Vogtle Electric Generating Plant

Generic Letter 89-10
Close-Out Submittal

Volume 1

Program Summary
Report

January 1996

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1.0 INTRODUCTION

VEGP has implemented a program to address the recommendations contained in Generic Letter 89-10, "Safety Related Motor-Operated Valve Testing and Surveillance", and the design-basis capability verification portion of the program is complete. The purpose of this document is to summarize the program and to present the justification for verifying the design-basis capability of each motor-operated valve (MOV) included within the scope of the program.

The flowchart outlined in Figure 1-1 depicts the major activities undertaken in conjunction with the design-basis capability verification portion of the VEGP GL 89-10 program. The VEGP GL 89-10 program is the product of an evolutionary process which continually refined the overall methodology utilized to design, test and maintain safety-related MOVs. The flowchart identifies the major activities associated with this process, and the relationship of these activities to one another as the program progressed.

When Generic Letter 89-10 was issued VEGP already had a program in place which utilized diagnostic test equipment to setup and test safety-related MOVs. This program was founded in conjunction with VEGP's response to I & E Bulletin 85-03 and had been expanded to include additional safety-related valves prior to the issuance of GL 89-10. The setpoint information utilized in this initial program was supplied primarily by Westinghouse and Bechtel and was based on the original plant design.

The initial activity performed in conjunction with VEGP's response to GL 89-10 was the performance of a detailed engineering design review. The initial phase of this review began with the identification of the scope of valves covered by GL 89-10 and concluded with the issuance of Setpoint Documents for Units 1 & 2 in November 1992 and March 1993 respectively. The engineering design review was a very complex activity and is discussed in detail later in this document. The design review was also a very dynamic process in that it continually evolved over the past six years as additional industry and VEGP specific data were factored into the program.

The initial design review utilized the best available methodology at that time. In many cases this methodology was significantly more conservative that that employed in the original plant design. Consequently, the initial design review results identified many instances in which the current plant configuration did not appear adequate based on this conservative methodology. As problems were identified, Operability Determinations were performed to ensure that the MOVs would be capable of performing their design-basis function until the individual MOV's could be setup and/or modified to an more acceptable configuration. In cases where this was not possible the appropriate actions were taken as delineated in the VEGP Technical Specifications. The practice of performing Operability

Determinations as problems were identified continued throughout the engineering design review process.

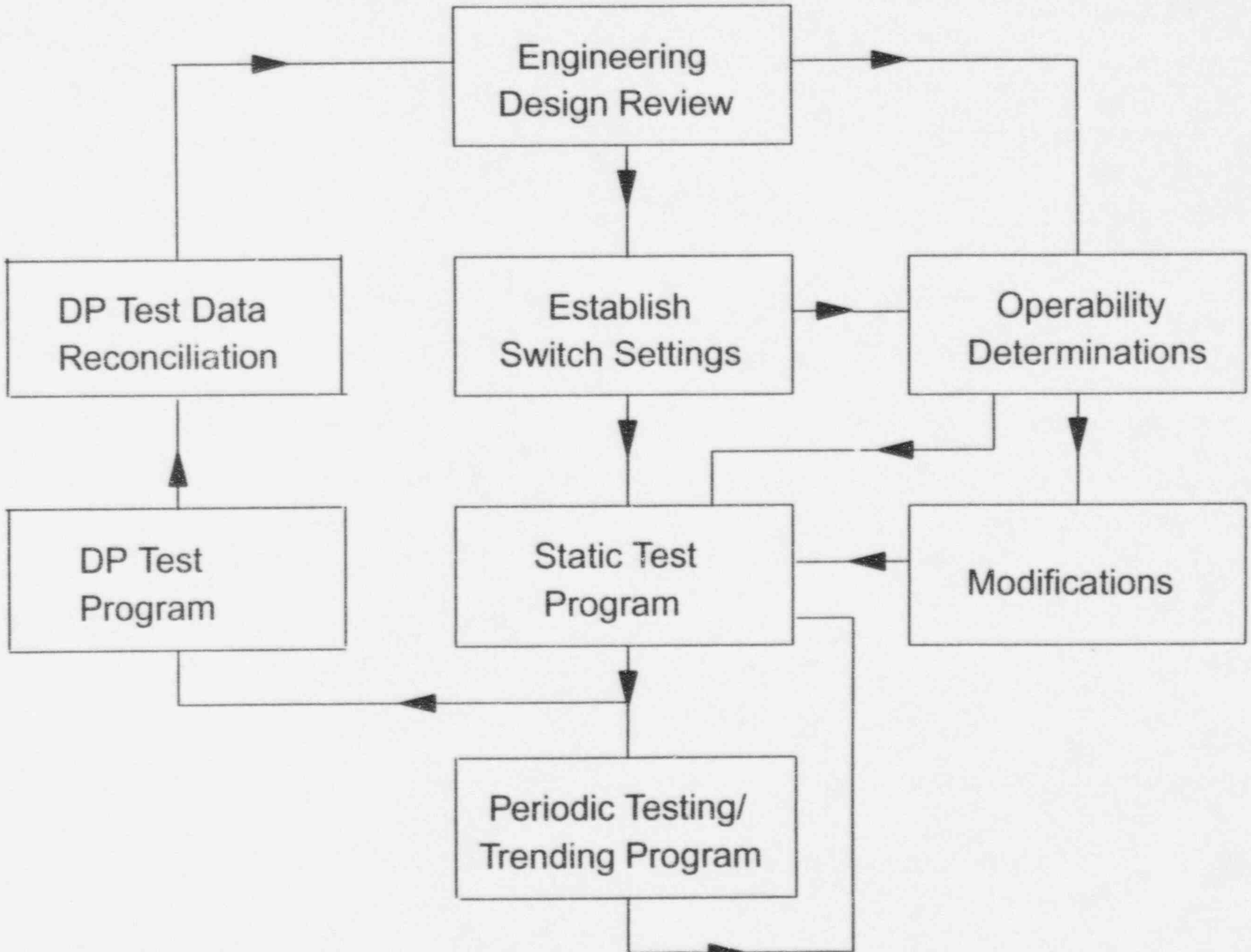
At the conclusion of the 1993 refueling outages all of the 256 MOV's included within the VEGP GL 89-10 program had been setup utilizing diagnostic equipment to operate within the revised windows as outlined in the Setpoint Documents. Included within this scope of work was the modification of 53 MOVs to provide additional margin as required to support the setup of these MOV's within the revised windows.

A differential pressure testing program was also initiated during the 1993 refueling outages. The objective of this program was to validate the methodology utilized in the engineering design review. The test scope was chosen based on a prioritization system which considered the valves design-basis operating requirements and the available margin. The differential pressure testing associated with this program and the subsequent evaluation of the test data was completed in May 1995.

As a result of the differential pressure test program, and the availability of additional industry data, a number of changes were incorporated into the engineering design review methodology. The incorporation of these additional factors into the design review necessitated the issuance of revised Setpoint Documents and the implementation of a final round of modifications to provide additional margin for certain MOV's. As has been the policy throughout the program, Operability Determinations have been established, as required, to support the continued use of valves in their current configuration until the required changes can be implemented.

The implementation of the final modifications will conclude the initial implementation of the VEGP GL 89-10 program. The program is entering the maintenance mode aimed at ensuring that the valves will remain operable for the life of the plant. The programs necessary to accomplish this objective are in place and are outlined in the site maintained "Motor-Operated Valve Program Manual".

Figure 1-1
VEGP GL 89-10 Program Flowchart



2.0 PROJECT ORGANIZATION

The VEGP Generic Letter 89-10 project is a matrix organization staffed by personnel from three Southern Company subsidiaries including Georgia Power Company (GPC), Southern Company Services (SCS) and Southern Nuclear Operating Company (SNC). Figure 2-1 outlines the major elements of the organization. The SNC Project Manager, the SCS Lead Engineer and the GPC Maintenance Engineer have been dedicated primarily to the GL 89-10 program. The remaining resources have been utilized on an intermittent basis to support the program as schedule and manpower requirements dictate.

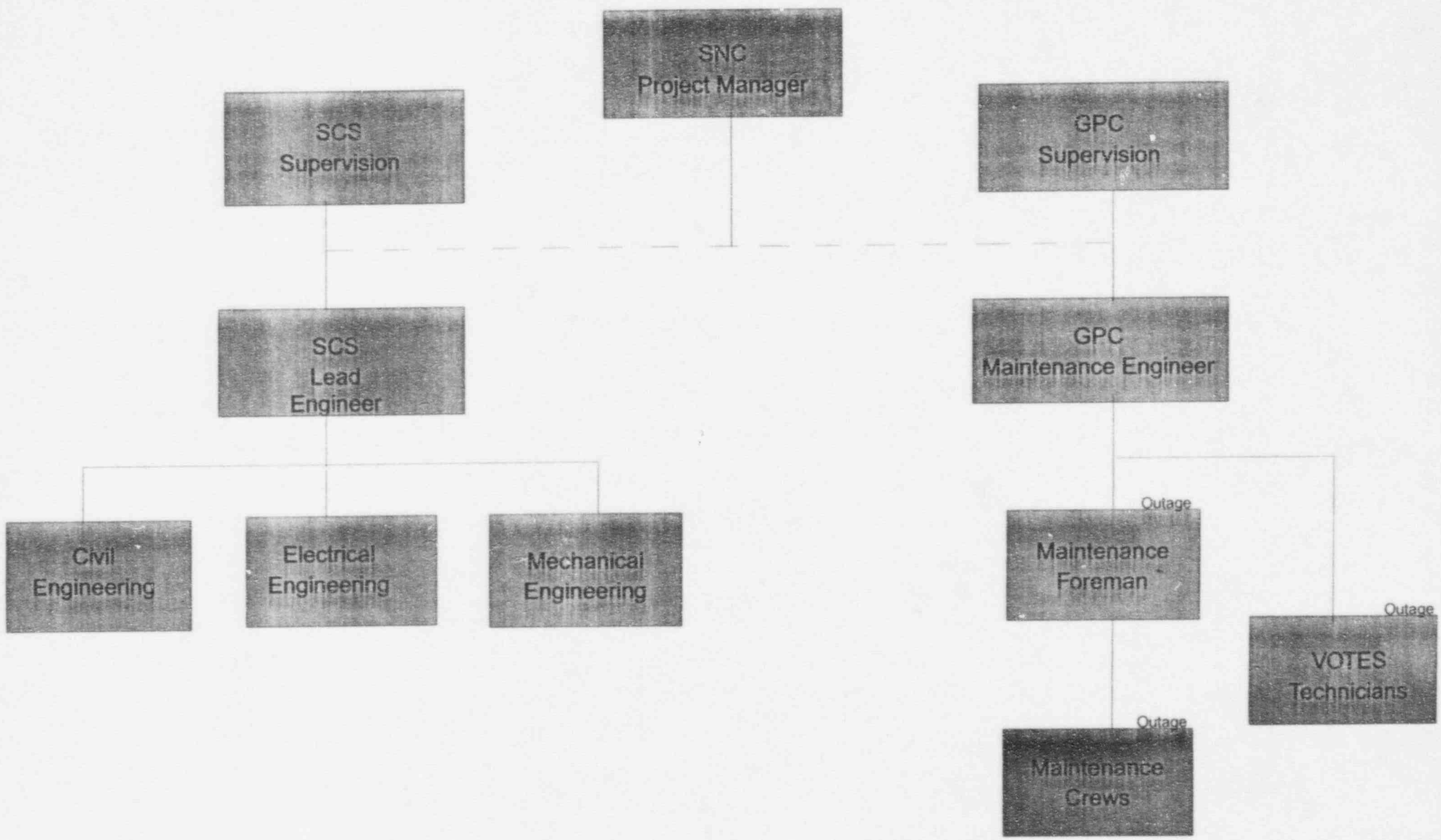
The SNC Project Manager has overall responsibility for the VEGP Generic Letter 89-10 Program. The Project Manager's duties include establishing and tracking a project scope, schedule and budget which satisfactorily addresses the recommendations contained in the generic letter while being consistent with the overall objectives of the Southern Nuclear Operating Company.

