \quad \text{and} \quad f_2 = \left[\frac{C_T \times 4}{f_1 \times \pi} \right]$$

Where:

Symbol		Description	Units
$F_{FLUIDYNAMIC}$	=	Fluidynamic force	lbf
$L_{LEVER\ AMR}$	=	Lever arm length	in
T_D	=	Fluidynamic torque	in-lb
f_1	=	Fractional component of differential pressure that acts on the disc, approximately = 1.0	none
ΔP	=	Differential pressure	psid
D_{DISC}	=	Disc diameter = 25.5 in from reference 3	in
f_2	=	Fractional component of D_{DISC} that defines the location of the aerodynamic center from the shaft centerline	none

Symbol		Description	Units
C_T	=	Fluidynamic torque coefficient	none

Therefore, the location of the aerodynamic center can be approximated as the following when f_1 is set to 1.0:

$$f_2 = \left[\frac{4 \times C_T}{\pi} \right]$$

The reference 4 document has three values for C_T at the 75° position of 0.2270, 0.3457 and 0.2074 with a corresponding resistance coefficient of 0.88 (for all three). Use of the lowest value provides the most conservative (lowest) distance to the aerodynamic center. Therefore:

$$f_2 = \left[\frac{4 \times 0.2074}{\pi} \right] = 0.264$$

As the disc diameter from reference 3 is 25.5 in, then:

$$L_{LEVER\ ARM} = 0.264 \times D_{DISC} = 0.264 \times 25.5 = 6.732 \text{ in}$$

As the effective lever arm is forward of the disc center by 6.732 in and the hinge pin is located 15.5 in back of the disc center (reference 3), the total effective lever length ($L_{EFFECTIVE}$) from the aerodynamic center to the hinge pin is then:

$$L_{EFFECTIVE} = 6.732 + 15.5 = 22.232 \text{ in}$$

The reference 4 states that the combination of the $C_T \times K$ of this publication produces an upper bound or high valve of torque. More conservatism is added by reviewing many low pressure valve data sets for the lowest value of C_T and by using the swing check valve manufacturer K value of 0.8 in lieu of the Reference 4 value of 0.88. The lowest 75° disc angle C_T value found in 52 butterfly valve representative data sets was 0.163 which generally have aspect ratios equal to or greater than 0.15 (listed in Attachment C). Low pressure (150 psi or less) valves were selected as these tend to be the lower aspect ratio disc designs. The aspect ratio is the comparison of disc thickness over the disc diameter. Lower aspect ratio discs produce lower fluidynamic torque. Using this value $L_{EFFECTIVE}$ is calculated as:

$$f_2 = \left[\frac{4 \times 0.163}{\pi} \right] = 0.208$$

$$L_{LEVER\ ARM} = 0.208 \times D_{DISC} = 0.208 \times 25.5 = 5.304 \text{ in}$$

$$L_{EFFECTIVE} = 5.304 + 15.5 = 20.804 \text{ in}$$

The differential pressure calculation will use the reduced K value determined in the first alternate method for conservatism of 0.7022. From reference 6 the differential pressure across the valve may be calculated as:

$$\Delta P = \left[\frac{w^2 \times K \times \overline{V_1}}{0.525^2 \times Y^2 \times D_{PIPE}{}^4} \right]$$

$$\Delta P = \left[\frac{w^2 \times 0.7022 \times 0.5689}{0.525^2 \times 1.0^2 \times 27.5^4} \right]$$

Where:

Symbol		Description	Units
w	=	Rate of Flow = 603.3 from reference1; and 881.6, 816.9 and 774.8 from reference 8.	lbm/sec
K	=	Valve reverse resistance coefficient = 0.8 per manufacturer less the estimated body losses	none
V_1	=	Specific volume of fluid = 0.5689 ft^3/lb per reference 3	ft^3/lb
Y	=	Net expansion factor for compressible fluid = 1.0 for flows well below mach 1 and per reference 3	none
D_{pipe}	=	Approach pipe inside diameter = 27.5 in from reference 3	in
ΔP	=	Differential pressure	psid

Results:

W, lbm/sec	ΔP, psid
881.6	1.970
816.9	1.691

W, lbm/sec	ΔP, psid
774.8	1.521
603.3	0.922

Note: Smaller Y values increase the ΔP.