The SCS Lead Engineer is responsible for all off-site engineering activities undertaken to support the GL 89-10 Program. Major activities completed by SCS to date include the GL 89-10 design review, the issuance of Setpoint Documents for each unit and the development of Design Change Packages for MOV modifications. In addition, SCS performed the detailed evaluation of the differential pressure test data and reconciled the differential pressure test results with the design review calculations.

The GPC Maintenance Engineer is responsible for the on-site implementation of the GL 89-10 Program. Major activities completed to date include baseline testing of each valve included in the GL-89-10 Program, the implementation of modifications to 53 MOVs and the performance of the differential pressure testing program. The Maintenance Engineer is responsible for the implementation of the long term activities identified in the "Motor Operated Valve Program Manual".

Figure 2-1

VEGP GL 89-10 Project Organization



3.0 ENGINEERING DESIGN REVIEW

A comprehensive design review was performed by Southern Company Services (SCS) to evaluate each safety-related MOV at VEGP. The review documented in detail the installed configuration of each MOV and provided a measure of each valves capabilities utilizing the best available analytical techniques. The methodology incorporated in this review was developed as the study progressed and was refined substantially as additional plant specific and industry wide information became available.

3.1 MOV Scope

An extensive documentation review was performed to identify all of the safety-related MOVs at VEGP. As a result of this review a total of 340 motor-operated devices were originally identified for inclusion in the program. When Supplement 1 to the generic letter was issued the requirement to include safety-related dampers was deleted and this reduced the VEGP scope by a total of 60 to 280 safety-related MOVs. As the design review progressed further it was determined that 24 of the MOVs did not perform an active safety function and these MOVs were also excluded from the overall scope. A total of 256 valves were ultimately identified for inclusion in the scope of the VEGP GL 89-10 program.

3.2 Design-Basis Review

A design-basis review was performed to determine the operating requirements under both normal and abnormal conditions, including mispositioning, for each MOV included in the GL 89-10 program . The scenarios under which the MOVs would be required to operate were identified and calculations were performed to determine the line pressure and differential pressure each MOV would be required tc operate against to perform the various design functions. The methodology contained in Westinghouse WCAP-13907 entitled "System Operating Basis for Motor-Operated Valves" was utilized extensively in performing this review. The results of this review were subsequently utilized in determining the torque or thrust requirements for each valve.

A detailed explanation of the methodology utilized in performing this review, the various functions of each MOV and the resulting line pressure and differential pressure for each function are outlined in Calculation Number X4C1000U01. Revision 0 of this calculation was approved on November 14, 1991. The current revision of this calculation is Revision 11 dated January 19, 1996.

3.3 Valve/Operator Calculations

The original valve and operator calculations were performed by the valve and/or operator manufacturers. The calculations were performed utilizing each manufacturer's empirical methodology which typically varied from vendor to vendor. In conjunction with the VEGP engineering design review these calculations were updated utilizing the best available methodology. The methodology utilized to evaluate and size MOVs evolved substantially as additional industry and plant specific data became available. The methodology currently utilized in the calculations represents the best available for each valve type based on both industry and VEGP specific data.

3.3.1 Valve Required Torque/Thrust Calculation

The thrust or torque required to operate a valve is a function of the physical characteristics of the valve and the design-basis conditions under which the valve must function. The design-basis conditions for each valve were determined as outlined in Section 3.2. The effects that the physical characteristics of the valve have on the torque or thrust requirements necessary to operate that valve are more difficult to assess and are typically determined based on an empirically derived methodology. This methodology varies for the different valve types such as gate, globe and butterfly valves and in some cases from vendor to vendor within a given valve type. The valves utilized at VEGP fall into the following five basic valve groups:

1. Anchor-Darling Gate Valves
2. Westinghouse Gate Valves
3. Fisher Globe Valves
4. Fisher Butterfly Valves
5. Velan Globe Valves

The available methodologies were reviewed for each of the valve groups to ensure that the most appropriate methodology was selected for evaluating each group.

3.3.1.1 Anchor-Darling Gate Valves

The original thrust requirement calculations for the Anchor-Darling gate valves were performed utilizing the industry standard "Limitorque Equation" with a 0.3 valve factor. Industry testing suggested that a valve factor of 0.3 may not conservatively predict thrust requirements for certain gate valves. In order to ensure a conservative calculation for the VEGP valves the decision was made to utilize the Limitorque Equation with a 0.5 valve factor in the original VEGP GL 89-10 design review calculations.

The opening and closing thrust requirements for the Anchor-Darling gate valves were initially calculated utilizing the following equations:

$$T_o = VF \times DP_o \times A_d - LP_c \times A_s + PL$$

$$T_c = VF \times DP_c \times A_d + LP_o \times A_s + PL$$

Where:
T_o = Required Opening Thrust
T_c = Required Closing Thrust
VF = Valve Factor
DP_o = Differential Pressure Opening
DP_c = Differential Pressure Closing
A_d = Disk Area
A_s = Stem Area
LP_o = Line Pressure Opening
LP_c = Line Pressure Closing
PL = Packing Load

This methodology, utilizing a 0.5 valve factor, yielded a much more conservative result than the original design calculations. Differential pressure tests were performed on two valves in this group and the thrust requirements measured in these tests were bounded by this methodology. However, since only two of the twenty valves of this type were testable under significant differential pressure, the decision was made to re-evaluate the valves utilizing the EPRI Performance Prediction (PPP) methodology. The EPRI methodology has been verified based on extensive differential pressure test data, including Anchor-Darling valves, therefore, thrust requirements developed utilizing this methodology have a defensible basis.

The EPRI methodology was not applied to the 1/2HV-9380A/B valves since these valves are located in air systems and this fluid is not covered by the EPRI methodology. The thrust requirements for these valves will continue to be based on the Industry Standard methodology outlined above.

An additional consideration with regard to applying the EPRI methodology to the Anchor-Darling valves involves an issue regarding valve guide to guide slot clearances. The EPRI methodology stipulates that valves with carbon steel guides in blowdown applications must maintain a minimum guide clearance of 1/16" in order to be considered predictable. The concern is that the relatively high stresses applied to the guides in blowdown applications may cause galling at the carbon steel to carbon steel interface. The galling itself is considered to be predictable and is bounded by the friction coefficients utilized by the model. However, to remain predictable sufficient clearance must be maintained to allow the galling products to clear the guide slot. The Anchor-Darling valves employ carbon steel guides and guide slots and a number of the valves are utilized in blowdown applications, therefore, this criteria is applicable to those valves. An evaluation of the guide clearances for these valves, based on worst-case design dimensions and manufacturing tolerances, indicated that valves 1/2HV-2041,1/2HV-3009 and 1/2HV-3019 may not have a 1/16" guide clearance. These valves will be inspected during each units next refueling outage to ensure that the minimum guide clearance required by the EPRI methodology is available.

The EPRI gate valve methodology utilizes the following basic equation to calculate gate valve thrust requirements:

$$F_R = F_W + F_P + F_{SR} + F_{DP} + F_T$$

Where:
F_R = Required stem thrust
F_W = Gravity load of disk and stem
F_P = Packing friction force
F_{SR} = Stem rejection force
F_{DP} = Disk differential pressure force
F_T = Torque reaction friction force

The terms are relatively self explanatory, however, the actual determination of the disk differential force, which is typically the dominant term, requires an extremely complex algorithm to account for the myriad of mechanical interactions associated with the valve internals. The overall population of gate valves incorporate a variety of design features, each with their own specific operating characteristics. To address the various operational modes made possible by these design differences requires a flexible and sophisticated model.

The EPRI PPP gate valve methodology incorporates a complex computer-based model to evaluate the thrust requirements as a function of valve position. A system model is developed for each valve which allows the differential pressure acting on the valve disk to be calculated as a function of valve position and ultimately converted to a disk load. The disc load is input into a kinematic model which evaluates the relative interaction of the valve stem, disc, guides and seat. This model requires various valve dimensional data which is taken from the original design drawings. Where applicable, dimensions are adjusted based on design tolerances, to provide a worst-case dimension with respect to valve performance. The model evaluates the valve stroke from a kinematic standpoint and determines if the valve will operate in a predictable manner under the various disc loading conditions experienced over the course of a valve stroke. Thrust requirements were calculated utilizing the default friction coefficients included in the software. These friction coefficients were developed by EPRI through separate effects testing and represent the bounding values for the material combinations, contact loads and fluid conditions applicable to these valves.

A detailed discussion of this methodology is beyond the scope of this document. An in-depth description of the methodology is contained in the EPRI MOV Performance Prediction Program "Gate Valve Model Report" document number EPRI TR-103229.

3.3.1.2 Westinghouse Gate Valves

As a result of issues raised following Three Mile Island, Westinghouse implemented an in-house test program to evaluate the thrust requirements of the Westinghouse EMD gate valve. This program involved testing a number of valves under differential pressure conditions and measuring the required thrust utilizing strain gauge based instrumentation. Westinghouse subsequently developed a methodology for predicting thrust requirements for Westinghouse gate valves based on the results of this test program. The test program did not address all sizes and variations of the Westinghouse gate valve design and in cases where valve specific test data was not available the existing test data was extrapolated to cover additional designs and sizes. The Westinghouse methodology is based on a variation of the Limitorque Equation and utilizes disc coefficients derived from the Westinghouse test program. The Westinghouse developed equation and disc coefficients were utilized in the original VEGP design review to calculate the thrust requirements for the Westinghouse EMD valves.

The opening and closing thrust requirements for the Westinghouse gate valves were initially calculated utilizing the following equations:

$$T_o = DC_o \times (D_1 + 0.0625)^2 \times DP_o - 0.7854 \times D_2^2 \times LP_o + PL$$

$$T_c = DC_c \times (D_1 + 0.0625)^2 \times DP_c + 0.7854 \times D_2^2 \times LP_c + PL$$

Where:

T_o = Required Opening Thrust
T_c = Required Closing Thrust
DC_o = Disk Coefficient Opening
DC_c = Disk Coefficient Closing
D_1 = Seat Ring Bore
D_2 = Stem Diameter
DP_o = Differential Pressure Opening
DP_c = Differential Pressure Closing
LP_o = Lower of Opening LP/DP

LP_c = Higher of Closing LP/DP
PL = Packing Load

The disk coefficients utilized in the above equations were provided by Westinghouse and are valve group specific. It should also be noted that the disk coefficients utilized in the Westinghouse equations are not directly comparable to the valve factors utilized in the Limitorque equation.

Table 3-1
Westinghouse Gate Valve
Disk Coefficients

Valve Group	Valve Size (inch)	ANSI Rating (lb)	Open Disk Coefficient	Close Disk Coefficient
W-1	3.0	150	0.380	0.447
W-2A	3.0	2035	0.431	0.479
W-2B	3.0	2035	0.365	0.443
W-3	4.0	150	0.381	0.448
W-4	4.0	900	0.350	0.394
W-5	4.0	1525	0.350	0.394
W-6	6.0	150	0.368	0.433
W-7	8.0	150	0.361	0.424
W-8	8.0	316	0.361	0.424
W-9	8.0	1525	0.370	0.436
W-10	10.0	150	0.360	0.424
W-11	12.0	316	0.353	0.415
W-12	12.0	1525	0.377	0.443
W-13	14	316	0.353	0.415

A substantial amount of differential pressure testing was performed at VEGP to support the use of this methodology. While the majority of the test data supported the Westinghouse methodology a number of issues arose relative to its continued use as the methodology of choice for predicting Westinghouse gate valve thrust requirements.

As can be seen by reviewing Table 3-1, the Westinghouse methodology employs many different disk coefficients depending on the valve size and pressure class. In many cases the differences in disk coefficients are relatively small and result in only minor differences in calculated thrust requirements. With respect to the VEGP in-situ differential pressure results, it became apparent that the relatively small differences in disk coefficients were not quantifiable based on the test results, given the overall accuracy of the test process. The multiple disk coefficients were unnecessarily complicating the process of calculating thrust requirements for these valves without contributing noticeable to the overall accuracy of the methodology.

In addition, a number of the Westinghouse valves groups do not include any valves which can be differential pressure tested in-situ at differential pressures approaching design-basis conditions. Utilizing different disk coefficients for each group of valves makes it difficult to utilize test data from valve groups with testable valves to support the methodology utilized for groups which do not have testable valves.

Due to the issues outlined above the decision was made to standardize the design review methodology for the Westinghouse gate valves. The methodology developed in conjunction with the EPRI PPP to address the Westinghouse gate valves was not available at this time, therefore, this methodology was not considered. After reviewing the available methodologies the EPRI NMAC equation was selected for use in performing the Westinghouse gate valve calculations. This methodology is discussed in detail in the NMAC "Application Guide for Motor-Operated Valves in Nuclear Power Plants". This methodology was also utilized by EPRI in conjunction with the evaluation of differential pressure test data associated with the Performance Prediction Program.

The basic EPRI NMAC equation for calculating gate valve thrust requirements is as follows:

$$F_R = \left(F_W + F_P + F_{SR} + F_{DP} + F_S\right)/TRF$$

Where:

F_R = Required stem thrust
F_W = Gravity load of disk and stem
F_P = Packing friction force
F_{SR} = Stem rejection force
F_{DP} = Disk differential pressure force
F_S = Sealing Load
TRF = Torque Reaction Factor

In evaluating the various terms included in this equation, it was determined that the weight of the disk and stem, F_W, and the torque reaction factor, TRF have a negligible effect on the thrust requirements for Westinghouse EMD gate valves. Therefore, these terms were deleted from the equation. In addition, the sealing load, F_S, was also deleted because the objective of this calculation is to determine the thrust required to reach flow cutoff under differential pressure conditions.

Deleting these terms, the NMAC equations for calculating the opening and closing thrusts for gate valves are as follows:

$$T_o = PL - LP_o \times A_s + (\mu \times DP_o \times A_d)/(\cos\theta + \mu \times \sin\theta)$$

$$T_c = PL + LP_c \times A_s + (\mu \times DP_c \times A_d)/(\cos\theta - \mu \times \sin\theta)$$

Where:
T_o = Required Opening Thrust
T_c = Required Closing Thrust
μ = Friction Coefficient
DP_o = Differential Pressure Opening
DP_c = Differential Pressure Closing
A_s = Stem Area
A_d = Disk Area
LP_o = Line Pressure Opening
LP_c = Line Pressure Closing
PL = Packing Load
θ = Seat Angle

The friction coefficients utilized to evaluate the Westinghouse valves were 0.55 for valves in steam service and 0.60 for valves in water service. These values were taken from the EPRI Performance Prediction Program "Gate Valve Model Report Volume 1" Table 2-3 which outlines friction coefficients for Stellite on Stellite contact.

3.3.1.3 Fisher Globe Valves

The required thrust values for the Fisher globe valves were determined by Fisher utilizing proprietary methodology. The Fisher methodology was developed based on a vast amount of practical experience and is specific to the Fisher family of globe valves. The differential pressure testing performed at VEGP supported this methodology, therefore, there did not appear to be any basis for considering one of the generic methodologies to address these valves.

VEGP utilizes two basic types of Fisher globe valves and each valve type is addressed independently by the Fisher methodology. The unbalanced disk valves are similar in many respects to other conventional globe valves and the thrust requirements for these valves vary as a function of differential pressure. The balanced disk globe valves utilize pilot plugs which equalize the pressure across the main plug thereby making the thrust requirements for these valves essentially independent of differential pressure.

The thrust requirements for the Fisher globe valves were not recalculated following the development of the differential pressure calculation. The required thrusts for the Fisher globe valves continue to be based on the differential pressures utilized in the original plant design which in all cases are higher than the differential pressures contained in the current revision of the differential pressure calculation.

3.3.1.4 Velan Globe Valves

The original thrust requirements for the Velan globe valves were calculated utilizing an industry standard "Limitorque Equation" similar to that which was used to perform the original Anchor-Darling calculations. In the case of the globe valves a valve factor of 1.1 was utilized in this equation. In addition, the stem rejection load was ignored in the opening direction for conservatism and was only utilized in the closing direction when the differential pressure was equal to zero.

A significant scope of Velan globe valves were differential pressure tested and an evaluation of the test results indicated that the actual closing thrust for a number of the valves was not bounded by the this methodology. Therefore, the decision was made to re-evaluate this group of valves utilizing the EPRI PPP methodology for globe valves.

The EPRI globe valve model is relatively simple in comparison to the EPRI gate valve model. Since valve predictability is not an issue with globe valves the valves performance does not need to be evaluated through out the entire stroke. Thrust requirements are only calculated at seating and unseating and these are bounding values for the valve stroke. Since the valve is only evaluated at this point it is not necessary to develop a system model to calculate the differential pressure as a function of valve position.

The EPRI PPP methodology utilizes the following basic equation to calculate thrust requirements for globe valves:

$$F_R = F_{DP} + F_{DF} + F_{SR} + F_G + F_{PF} + F_{TR}$$

Where: F_R = Resultant stem force
F_{DP} = Disc differential pressure force
F_{DF} = Friction force between disk guides and valve body
F_{SR} = Stem rejection load
F_G = Gravity load of disc and stem
F_{PF} = Packing Friction Force

F_{TR} = Torque reaction friction force

The Velan globe valves are unbalanced disk valves, therefore, the term F_{DF} can be deleted from the equation since it only applies to balanced disk valves. In addition, the weight of the disc and stem are insignificant in relation to the differential pressure loads for these valves so the term F_G may also be deleted from the equation. Finally, the torque reaction friction force, F_{TR}, is a function of the resultant stem force and may be represented by the dimensionless torque reaction factor (TRF) which is defined as follows:

$$TRF = 1 - (\mu_t \times FS)/r_t$$

Where:
- μ_T = Coefficient of Friction at torque reaction surface
- FS = Stem Factor
- r_t = Mean radial distance from center of stem to torque reaction surface

Deleting the above referenced terms and substituting the torque reaction factor, the EPRI globe valve equation simplifies to the following:

$$F_R = (F_{DP} + F_{SR} + F_{PF})/TRF$$

It should be noted that this equation does not employ a valve factor. Rather, the equation addresses each of the relevant forces which act upon the valve stem at the point of seating and unseating. The methodology is suitable for evaluating both seat-based and guide-based valves as well as applications with flow over or under the seat. The VEGP Velan valves are seat-based and are utilized in flow over the seat and flow under the seat applications. The methodology is described in more detail in the EPRI MOV Performance Prediction Program "Globe Valve Model Report" document number EPRI TR-103227.

3.3.1.5 Fisher Butterfly Valves

The torque requirements for the Fisher butterfly valves were originally calculated by Fisher utilizing a proprietary methodology. The Fisher methodology was developed based on a vast amount of practical experience and is specific to the Fisher family of butterfly valves. Due to variations in the design-basis differential pressures, as a result of the design review, the torque requirements were re-calculated for some valves to reflect the revised differential pressures. The revised torque requirements for these valves were determined utilizing the original Fisher methodology.

Initial differential pressure testing indicated that the Fisher methodology provided a good prediction of the dynamic loads required to open and close the valves, however, it also indicate that the static loads associated with the valves packing were much higher than the loads utilized by Fisher. As a result of this apparent anomaly additional static testing was performed to evaluate packing loads. The results of the static testing confirmed that the actual packing loads were substantially higher than those included in the original Fisher calculations.

To address this problem a standardized packing configuration has been developed for butterfly valves which utilizes a three ring graphite packing set on each end of the valve stem. To determine the torque required to rotate the stem through this packing configuration, calculations were performed utilizing the following equation taken from the EPRI NMAC "Application Guide for Motor-Operated Butterfly Valves in Nuclear Power Plants":

$$PT = \pi/48(\mu_P \times GS \times L_P)$$

Where:
PT = Packing Torque
μ_P = Packing Coefficient of Friction
GS = Gland Stress
L_P = Packing Length

Utilizing this equation with a gland stress of 3000 psi, a coefficient of friction of 0.2 and three rings of packing on each end of the valve stem results in the following calculated packing loads for the VEGP butterfly valves:

Table 3-2
Butterfly Valve Packing Loads

Valve Size (inch)	Stem Diameter (inch)	Packing Load (ft-lb)
4	0.625	17
8	1.0	44
10	1.25	69
18	2.0	235
24	2.0	235
24	2.5	368

The packing loads predicted by the EPRI methodology for this packing configuration are substantially higher than those utilized in the original Fisher calculations. To evaluate the EPRI packing equation a limited scope of valves were repacked utilizing the standardized packing configuration and measurements were made of the running loads for each valve. The limited amount of testing performed indicated that the EPRI equation provided a reasonable prediction of the loads associated with this packing configuration. Therefore, the torque requirements for the Fisher butterfly valves were recalculated incorporating the EPRI packing loads while continuing to utilize the Fisher methodology to predict the dynamic torque requirements.