Therefore, the total fluidynamic torque should be the differential pressure times the disc area times the effective moment arm as follows:

$$T_D = \frac{\Delta P \times \pi \times D_{DISC}^2 \times L_{EFFECTIVE}}{4 \times 12}$$

$$T_D = \frac{\Delta P \times \pi \times 25.5^2 \times 20.804}{4 \times 12} \text{ ft-lb}$$

Results:

W, lbm/sec	ΔP, psid	T_D, ft-lb	Margin at 912 ft-lb
881.6	1.970	1743.9	91%
816.9	1.691	1497.3	64%
774.8	1.521	1347.0	48%
603.3	0.922	816.7	-10%

This fluid moment plus moment due to the disc gravitational torque are higher than the summation of moments due to friction (912 ft-lb) and counter weight gravitational torque when the above margin is positive.

This methodology also concludes that this swing check valve under reverse flow will close and closure will be initiated within the first sec of flow with a 91% margin.

Based on the above even if the C_T value were equal to zero the results are as follows:

$$T_D = \frac{\Delta P \times \pi \times 25.5^2 \times 15.5}{4 \times 12} \text{ ft-lb}$$

Results:

W, lbm/sec	ΔP, psid	T_D, ft-lb	Margin at 912 ft-lb

W, lbm/sec	ΔP, psid	T_D, ft-lb	Margin at 912 ft-lb
881.6	1.970	1299.305	42%
816.9	1.691	1115.593	22%
774.8	1.521	1003.569	10%
603.3	0.922	608.4638	-33%

Additionally the results of the asymmetric flow pattern tests for the EPRI PPM and reference 4 showed that discs in an offset velocity profile result in even higher fluidynamic torque. This valve forces an asymmetric flow pattern on the disc due to the offset of the shaft center of rotation.

4.0 Margins

While extensive search yielded no directly applicable test data, two methods used here and the approach of the reference 3 calculation are all based on reasonable variations of fluidynamic methods and all three yield similar results. Even a three dimensional Computational Fluid Dynamic (3D CFD) calculation will not be accurate unless benchmarked and conformed against actual results of a similar fluid model. Therefore, the employment of a CFD model, without supportive data, has no more certainty than any of the three forgoing analyses.

Margin exists in the selection of all variables used. The greatest margin is in the flow rate used. This is because the fluidynamic torque is related to the flow rate squared. As the peak flow with the intact steam generator at 8.2 seconds is approximately 881 lbm/sec in lieu of the 603.3 used in the original calculation the margin on these results is:

$$\text{MARGIN} = \left[\frac{881^2 - 603^2}{603^2} \right] \times 100 = 113\%$$

5.0 Flow Rate Transients

Transients always increase load results by large amounts. As can be seen in the Attachment B figure the flow at 8 seconds into the intact steam generator failure flow increases from essentially zero to 881.6 lbm/sec in less than 0.2 seconds. (This is a ramp speed of 4,408 lbm/sec^2.) This rate of change will increase the initial torque to start the valve closure motion strictly on the basis of a momentum transfer. This impact load will also add directly and significantly to the closing torque calculated by this or any other approach. Transient load application results are often 1.5 to 4 times greater than normal loads when the rate of loading is high. Although no direct test data was located to determine this effect, it is generally accepted that high rates of load application will increase, rather than decrease, the amount of torque generated.

6.0 Bearing Friction

None of these analyses takes into account the bearing friction torque. This torque is related to the bearing coefficient of friction and the differential pressure force. This is non-conservative. The coefficient of friction is generally around 0.25 but could be as high as 0.6 in a raw water (e.g. dirty) system. This system is anticipated to be a clean system. However, this torque can be calculated based on reference 4 as follows:

$$T_{BRG} = \frac{\pi \times D_{DISC}^2 \times D_{SHAFT} \times \mu \times \Delta P}{96}$$

$$T_{BRG} = \frac{\pi \times 25.5^2 \times 3 \times 0.25 \times 1.05}{96} = 16.8 \text{ ft} - \text{lb}$$ at a reasonable bearing coefficient of friction;

$$T_{BRG} = \frac{\pi \times 25.5^2 \times 3 \times 0.6 \times 1.05}{96} = 40.2 \text{ ft} - \text{lb}$$ at a bounding bearing coefficient of friction.

Where:

Symbol		Description	Units
T_{BRG}	=	Bearing friction torque	ft-lb
D_{DISC}	=	Disc diameter = 25.5 in from reference 3	in
D_{SHAFT}	=	Shaft diameter = 3 in scaled from reference 7	in
μ	=	Bearing coefficient of friction = 0.25 or 0.6 from reference 4	none
ΔP	=	Differential pressure	psid

In any case, this is a small amount in comparison to the other unknowns.