Based on the static testing performed to date, the packing loads associated with the standardized packing configuration are typically less than the current packing loads for the 4, 8 and 10 inch valves. The current packing loads for the 18 and 24 inch valves appear to be bounded by the standard packing configuration loads. Packing loads for the GL 89-10 butterfly valves will continue to be measured and evaluated in conjunction with the scheduled periodic test program utilizing the stem mounted strain gauge based sensor. Valves will be repacked as required, utilizing the standardized packing configuration, based on measured packing loads and available margins. Data regarding packing loads will be forwarded to engineering for review to ensure that the revised packing loads being utilized in the design calculations are conservative. If the additional data does not support the continued use of these packing loads, then the butterfly valve calculations will be revised to incorporate packing loads which are supported by VEGP test data.

In the interim, the 4, 8 and 10 inch valves were evaluated utilizing a bounding packing load of ten times the values utilized by Fisher. This was done to ensure that the valves would be capable of performing their design-basis function until such time as they could be repacked with the revised packing configuration. The results of this evaluation are discussed in Section 5 of this document.

3.3.2 Operator Capability Calculations

All of the motor-operators installed on safety-related valves at VEGP were manufactured by Limitorque Corporation. The capability of the individual operators is basically a function of the applied voltage, motor starting torque, the overall gear ratio of the operator and several empirical factors related to the efficiency of the gear train. In conjunction with the VEGP design review the operator capabilities were re-calculated utilizing the current Limitorque methodology.

3.3.2.1 Limitorque Selection Procedure

Limitorque publishes a selection procedure which is considered a standard for determining the capability of Limitorque operators. The basic methodology outlined in this procedure was utilized to perform the VEGP operator calculations. The only significant exception taken to this procedure was with regard to the determination of the closing capability of operators equipped with A. C. motors.

In the closing capability calculation for A. C. motor equipped operators the running efficiency was substituted for the pullout efficiency. In the closing direction the operator is not required to come up to speed under a load, therefore, utilizing the pullout efficiency in this application would yield an overly conservative result.

3.3.2.2 Degraded Voltage Calculations

The voltage available at each MOV included in the GL 89-10 Program was calculated for use in the operator capability calculations. The A. C. voltage calculations were performed based on the assumption that the 4160 volt busses were operating at the minimum setpoint of the under-voltage relays. The dynamic response of the bus voltages and corresponding MOV voltages were then modeled during a simulated LOCA scenario utilizing the STAUX and START computer codes developed by Southern Company Services. The D. C. voltage calculations were performed based on a loss of offsite power scenario assuming the battery was at the end of its discharge cycle. The assumptions utilized in performing the voltage calculations results in extremely conservative estimates of MOV terminal voltages. The Unit 1 A. C. voltage, Unit 2 A. C. voltage and the Units 1 and 2 D. C. voltage calculations are included in Calculations X3CA19, X3CA20 and X3CK08 respectively.

3.3.2.3 Temperature Effects on Reliance A. C. Motors

Limitorque issued a Part 21 Notification concerning the effects of elevated temperature on the starting torque capability of Reliance motors supplied on Limitorque operators. The notification basically indicated that the starting torque of A. C. motors decreased as the ambient temperature of the motor increased. Limitorque subsequently issued Limitorque Technical Update 93-03 in September 1993 which further elaborated on the problem and also revised the methodology utilized by Limitorque to determine operator capabilities. The VEGP GL 89-10 valves have been evaluated relative to this guidance and the starting torque for each motor has been adjusted in accordance with the design-basis ambient temperature for the respective MOV.

3.3.2.4 Kalsi Report

Kalsi Engineering has implemented a number of test programs and studies to evaluate the torque and thrust carrying capabilities of various Limitorque operators. As a result of this work, Kalsi has been successful in increasing the thrust capacity of the Limitorque SB and SMB 000, 00, 0, and 1 operators to 140% of the original Limitorque rating. Limitorque has reviewed this report and approved the new ratings. The revised ratings have been incorporated into the VEGP design review.

3.3.3 Additional Factors Effecting Valve/Operator Calculations

In performing an analytical evaluation of rising stem valves it is necessary to convert the rotational torque produced by the operator to linear thrust to operate the valve. This conversion is handled analytically by a term referred to as the stem factor. There are several issues associated with this term and its use in MOV calculations.

3.3.3.1 Stem Friction Coefficient

To operate a rising stem valve the rotational torque output of the motor-operator must be converted to linear thrust. From a practical standpoint this conversion is performed by the interaction of the valve stem and the operator stem nut. The stem nut rotates which causes the non-rotating stem to move up or down depending upon the direction of rotation of the stem nut. From an analytical standpoint the conversion is handled by a term referred to as the stem factor. Dividing the operator output torque by the stem factor results in the thrust available to drive the valve. The stem factor is a function of the physical geometry of the stem and stem nut and an empirical term referred to as the stem friction coefficient. The friction coefficient is a measure of the efficiency of the torque to thrust conversion and is influenced by the physical condition of the stem and stem nut as well as the efficiency of the valve stem lubricant.

There has been a considerable amount of controversy concerning the appropriate value to utilize for the stem friction coefficient when performing calculations in support of Generic Letter 89-10. Assuming too low a value can result in non-conservative estimates of operator thrust capability. Conversely, assuming too high a value can result in non-conservative evaluations of valve and operator structural adequacy. A stem friction coefficient of 0.15 is generally considered a nominal value for nuclear plant applications, and this value was selected for use in the original VEGP GL 89-10 calculations.

In conjunction with the differential pressure test program implemented at VEGP, data was taken to evaluate the static and dynamic stem factors of the valves which were tested. This test data was subsequently evaluated to determine the actual stem friction coefficients and the results of this evaluation are outlined in Section 9.6 of this document. Based on the results of the testing it was concluded that the stem friction coefficients for the Anchor-Darling valves, which utilize stub ACME threads, should be increased to 0.20. The remaining rising stem valves, which are equipped with standard ACME threads, will continue to utilize a 0.15 stem friction coefficient.

3.3.3.2 Load Sensitive Behavior

Load Sensitive Behavior, or the rate-of-loading effect, as it is sometimes called, refers to a phenomenon in which the thrust at torque switch trip is higher under static conditions than under dynamic conditions. It was initially thought that the torque switch operated earlier under differential pressure conditions thereby limiting operator output torque. This theory has been abandoned however, and the phenomenon is now attributed to changes in the torque to thrust conversion under differential pressure conditions. The cause of the phenomenon is not well understood but it is believed to be related to characteristics of the stem, stem nut and stem lubricant.

VEGP did not explicitly address Load Sensitive Behavior in the original design review calculations. A suitable methodology was not available at the time and VEGP's intent was to revise the calculations as soon as a suitable methodology became available. In the interim, VEGP's policy was to set valves up as high in the design window as practical to ensure that adequate margin would be available to compensate for the effect should it occur.

In conjunction with the VEGP differential pressure test program data was taken to evaluate Load Sensitive Behavior. This test data was subsequently evaluated and the results of this evaluation are outlined in Section 4.2.3 of this document. The evaluation concluded that Load Sensitive Behavior was a randomly occurring phenomenon and that it was best addressed in combination with the adjustment of switch settings to account for torque switch repeatability and test equipment accuracy. Therefore, Load Sensitive Behavior is not explicitly addressed in the design review calculations.

3.3.4 MOV Torque/Thrust Calculation

The culmination of the valve and operator calculations was the issuance of Calculation X4C1000U02 Revision 0 on May 29, 1992. This calculation, entitled " Valve Required Thrust/Torque and Operator Capabilities and Limitations for the GL 89-10 Scope MOVs" outlines in detail the methodology utilized to perform the VEGP design review. This calculation will be maintained as a living document and is currently in Revision 9 dated January 26, 1996.

4.0 ESTABLISHMENT OF SWITCH SETTINGS

The engineering design review determined a minimum required and maximum available thrust and/or torque value for each MOV included in the GL 89-10 program. In order to ensure that each MOV will be capable of performing its design-basis function, the MOV must be set-up to operate within this minimum required and maximum available range. Inherent in the process of setting-up and verifying that each MOV is operating within this design range are a number of factors which must be considered in conjunction with the establishment of final switch settings.

4.1 Determination of Minimum and Maximum Torque/Thrust Requirements

The minimum required and maximum available torque and/or thrust for each MOV was determined in conjunction with the engineering design review which was discussed in detail in Section 3.0. The design review information necessary to set-up each valve was consolidated into a Setpoint Document which was transmitted to the site in the form of a design change package (DCP). The Setpoint Documents will be discussed in more detail in Section 6.0 of this document. The Setpoint Documents contain the design range for each valve and this range becomes the basis for establishing the actual switch settings in the field.

4.2 Sources of Uncertainty

Once the design range has been determined, each valve must be set-up and verified to be operating within that range. There are a number of uncertainties associated with the set-up and verification process which must be quantitatively addressed to ensure that the valve is, in fact, set-up within its design range. The primary uncertainties which must be addressed are torque switch repeatability, test equipment accuracy and Load Sensitive Behavior.

4.2.1 Torque Switch Repeatability

The control scheme for many gate and globe valves at VEGP utilize a torque switch to trip the operator in the closing direction. The torque switch is a mechanical device which acts in combination with the spring pack to trip the operator when a specific torque output is reached. Spring pack compression is directly related to operator output torque, therefore, by varying the amount of spring pack compression required to trip the torque switch the maximum operator output can be controlled. However, since this is essentially a mechanical process and various frictional forces are involved, there can be stroke to stroke variations in the actual operator output torque for a given switch setting.

Limitorque performed a substantial amount of testing in 1992 to determine the repeatability of the torque switch under various operating conditions. The results of this testing are docum nted in Limitorque Maintenance Update 92-2. The Limitorque derived values for torque switch repeatability are outlined in Table 4-1.

Table 4-1
Torque Switch Repeatability

Torque Switch Setting	Operator Output Torque (ft-lb)	Torque Switch Repeatability
1	< 50	+/- 20%
1	> 50	+/- 10%
> 1	< 50	+/- 10%
> 1	> 50	+/- 5 %

The torque switch repeatability numbers outlined in Table 4-1 are accounted for in the establishment of torque switch settings at VEGP. Section 4.5 of this document outlines the actual methodology utilized to adjust switch settings to account for this uncertainty.

4.2.2 Test Equipment Accuracy

A number of diagnostic test systems have been used in conjunction with the set-up and testing of MOVs at VEGP. These systems are discussed in more detail in Section 7.0 of this document. The test equipment is important to switch settings because in order to ensure that a valve is operating within its design range, the accuracy of the test equipment must be accounted for in the set-up and verification process.

Rising stem valves are currently set-up utilizing VOTES diagnostic equipment. The VOTES system utilizes a yoke mounted strain gauge called the VOTES Force Sensor to measure stem thrust. The Force Sensor is calibrated in place and the overall accuracy of the thrust measurement is a function of several factors associated with the calibration process. Since the accuracy of this device is a function of the calibration process, the overall accuracy of this measurement is valve specific and must be determined in conjunction with the sensor calibration.

Rotating stem valves are currently set-up utilizing VOTES diagnostic equipment designed specifically for quarter-turn valves. The system utilizes a Torque Plug which is mounted on the HBC gearbox to measure the HBC output torque at torque switch trip. The accuracy of the Torque Plug torque measurement is 5.4% for all rotating stem valves.

The accuracy of the VOTES diagnostic test equipment is outlined in the test equipment manuals, document number VM-3094, supplied by the manufacturer, Liberty Technologies. Book 1 entitled "VOTES Users Manual" includes the necessary information to calculate the valve specific accuracy for the rising stem equipment. Book 3 entitled, "Quarter Turn Operators" contains information relative to the accuracy of the rotating stem equipment.

The VEGP static test program accounts for test equipment accuracy in conjunction with the establishment of switch settings. Section 4.5 of this document outlines the actual methodology utilized to adjust the switch settings to account for this uncertainty.

4.2.3 Load Sensitive Behavior

Load Sensitive Behavior, or the Rate-of-Loading effect as it was previously referred to, is a phenomena in which the thrust at torque switch trip tends to be higher under static conditions than under dynamic conditions. Load Sensitive Behavior is defined as:

$$LSB = (T_d - T_s)/T_s$$

Where: T_d = Dynamic Thrust at Torque Switch Trip (lb.)
T_s = Static Thrust at Torque Switch Trip (lb.)

MOVs which have a negative Load Sensitive Behavior produce less thrust under differential pressure conditions than under static conditions. Conversely, MOVs which have a positive Load Sensitive Behavior produce more thrust under differential pressure conditions than under static conditions.

It was initially thought that Load Sensitive Behavior was caused by the torque switch operating earlier under differential pressure conditions, thereby limiting operator output torque. This theory has been disproved however, and the phenomenon is now attributed to differences in the efficiency of the torque to thrust conversion under static and differential pressure conditions. The cause of the phenomenon is not well understood but it is believed to be related to characteristics of the stem, stem nut and stem lubricant.

EPRI, in conjunction with the Performance Prediction Program, set out to develop an analytical model for use in evaluating Load Sensitive Behavior. A substantial amount of testing was performed and a considerable amount was learned about the phenomenon. Based on the testing performed to date it appears that the majority of valves do not exhibit Load Sensitive Behavior to a significant degree.

Unfortunately, the testing has not provided sufficient insight to enable a predictive methodology to be developed which would allow susceptible valves to be identified through analytical means alone.

EPRI abandoned plans to develop a computer model of the operator when it was determined that the phenomenon did not lend itself to a purely analytical solution. EPRI did identify a number of alternatives which could be utilized, subject to applicability, to address Load Sensitive Behavior. The EPRI alternatives were developed primarily utilizing gate valve test data and are not necessarily applicable to globe valves. In addition, the testing was performed utilizing stem lubricants other than anti-seize compounds and may not be applicable to VEGP which currently utilizes Fel-Pro N-5000 as a stem lubricant.

To address Load Sensitive Behavior at VEGP, the VEGP differential pressure test data has been analyzed in an attempt to quantify the phenomenon. Table 4-2 outlines the data obtained in conjunction with the VEGP differential pressure test program. It should be noted that many of the valves which were differential pressure tested at VEGP were controlled by limit switches in the closing direction, therefore, these valves could not be evaluated relative to Load Sensitive Behavior.

A review of the data contained in Table 4-2 indicates that there is a great deal of scatter associated with the Load Sensitive Behavior phenomenon. Some of this scatter can be attributed to torque switch repeatability and test equipment accuracy, each of which can effect the Load Sensitive Behavior calculation. In order to address this phenomenon with respect to switch settings the data was evaluated statistically. The sample mean and sample standard deviation were calculated as follows:

$$\bar{X} = \frac{\sum X}{n}$$

Where: $\bar{X}$ = Sample Mean
n = Sample Size
X = Individual Observations

$$S_x = \sqrt{\frac{\sum (X - \bar{X})^2}{n-1}}$$

Where: S_x = Sample Standard Deviation

Table 4-2
Load Sensitive Behavior Data

Valve Tag No.	Load Sensitive Behavior		Valve Tag No.	Load Sensitive Behavior
1FV-0610	-2%		2FV-0610	-6
1FV-0611	-16%		2HV-5106	2
1FV-5155	7%		2HV-5120	7
1HV-5106	-15%		2HV-5122	-20
1HV-5137	-4%		2HV-8110	10
1HV-5139	-3%		2HV-8111A	-14
1HV-8110	-20%		2HV-8111B	4
1HV-8111B	-4%		2HV-8116	3
1HV-8116	17		2HV-8146	6
1HV-8508A	-5%		2HV-8147	-3
1HV-8508B	1%		2HV-8508A	-3
1HV-8509A	17%		2HV-8508B	-1
1HV-8509B	11%		2HV-8509A	4
1HV-8716A	5%		2HV-8509B	-6
1HV-8716B	1%		2HV-8716A	-13
1HV-8802A	-13%		2HV-8716B	-6
1HV-8804A	5		2HV-8804A	-7
1HV-8804B	-6%		2HV-8804B	-2
1HV-8806	-3%		2HV-8806	-10
1HV-8807A	-1%		2HV-8807A	-2
1HV-8807B	-13%		2HV-8807B	-5
1HV-8813	7%		2HV-8813	2
1HV-8920	-1%		2HV-8923A	-6
1HV-8923A	-2%		2HV-8923B	-11
1HV-8923B	-11%		2HV-8924	10

Evaluating the data in Table 4-2 utilizing this methodology results in the following values for the mean and the standard deviation of the Load Sensitive Behavior data:

$$\overline{X} = -2.3$$
$$S_x = 8.58$$

For the purposes of this evaluation a confidence level of 95% is desired. This confidence level corresponds to approximately two sample standard deviations. Therefore, the 95% confidence band becomes:

$$\bar{X} \pm 2S_x$$

Substituting the previously calculated values results in a 95% confidence band of +14.86 to -19.46. Torque switch settings at VEGP will account for Load Sensitive Behavior based on the 95% confidence band calculation. Section 4.5 of this document outlines the actual methodology utilized to adjust switch settings to account for this uncertainty.

4.3 Other Considerations for Establishing Switch Settings

There are several additional factors which need to be considered when establishing switch settings for MOVs. These factors are related to potential degradations which are not readily quantifiable but which must be considered in the overall MOV set-up and verification process.

4.3.1 Stem Lubricant Degradation

Stem lubrication is important to MOV switch settings because it directly influences valve stem factors. As discussed in Section 3.0, the stem factor is essentially a measure of the efficiency of the torque to thrust conversion. The lower the stem factor, the more efficient the conversion of torque to thrust. If the stem factor decreases for a given torque switch setting then the thrust output at that setting will increase. Conversely, if the stem factor increases for a given torque switch setting then the thrust output will decrease.

The stem factor varies as a function of the stem friction coefficient. The primary factor which could cause the stem friction coefficient to vary following the initial valve set-up is a change in stem lubrication. If the stem lubricant looses effectiveness over the course of a lubricant cycle then the stem friction coefficient may increase. If the stem friction coefficient increases there will be a corresponding increase in the stem factor which will result in a lower thrust output at the previously established torque switch setting.

At VEGP the stems on safety-related rising stem valves are lubricated with Fel-Pro N-5000 on a 36 month interval. Fel-Pro N-5000 is an anti-seize compound with a relatively high viscosity. The lubricant was selected due to its ability to provide effective lubrication, in this application, over an extended period of time.

EPRI performed a substantial amount of stem lubricant testing in conjunction with the EPRI Performance Prediction Program. The results of this program were issued in August 1993 in a report entitled "Stem/Stem Nut Lubrication Test Program". A total of 21 lubricants, including Fel-Pro N-5000, were evaluated in conjunction with this program. The test program evaluated the effectiveness of each lubricant over the course of 500 strokes under a simulated differential pressure loading profile. Unfortunately, the test program did not address the performance of the lubricants in adverse environments and over extended periods of time as would be typical of actual plant conditions. The stem friction coefficient for Fel-Pro N-5000 was 0.143 when the testing began and the stem friction coefficient steadily decreased over the course of the testing to a final value of 0.077 at the conclusion of the 500 strokes. The EPRI testing indicates that the effectiveness of this lubricant actually improves with use.

Table 4-3
Stem Friction Coefficients
Before and After Stem Lubrication

Valve Tag Number	Stem Friction Coefficient Before Lubrication	Stem Friction Coefficient After Lubrication
2R2 Test Data		
2FV-0611	0.150	0.147
2HV-19057	0.115	0.147
2HV-3548	0.128	0.122
2HV-5125	0.103	0.093
2HV-5132	0.091	0.089
2HV-5137	0.114	0.114
2HV-8116	0.082	0.103
2HV-8702A	0.130	0.157
2HV-8811B	0.108	0.102
2LV-0112C	0.101	0.099
2R4 Test Data		
2FV-0610	0.157	0.149
2HV-8110	0.175	0.155
2HV-8146	0.064	0.060
2HV-8485B	0.085	0.079
2HV-8701B	0.126	0.097
2HV-8802B	0.134	0.106
2HV-8835	0.079	0.067
2HV-8840	0.068	0.071

To evaluate the effectiveness of Fel-Pro N-5000 over an extended interval and under actual plant conditions a test program was implemented at VEGP. The initial testing was conducted during 2R2. A sample of ten valves, each of which was due for a 36 month stem lubrication, was selected for the testing and the MOVATS Torque-Thrust Cell was utilized to collect data. Each valve was statically tested in its as-found condition and then retested following stem lubrication. An additional eight valves were identified for testing during 2R4. Each of these valves was tested utilizing the VOTES Force Sensor to measure thrust and the MOVATS TMD in conjunction with a calibrated spring pack to measure torque. The results of the testing were utilized to calculate stem friction coefficients for each of the valves before and after lubrication. The results are outlined in Table 4-3.

The average stem friction coefficient in the as-found condition was 0.112 and the average stem friction coefficient following lubrication was 0.109. Based on the results of this testing it was concluded that Fel-Pro N-5000 did not experience any significant degradation over a 36 month interval under actual plant conditions.

Considering the results of the EPRI testing, as well as the VEGP specific testing, it was concluded that Fel-Pro N-5000 is an effective valve stem lubricant and that the lubricant does not experience any measurable degradation over a 36 month interval. Therefore, it is not necessary to include margin for stem lubricant degradation when establishing MOV control switch settings at VEGP.

4.3.2 Spring Pack Relaxation

Spring pack relaxation is primarily associated with normally closed valves in which the spring pack remains compressed for extended periods of time. When the spring pack remains compressed there is the potential for the belleville washers to relax to some degree thereby reducing the spring pack preload. Since the operation of the torque switch is directly affected by the preload any reduction in preload will result in a reduction in operator output torque at a given torque switch setting.

Limitorque performed testing to quantify spring pack relaxation as a function of spring compression and time. The study evaluated belleville washers at stresses up to 230 KSI for time periods up to 24 months. The results of this testing were then applied to various spring pack configurations to determine the net effect on operator output at various torque switch settings. The results of this study are outlined in Limitorque Technical Update 93-02.

Several interesting conclusions can be reached based on a review of the Limitorque test data. First, the data indicates that the majority of the relaxation occurs in the first twelve months of operation. There is very little additional relaxation in the second twelve month period. Second, the total relaxation experienced by the belleville washers has a relatively insignificant effect on the torque output of the operator at the maximum nominal torque switch setting. The average total relaxation experienced by the spring packs tested was less than 4% which is much lower than the overall accuracy of the equipment utilized to set-up and test MOVs in the field.