7.0 Conclusions and Recommendations

7.1 Conclusions Summary:

DE&S concludes that reasonable assurance exists that the fluidynamic forces will close the subject valves at RG&E under the flow rates provided in Attachment B. This conclusion is based on alternate engineering assessments and not conclusive analysis or test data. DE&S could not locate directly applicable test or research data.

If a more definitive and conclusive analysis is desired, DE&S recommends testing to substantiate the analytical results and confirm that this check valve will close. DE&S recommends that a 3D CFD analysis is not necessary but could be helpful in understanding the phenomenon at the full open position. However, this type of analysis will still require testing to baseline and conform the model. Therefore, we still recommend testing alone. DE&S also recommends rework and/or repair of the valves to reduce the amount of the parasitic required breakaway torque.

7.2 Recommendations and Discussion:

These approaches can be validated by appropriate model testing only. It is recommended that a hydraulically similar model valve (1/4 scale or larger) be tested to determine what actual results are and validate an analytical model with correct coefficient data. This will be of great interest to the industry due to the void in our available knowledge base. It is not recommended that a 3D CFD model be developed, as this is expensive and still requires validation testing and model conformance. Additionally, once testing is performed its further value will be specific and limited to this analysis only.

While these analyses show that the subject valve will close on the minimum reverse flow of greater than 774.8 lbm/sec, it is our recommendation that the valve be reworked. In our experience, the combined frictionally induced torque of approximately 1200 ft-lb is too high for this size valve and shaft. The valve shaft packing and bearings should be checked, cleaned adjusted or replaced to lower the amount of parasitic torque loss.

Once the bearing and packing are properly cleaned and adjusted the counterweight can be adjusted to where it just balances the disc slightly prior to hitting the full open stop. While it is important to keep the valve disc against the full open stop during normal operation, it is also important that it should not restrict initial reverse flow closure. This may be accomplished by rotating the counterweight arm downward when the disc is full open. A separate counterweight balance and adjustment procedure can be developed to optimize counter weight torque once operation is restored to like new conditions.

8.0 References

1. E.B. Pool, et al., "Prediction of Surge Pressure from Check Valves for Nuclear Loops," ASME Paper 62-WA-219, October 1962

2. R.S. Kane and S.M. Cho, "Hydraulic Performance of Tilting-Disk Check Valves," Journal of the Hydraulics Division, ASCE, Volume HY1, January 1976, pp. 57-72

3. RG&E DA-ME-92-147, Rev. 2, Main Steam Non Return Check Valve Closure Analysis

4. EPRI "Application Guide for Motor Operated Valves in Nuclear Power Plants, Volume 2: Butterfly Valves", TR-106563-V2

5. "Marks' Standard Handbook for Mechanical Engineers", 10th Edition, Mc-Graw-Hill, 1996

6. Crane Technical Paper No. 410, 16th printing, 1976

7. Atwood & Morrill vendor scaled drawing 20729-H dated 6/30/67

8. RG&E Design Analysis DA-NS-99-054, Rev. 1, Main Steam Non-Return Check Valve Flow During a Small Steam Line Break

Attachment A: Reverse Flow Resistance Factor Fax from Atwood & Morrill

1 Page

ATWOOD & MORRILL CO., INC.

285 CANAL STREET SALEM, MA 01970
TEL: (978) 744-5690 FAX: (978) 744-9385

FAX DATA SHEET

DATE: 9/9/99 # OF PAGES TO FOLLOW: 0

TO: Jeu-Sheng Hsieh

COMPANY: Duke Engineering & Services

FAX NO.: 978-568-3704

FROM: Art Weldon

The K Factor in The reverse direction is 0.8

This figure was extrapolated from Flow Test Data on values from 14"-20" in The Forward direction.