Prior to the advent of diagnostic testing, spring pack relaxation would have been a relatively difficult problem to identify. However, with routine periodic testing, utilizing equipment capable of measuring operator output at the torque switch trip point, spring pack relaxation is easily identified and corrected. The VEGP MOV program requires that valves be tested, utilizing diagnostic equipment, at least every five years. The periodic test program ensures that any significant spring pack relaxation will be identified and corrected before it can have a significant affect on operator output. Therefore, it is not necessary to include margin for spring pack relaxation in torque switch settings for VEGP valves.

4.3.3 Miscellaneous Degradations

There has been concern expressed regarding the potential for various age-related degradations to effect the capability of MOVs. The various mechanisms which may be involved are not well understood nor have they been explicitly identified. There is currently little, if any, validated data to suggest that age-related degradation plays a significant role with regard to the performance of safety-related valves in nuclear power plants. Nevertheless, it is a prominent topic of discussion within the industry and one that warrants consideration when developing switch settings.

The primary concern relative to rising stem valves appears to be the potential for a change in valve factors over time. As discussed in Section 3.0, the valve factor is an empirically derived term which accounts for the friction between the valve disk and seat. The various valve factors which are utilized to evaluate MOVs have been developed based on extensive testing in both laboratory and field environments. The valve factors are typically bounding values, which in most cases contained substantial margins over the values measured during actual testing.

The concerns which have arisen relative to the potential for valve factors to experience degradation over time may be largely founded as a result of the EPRI Performance Prediction Program (PPP). EPRI performed extensive differential pressure testing to evaluate valve factors in conjunction with the (PPP). In the early stages of this program EPRI observed a phenomenon characterized by the tendency for valve factors in newly refurbished gate valves to trend up as a function of valve strokes until a plateau was reached, at which point the valve factors stabilized and remained constant. As a result of this observation EPRI began the practice of preconditioning gate valves, to stabilize the valve factor, prior to beginning testing. The valve strokes required to reach this plateau varied from valve to valve but typically involved several hundred strokes. This phenomenon was not observed with respect to globe valves.

The implications this phenomenon may have with respect to safety-related valves in nuclear power plants is unclear. Most of the valves utilized in safety-related applications are rarely stroked except for testing. Valves which are covered by ASME Section XI would be required to be tested on a maximum frequency of once a quarter, although in many cases the valves are only tested during cold shutdown. Testing the valve quarterly would result in a total of 20 strokes between normal GL 89-10 required periodic testing. It is doubtful that 20 strokes would have any measurable affect on valve factors.

With respect to rotating stem valves, there is even less data available to suggest that age related degradation may be a problem. Butterfly valve performance is not evaluated utilizing valve factors, therefore, valve factors are not an issue. There has been some concern expressed regarding the potential for hardening of resilient valve seats and the affect this may have on seating and unseating torque requirements. However, test equipment recently released allows changes in the seating and unseating torque for rotating stem valves to be easily identified in a static test. Therefore, VEGP's periodic test program would be capable of identifying this problem before it could substantially effect the valves performance.

Since time related degradations are not well understood and the potential effects have not been quantified, no attempt has been made to explicitly account for this potential issue. However, the conservative nature of the overall engineering design and field setup process ensure that sufficient margin exists to account for degradations should they occur.

4.4 Control Schemes

In determining the appropriate margins necessary to account for the factors discussed in this section, a review of the various control schemes utilized by the MOVs is in order. Since many of the uncertainties are primarily associated with torque switch settings, it is important to understand the function of the torque switch with respect to the different control schemes. In many cases the torque switch does not play a significant role in the functionality of the MOV and many of the uncertainties do not need to be considered.

4.4.1 Rising Stem Valves

Rising stem valves, which include all gate and globe valves in the GL 89-10 program, utilize several different control schemes at VEGP. Each of these schemes must be addressed independently when developing control switch settings.

4.4.1.1 Opening

The control schemes for all GL 89-10 gate and globe valves at VEGP utilize the open limit switch to trip the operator in the opening direction. In the original plant design, the open torque switch was utilized as a back-up and was capable of tripping the operator in the event the open limit switch failed. However, the open torque switch has been bypassed on all GL 89-10 gate and globe valves with the exception of valves 1HV-8994A/B, 2HV-9017A and 2LV-0112C to ensure that an inadvertent operation of the open torque switch will not impact the capability of the MOV to perform its design function. Valves 1HV-8994A/B will be abandoned in place following the 1996 Unit 1 refueling outage and the open torque switches on valves 2HV-9017A and 2LV-0112C will be bypassed during the 1996 Unit 2 outage. This philosophy parallels the practice of bypassing thermal overload devices in operation.

Since the opening function is controlled exclusively by the open limit switch the open torque switch setting is of no consequence. This means that torque switch repeatability, test equipment accuracy and Load Sensitive Behavior do not need to be considered with respect to opening capability. The open limit switch operates based on position only and is not affected by load, therefore, the entire capability of the operator is available to unseat and open the valve.

4.4.1.2 Closing -Torque Switch Controlled

The control scheme for the majority of gate and globe valves in the GL 89-10 program utilize the close torque switch to trip the operator in the closing direction. The close limit switch is bypassed on these valves and does not perform a control function.

The close torque switch must be set to operate at a thrust greater than the thrust required to close the valve and less than the operator capability at degraded voltage. Valve and operator allowables must also be considered in establishing the maximum setting. Since the torque switch is the normal control device its setting is critical to ensure that the valve will be capable of performing its design function. Torque switch repeatability, test equipment accuracy and Load Sensitive Behavior must all be accounted for in establishing this switch setting.

4.4.1.3 Closing - Limit Switch Controlled

The control scheme for a number of gate and globe valves at VEGP utilize the close limit switch to trip the operator in the closing direction. The close torque switch on these valves is bypassed and does not perform a control function.

The MOVs whic' .ie controlled by the close limit switch utilize Limitorque SB operators. The S.3 operators are equipped with a compensator spring which locates the stem nut and essentially allows the nut to float. As the operator produces thrust to move the valve in the closing direction the load is reacted by the compensator spring and the compensator spring is compressed as a function of the applied load. Since compensator spring compression is directly related to valve thrust, the initial setting of the close limit switch is based on a specified amount of compensator spring compression. The valve is then stroked utilizing diagnostic test equipment to determine if the thrust at the close limit switch trip point is greater than the required closing thrust. MOVs which do not produce the required closing thrust, based on the specified compensator spring deflection, are adjusted to obtain additional thrust, to the extent possible, without exceeding the operator capability or any valve or operator allowables.

It should be noted that a limit controlled valve may not produce the required closing thrust under static conditions at the limit switch trip point. The limit switch operates based on stem travel and the operator may not be required to develop the required closing thrust during a static test. However, since the limit switch does not limit operator output torque, the operator will be capable of producing the required thrust, within the limits of its capability, under differential pressure conditions. The manufacturer's recommended set-up procedures only require that the limit switch be set based on compensator spring deflection. The diagnostic testing based adjustments were added to provide further assurance that limit controlled valves would attain hard seat contact under differential pressure conditions.

Since the closing function is controlled exclusively by the close limit switch the close torque switch setting is of no consequence. This means that torque switch repeatability, test equipment accuracy and Load Sensitive Behavior do not need to be considered with respect to closing capability. The close limit switch operates based on position only and is not affected by load, therefore, the entire capability of the operator is available to close and seat the valve.

4.4.2 Rotating Stem Valves

The control scheme for the rotating stem valves is identical in the open and close direction. The open and close limit switch trip the operator in the opening and closing directions respectively. The open and close torque switch currently provide a backup in the event that the limit switch fails to operate or an excessive load is encountered at an intermediate position in the valve stroke.

Although the open and close torque switch do not perform the primary control function on these valves, since the switches are currently enabled, the settings are important. The switches must be set to allow the operator to produce the required opening and closing torque to ensure that the switch does not inhibit the ability of the valve to perform its design-basis function. Therefore, torque switch repeatability and test equipment accuracy must be accounted for in establishing torque switch settings for rotating stem valves.

It should be noted that the open and close torque switches will be bypassed on all rotating stem valves included in the GL 89-10 program during each valves next scheduled preventive maintenance or periodic static test. Bypassing the torque switches on these valves will ensure that an inadvertent operation of the torque switch will not impact the capability of the MOV to perform its design function. This philosophy parallels that of bypassing the open torque switch on rising stem valves.

4.4.3 Thermal Overloads

The power supplies to all of the safety-related MOVs at VEGP are equipped with thermal overload devices. The thermal overloads are sized to protect the motor in the event of an overload condition and the devices may be enabled while performing valve testing. The thermal overloads are not enabled during plant operation, therefore, the sizing of these devices is not critical to the valves ability to perform its design-basis function. However, the electrical resistance of the thermal elements is significant and is considered in the determination of terminal voltages.

4.5 Switch Setting Adjustments and Margins

The various factors which must be considered with respect to MOV switch settings have been identified. The potential effect each of these factors may have on an MOV, depending on the control methodology, have also been discussed. It is now possible to outline a methodology for adjusting switch settings as required to account for these factors and for determining the as-left setup margin.

4.5.1 Switch Setting Adjustments

The criteria for adjusting switch settings is driven by the valve type and the associated control scheme. Table 4-4 outlines the various factors which must be account for when developing switch settings.

It should be noted that this table was developed explicitly to address switch setting adjustments. It is not meant to infer that test equipment accuracy does not need to be considered with regard to many aspects of the MOV testing process. Rather, it simply indicates that so far as actual switch settings are concerned, test equipment accuracy is only a factor with respect to torque switch settings.

Table 4-4
MOV Switch Setting
Uncertainty Adjustment Guide

Valve Type	Direct. of Travel	Primary Control	Second. Control	Test Equip. Accur.	Torque Switch Repeat.	Load Sensitive Behavior
Rising Stem	Open	Limit Switch	None	N/A	N/A	N/A
Rising Stem	Close	Torque Switch	None	Yes	Yes	Yes
Rising Stem	Close	Limit Switch	None	N/A	N/A	N/A
Rotating Stem	Open	Limit Switch	Torque Switch*	Yes**	Yes**	N/A
Rotating Stem	Close	Limit Switch	Torque Switch*	Yes**	Yes**	N/A

* Torque switches will be bypassed on all rotating stem valves in conjunction with each valves next scheduled preventive maintenance or periodic static test.

** This uncertainty will be N/A following bypassing of torque switches on rotating stem valves.

An error analysis can now be performed to determine the maximum potential error associated with these uncertainties. In applying these uncertainties to switch settings it is not necessary to combine each of the terms algebraically to determine the total uncertainty. Since the uncertainties are independent and random, it is possible to statistically combine these uncertainties into a single factor. For the purposes of adjusting switch settings the square root of the sum of squares (SRSS) was chosen to statistically combine the various uncertainties. When utilizing the SRSS methodology the total uncertainty becomes:

$$U_T = \sqrt{TSR^2 + TEA^2 + LSB^2}$$

Where: U_T = Total Uncertainty
TSR = Torque Switch Repeatability
TEA = Test Equipment Accuracy

LSB = Load Sensitive Behavior

In addition to the total uncertainty, an additional term must be included to account for the bias associated with any of the individual uncertainties which do not have a mean of zero. This does not apply to TSR or TEA since each of these uncertainties have a mean of zero. The mean of the LSB data is -2.3 with a 95% confidence uncertainty of ±17.16, therefore, the bias associated with this uncertainty must be considered. However, to simplify the implementation of this methodology, LSB will be included in the uncertainty calculation based on a mean of zero and an uncertainty of -20. An uncertainty of -20 is conservative relative to a bias of -2.3 and an uncertainty of ±17.16 and will simplify the implementation of this methodology in the field.