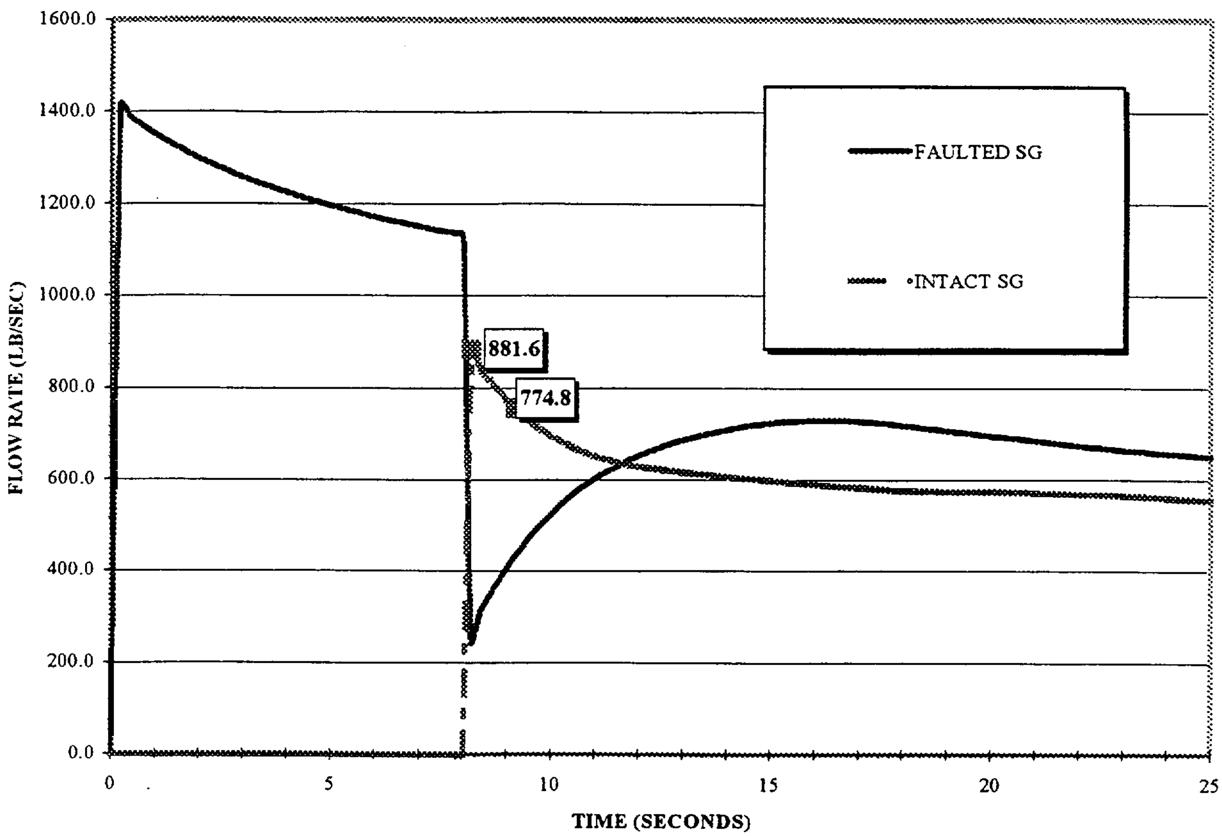
Best Regards

Art Weldon

Attachment B: Figure 1 - Break Flow Distribution From RG&E Calculation (reference 8)

1 Page

FIGURE 1 - BREAK FLOW DISTRIBUTION



Attachment C: Butterfly Valve Torque Coefficient List

1 Page

Low Pressure Butterfly Valve 75 degree Torque Coefficients

	CT
Average	0.233
Minimum	0.163

	CT			CT			CT			CT
Valve 1	0.191		Valve 14	0.237		Valve 27	0.278		Valve 40	0.236
Valve 2	0.218		Valve 15	0.261		Valve 28	0.278		Valve 41	0.236
Valve 3	0.225		Valve 16	0.228		Valve 29	0.163		Valve 42	0.236
Valve 4	0.199		Valve 17	0.173		Valve 30	0.163		Valve 43	0.229
Valve 5	0.246		Valve 18	0.272		Valve 31	0.163		Valve 44	0.194
Valve 6	0.246		Valve 19	0.288		Valve 32	0.163		Valve 45	0.264
Valve 7	0.191		Valve 20	0.255		Valve 33	0.163		Valve 46	0.174
Valve 8	0.219		Valve 21	0.264		Valve 34	0.163		Valve 47	0.292
Valve 9	0.250		Valve 22	0.264		Valve 35	0.174		Valve 48	0.229
Valve 10	0.284		Valve 23	0.282		Valve 36	0.236		Valve 49	0.292
Valve 11	0.259		Valve 24	0.278		Valve 37	0.236		Valve 50	0.255
Valve 12	0.259		Valve 25	0.278		Valve 38	0.236		Valve 51	0.206
Valve 13	0.259		Valve 26	0.278		Valve 39	0.236		Valve 52	0.215