The torque switch settings for rising stem valves would be adjusted based on this calculation to ensure that the torque switch would not operated below the minimum required thrust. A similar calculation would be utilized to evaluate the torque switch settings on rotating stem valves. In the case of rotating stem valves, the LSB phenomenon does not apply, therefore, this term would be deleted from the error analysis.

To apply the total error to the switch setting a correction factor would be calculated. The correction factor would be calculated as follows:

$$CF_{MIN} = \frac{1}{1+U_T}$$

Where: CF_{MIN} = Correction Factor for Minimum Evaluation

The torque or thrust measured at torque switch trip would be multiplied by the correction factor to correct the measured reading for the various uncertainties which apply to the valve being tested. The corrected thrust or torque would then be compared to the required thrust or torque to ensure that the corrected value at torque switch trip is higher than the minimum required value.

A similar analysis would also be performed to ensure that the torque switch is not set higher than the maximum available thrust or torque. In this case the correction factor becomes:

$$CF_{MAX} = \frac{1}{1-U_T}$$

Where: CF_{MAX} = Correction Factor for Maximum Evaluation

The torque or thrust measured at torque switch trip would be multiplied by the correction factor to correct the measured reading for the various uncertainties which apply to the valve being tested. It should be noted that the LSB term would be deleted for the total error calculation since it would not apply to the maximum thrust evaluation. The corrected thrust or torque would then be compared to the available thrust or torque to ensure that the corrected value at torque switch trip is less than the maximum available value. In addition, the final thrust or torque, which includes inertial effects, would be multiplied by a correction factor based on test equipment accuracy and compared to the valve and operator allowables to ensure that these values are not exceeded.

4.5.2 MOV Margins

Several calculations may be performed to determine the margin an MOV has relative to its design-basis requirements. These calculations can be useful with respect to evaluating the overall capability of an MOV and determining the sensitivity of the MOV to the various uncertainties associated with the MOV design and maintenance process.

The Capability Margin of an MOV is a measure of the capability an MOV has available beyond that required to perform its design-basis function. The Capability Margin is an engineering calculation which takes credit for the full capability of the MOV without consideration for any potential limitations which may be imposed due to the actual field setup of the valve. Capability Margin is defined as:

$$M_{CAP} = \frac{T_{AVL} - T_{REQ}}{T_{REQ}}$$

Where: M_{CAP} = Capability Margin (%)
T_{AVL} = Available Thrust or Torque (lb or ft-lb)
T_{REQ} = Required Thrust or Torque (lb or ft-lb)

Since the capability margin calculation does not take into account any setup imposed limitations, a more meaningful measure of capability for MOV's utilizing torque switches is the Setup Margin. Setup Margin is defined as:

$$M_{SU} = \frac{T_{AVLADJ} - T_{REQADJ}}{T_{REQADJ}}$$

Where: M_{SU} = Setup Margin (%)

$$T_{AVLADJ} = \frac{T_{REQ}}{1+U_T} = \text{Adjust. Available Thrust or Torque (lb or ft-lb)}$$

$$T_{REQADJ} = T_{REQ} \times (1+U_T) = \text{Adjust. Required Thrust or Torque (lb or ft-lb)}$$

In calculating the setup margin it should be noted that the total uncertainty, U_T, will vary depending upon valve type and whether the available thrust or required thrust are being adjusted. In the case of adjusting the required thrust for rising stem valves, utilizing torque switch control, the total uncertainty must include torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. When adjusting the available thrust the Load Sensitive Behavior term may be deleted. In the case of rotating stem valves, utilizing a torque switch backup, the total uncertainty must include torque switch repeatability and test equipment accuracy when adjusting both the required and available thrust.

The setup margin takes into account the adjustments which must be made to the required and available thrust to account for the uncertainties associated with switch settings. Valve functions which are limit switch controlled and which do not incorporate a torque switch backup are not limited by switch settings. The capability margin and the setup margin for these valve functions are equivalent. However, valve functions which are controlled by a torque switch or which incorporate a torque switch backup are limited by the switch setting. The torque switches must be set to account for the uncertainties identified in Table 4-3 as they apply to the specific valve type. For these valve functions the setup margin will always be less than the capability margin.

5.0 OPERABILITY DETERMINATIONS

The design review performed to support the VEGP GL 89-10 program utilized the best currently available methodology which in many cases was substantially more conservative than that used in the original plant design. The thrust/torque required to operate each valve and the corresponding operator capabilities often varied considerably from the values contained in the "VEGP Motor-Operated Valve Setpoint Manual" which was the document controlling MOV switch settings at the time GL 89-10 was issued. As the design review progressed and the initial revision of the Torque/Thrust Calculations neared completion, Operability Determinations were performed for each of the GL 89-10 MOVs. The Operability Determinations evaluated the as-left configuration of each MOV against the updated GL 89-10 calculations. In cases where apparent problems existed, interim operability justifications were developed until the specific MOV set-ups could be adjusted or modifications could be implemented . The initial Operability Determinations were completed March 4, 1992.

During the 1993 outages, MOV modifications were implemented on each unit to provide additional margin to allow the valves to be set-up to operate within the revised design windows developed in conjunction with the GL 89-10 design review. A total of 53 MOVs were modified during the 1994 outages and at the conclusion of these outages all of the VEGP GL 89-10 had sufficient margins to be set-up within the new design windows.

Beginning with the 1993 outages VEGP implemented a differential pressure test program with the objective of validating the methodology utilized in the design review. This test program was completed in the spring of 1995 and the results of this testing are discussed in detail in Section 9 of this document. As a result of the reconciliation of the test results with the design review calculations, the methodology utilized to predict valve thrust and torque requirements for certain families of valves was revised. The revised methodology was generally more conservative than the methodology utilized in the original design review calculations and in certain cases the existing valve set-ups were outside the revised design windows. In cases where this occurred, Operability Determinations were once again performed to justify the continued use of the MOV in its current configuration until set-ups could be adjusted or modifications could be implemented.

This section identifies the valves which are currently set-up outside the revised design windows and provides a brief summary of the Operability Determination which were developed to support the continued use of these valves during the current operating cycles.

5.1 Rising Stem Valves - Thrust at Torque Switch Trip Less than Calculated Required Closing Thrust

The rising stem valves in Table 5-1 have close torque switch settings which are less than the calculated required closing thrust when adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior.

Table 5-1
As-Left Closing Thrust Evaluation

Valve Tag No.	Thrust at Torque Switch Trip (lb)	Adjusted Required Closing Thrust (lb)	As-Left Margin (%)
1HV-2041	14016	14131	-1
1HV-3009	11510	11980	-4
1HV-8103A	12260	13925	-12
1HV-8103B	12829	13925	-8
1HV-8103C	11660	13925	-16
1HV-8103D	12023	13925	-14
1HV-8111A	11563	13843	-16
1HV-8111B	12144	13843	-12
1HV-8508A	11838	13120	-10
1HV-8509A	11821	13120	-10
1HV-8509B	12367	13120	-6
1HV-8701A	23669	27808	-15
1HV-8702A	23707	27808	-15
1HV-8702B	23082	27808	-17
1HV-8716A	8190	9522	-14
2HV-8103A	11269	13925	-19
2HV-8103B	11576	13925	-17
2HV-8103C	12623	13925	-9
2HV-8103D	11996	13925	-14
2HV-8110	12430	13843	-10
2HV-8508B	12621	13120	-4
2HV-8509A	12075	13120	-8
2HV-8509B	12063	13120	-8
2HV-8701B	25351	27808	-9
2HV-8702A	25738	27808	-7
2HV-8702B	24415	27808	-12
2HV-8920	8697	8934	-2

An Operability Determination has been developed for each of the valves identified in Table 5-1 to justify the continued use of that valve in its current configuration for the remainder of the current operating cycle. Each of the valves will be set-up within the revised design windows at the conclusion of each units next refueling outage.

1HV-2041 - Thermal Barrier Isolation Valve

This valve is normally open to provide ACCW cooling water to the RCP thermal barriers and the valve has a safety function to close in the event of a thermal barrier tube rupture. The as-left closing thrust for valve 1HV-2041 is approximately 1% less than the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. The calculation is based on a conservative design-basis differential pressure of 2235 psid. The analysis does not consider any pressure losses which would result in conjunction with a tube rupture event, when in reality, a substantial pressure drop would be expected across the break and associated piping. In addition, this valve is located downstream of the individual thermal barrier isolation valves which perform a redundant function in the event of a tube rupture. Considering the conservatism inherent in the calculation and the redundant nature of the valves function, this valve will be acceptable for the remainder of this cycle. The operator on this valve will be converted to an SB type operator with limit switch control in the closing direction during the upcoming refueling outage.

1HV-3009 - Steam to AFW Pump Turbine

This valve is normally open to provide steam to the AFW Pump Turbine and the valve has a safety function to close to isolate a downstream line break. The as-left closing thrust for valve 1HV-3009 is approximately 4% less than the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. The calculation is based on a conservative design-basis differential pressure of 1185 psid. The steam generators are operated at a no-load pressure of 1092 psig with an Atmospheric Relief Valve (ARV) setpoint of 1125. Evaluating the valve at the more realistic differential pressure associated with the ARV setpoint results in a positive as-left margin, therefore, the valve will be acceptable for the remainder of this cycle. The operator on this valve will be converted to an SB type operator with limit switch control in the closing direction during the upcoming refueling outage.

1/2HV-8103A/B/C/D - RCP Seal Water Inlet Isolation Valve

These valves are normally open to provide RCP seal injection flow and have a safety function to close to isolate a line break inside containment. The as-left closing thrust for these valves range from approximately 8% to 19% below the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. The calculations are based on a conservative differential pressure of 2715 psid which is equivalent to the shutoff head of the Centrifugal Charging Pumps (CCPs). The calculations do not consider any pressure drop associated with line losses between the CCPs and the break location inside containment. In addition, extensive in-situ differential pressure testing was performed on similar Velan globe valves at VEGP at comparable differential pressures. A total of fourteen valves were tested at differential pressures of greater than 2300 psid and each of the valves closed successfully. Sufficient data was collected in thirteen of the tests to compare the actual closing thrust to the EPRI PPP predicted closing thrust and the actual values ranged from 69% to 90% of the predicted value. Considering the conservative differential pressure utilized in the calculations and the overall conservatism inherent in the EPRI PPP methodology, as indicated by the VEGP in-situ test results, these valves will be acceptable for the remainder of this cycle. The valve stems on these valves will be replaced with a stem incorporating a modified thread configuration during the upcoming refueling outages.

2HV-8110 & 1HV-8111A/B - CCP Normal Miniflow Valve

Valves 2HV-8110 and 1HV-8111A/B are normally open to provide CCP miniflow to the seal water heat exchanger and have a safety function to close on an SI signal to isolate normal CCP miniflow. The as-left closing thrust for these valves range from 10% to 16% below the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. Each of these valves was differential pressure tested and operated successfully at differential pressures ranging from 86% to 97% of the design-basis differential pressure. The required closing thrusts for valves 2HV-8110 and 1HV-8111B were extrapolated to design-basis conditions and are less than the as-left available thrust for the respective valve. Therefore, it can be concluded that these valves are capable of performing their design-basis function. When evaluating the differential pressure test data for valve 1HV-8111A it was determined that the test equipment had been improperly zeroed prior to the test, therefore, the test data for this valve could not be extrapolated to design-basis conditions. However, the as-left closing thrust for this valve is approximately 20% higher than the extrapolated required closing thrust for valves 2HV-8110 and 1HV-8111B, which are identical valves operating under similar conditions. Considering the fact that the valve operated successfully at 86% of its design-basis differential pressure and that the as-left thrust is substantially higher than the extrapolated required thrust for identical valves in similar applications, this valve will be acceptable for the remainder of this cycle. The valve stems on these valves will be replaced with a stem incorporating a modified thread configuration during the upcoming refueling outages.

1HV-8508A, 2HV-8508B &1/2HV-8509A/B - CCP Alternate Miniflow Valve

Valves 1HV-8508A, 2HV-8508B and 1/2HV-8509A/B are normally closed and have a safety function to open on an SI signal concurrent with high CCP discharge pressure to provide CCP miniflow to the RWST. The valves have a safety function to close when transferring to recirculation. The as-left closing thrust for these valves range from 4% to 10% below the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. Each of these valves was differential pressure tested and operated successfully at differential pressures ranging from 90% to 97% of the design-basis differential pressure. The required closing thrust for each of these valves was extrapolated to design-basis conditions and was less than the as-left available thrust for the respective valve. Therefore, it can be concluded that these valves are capable of performing their design-basis function. The overall gear ratio and/or spring packs on these valves will be modified during the upcoming refueling outages.

1HV-8701A, 2HV-8701B, 1/2HV-8702A/B - RCS to RHR Pump Suction

These valves are normally closed and have a safety function to open to align the RHR system for safety-grade cold shutdown. The valves are closed during plant startup following the formation of a bubble in the pressurizer and may be required to close to isolate leaks in the RHR system in modes 4, 5 and 6. These valves have close torque switch settings ranging from 9% to 17% below the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. The required closing thrust for these valves was calculated based on a design-basis differential pressure of 425 psid which is the maximum RCS pressure for RHR system operation. The 425 psid differential pressure was assumed in the calculations for conservatism and to avoid the complexity of performing a dynamic evaluation of the closing differential pressures for these valves. When manipulating the valves in conjunction with a normal plant startup, the RHR pumps are secured prior to closing these valves. Since the pumps are not running there is no flow in the RHR system, therefore, these valves are not exposed to any differential pressure when performing this function. With respect to isolating a leak downstream of these valves, the RHR system is considered a moderate energy system. The limiting failures which must be considered are through-wall cracks which would result in minimal leakage. The resulting differential pressure would occur primarily across the crack, with only limited differential pressure present across the 1/2HV-8701A/B and 1/2HV-8702A/B valves. Considering the conservative differential pressure being utilized in the calculations, these valves will be acceptable for the remainder of this cycle. The operators on these valves will be converted to SB type operators with limit switch control in the closing direction during the upcoming outages.

1HV-8716A - RHR to RCS Hot Leg Isolation Valve

This valve is normally open to crosstie the two trains of the RHR system. The valve has a safety function to close when transferring to cold-leg recirculation and to open when transferring to hot-leg recirculation. The valve also has a safety function to close to isolate a passive failure leak of less than 50 gpm during recirculation and to close when aligning for safety-grade cold shutdown. The as-left closing torque switch setting is approximately 14% below the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. This valve was differential pressure tested at 82% of its design-basis differential pressure and operated successfully. The required closing thrust for this valve was extrapolated to design-basis conditions and was less than the valves as-left available thrust. Therefore, it can be concluded that this valve is capable of performing its design-basis function. The close torque switch on this valve will be adjusted during the upcoming refueling outage.

2HV-8920 - SI Pump B Miniflow Isolation Valve

This valve is normally open to provide SI pump B miniflow to the RWST and has a safety function to close when transferring to recirculation. The as-left closing thrust for valve 2HV-8920 is approximately 2% less than the calculated required closing thrust adjusted for torque switch repeatability, test equipment accuracy and Load Sensitive Behavior. The calculation is based on a design-basis differential pressure of 1521 psid which was determined based on pump shutoff head. This is a very conservative differential pressure because the SI pump will be operating well out on its curve when this valve is required to close in conjunction with the transfer to cold-leg recirculation. In addition, valve 2HV-8813 is located downstream of valve 2HV-8920 and performs a redundant function to isolate SI miniflow when transferring to recirculation. Considering the conservatism inherent in the calculation and the redundant nature of the valves function this valve will be acceptable for the remainder of this cycle. The close torque switch on this valve will be adjusted during the upcoming refueling outage.

5.2 Rotating Stem Valves - Torque at Torque Switch Trip less than Required Opening and/or Closing Torque

The opening and closing torque switch settings were evaluated for all rotating stem valves to ensure that the settings would not inhibit the valves from performing their design-basis functions. Although the torque switches do not perform a primary control function on these valves, the switches are enabled as a backup feature and are capable of tripping the operator if their respective settings are reached.

The revised butterfly valve calculations utilize a packing load which was determined based on EPRI methodology and a standardized packing configuration. Based on the results of both static and dynamic testing performed on butterfly valves at VEGP it was recognized that in some cases the current packing loads were not bounded by this methodology. Therefore, in performing these evaluations, the required opening and closing torque values were adjusted in some cases to compensate for the higher than design packing loads which were measured in conjunction with the static and dynamic test program.

In evaluating the 4, 8 and 10 inch valves it was determined that an assumed packing load of ten times the original Fisher packing load would be conservative for the current packing configuration. For the 4, 8 and 10 inch valves the EPRI calculated packing load was replaced by a packing load equivalent to ten times the original Fisher packing loads for conservatism in the Operability Determinations. In the case of the 18 and 24 inch valves, the packing loads calculated utilizing the EPRI methodology were determined to be bounding relative to the assumed ten times Fisher packing loads, therefore, the required torque calculations for these valves were not adjusted.

As a result of this evaluation, valve 1HV-11612 was determined to have an opening torque switch setting which was approximately 8% below the calculated required opening torque for this valve. Valve 1HV-11612 is an NSCW pump discharge valve with a safety function to open following the start of the valves respective NSCW pump. The valve strokes open each time the pump is started, thereby demonstrating the valves ability to perform its design-basis function. Since the valve is operated against differential pressures equivalent to design-basis conditions in normal operation, the valve has demonstrated its ability to perform its design-basis function. Therefore, this valve will be acceptable in its current configuration for the remainder of this cycle. The open and close torque switches on this valve will be bypassed during the upcoming refueling outage.

In addition, physical interferences on valves 1HV-1807, 1HV-2135 and 1HV-2139 prevent the installation of the VOTES Torque Plugs to measure the available torque at torque switch trip. Valve 1HV-1807 is an NSCW to Containment

Cooler isolation valve and valves 1HV-2135 and 1HV-2139 are NSCW to Containment Auxiliary Cooler & Reactor Cavity Cooler isolation valves. To verify the torque switch settings on valves 1HV-1807 and 1HV-2135, the valves were stroked against their normal operating pressure. The normal operating pressure for these valves is comparable to the design-basis pressure, therefore, this test provided assurance that the valves would be capable of performing their design-basis function. To verify that the torque switch settings on valve 1HV-2139 were acceptable the HB gearbox output torque was determined analytically based on the SMB operator output torque at torque switch trip. The SMB output torque was determined based on spring pack displacement utilizing a springpack curve developed on the B&W springpack tester. The opening and closing torque switches for valves 1HV-1807, 1HV-2135 and 1HV-2139 will be bypassed during the upcoming refueling outages, thereby eliminating the need for determining torque switch settings on these valves.

6.0 DESIGN CHANGE PACKAGES

As a result of the engineering design review a number of design changes were initiated in conjunction with the GL 89-10 Program. These design changes involved physical modifications to the MOVs themselves as well as documentation revisions necessary to implement the results of the design review.

6.1 Phase 1 Modifications

The Phase 1 modifications were implemented as a result of the initial VEGP GL 89-10 design review calculations which were approved November 14, 1991. These calculations were generally more conservative than those utilized in the original plant design and resulted in revised set-up requirements for all of the valves and in certain cases reduced margins to levels which were considered unacceptable for long term operation. DCRs were implemented to revise the existing MOV set-up requirements and to upgrade certain MOVs to provide additional margin.

6.1.1 Setpoint Documents

The switch settings for the VEGP MOVs were originally controlled by the Motor-Operated Valve Setpoint Manual which was maintained by the site Maintenance Department. As a result of the GL 89-10 design review many of the switch settings were changed from that of the original plant design. To implement the new switch settings and formally transmit the information to the site, design change packages (DCPs) were issued. DCP 92-V1N0138-0-2 was issued November 4, 1992 covering the Unit 1 MOVs and DCP 92-V2N0139-0-2 was issued February 26, 1993 covering the Unit 2 MOVs. In addition to implementing the new switch settings, modifications to the open torque switch and torque switch limiter plates were also addressed in these DCPs.

6.1.1.1 MOV Data Sheets

Each DCP contained a document entitled "Safety-Related MOV Setpoint Data Sheet Description" which provides a detailed description of the information contained in the MOV data sheets. The setpoint DCPs included a two page data sheet for each of the 256 valves included in the GL 89-10 Program. The data sheets included information-only and controlled data as well as the valve thrust/torque requirements and operator capabilities and limitations. The second page of the data sheets provides specific information for use in diagnostic testing. All of the VEGP GL 89-10 valves have been set-up based on the revised set-up requirements outlined in the above referenced data sheets.

6.1.1.2 Open Torque Switch Bypass Modification

The safety-related rising stem valves at VEGP utilize a limit switch to trip the operator in the open direction. The open torque switch was not utilized to provide a control function although the switch was enabled and capable of tripping the operator from approximately 30% open to full open. The open torque switch was originally set utilizing the MOVATS TMD based equipment by mounting a load cell above the stem and running the stem into the load cell and subsequently tripping the open torque switch. The conversion to VOTES test equipment prevented the open torque switch from being set in this manner. To simplify the valve set-up process and to give the valve the maximum opportunity to perform its design function the open torque switch on all rising stem valves, with the exception of 1HV-8994A/B, 2HV-9017A and 2LV-0112C have been disabled in conjunction with the Setpoint DCPs. Valves 1HV-8994A/B will be abandoned in place following the Unit 1 1996 refueling outage and the open torque switches on the remaining two valves will be bypassed during the Unit 2 1996 refueling outage.

6.1.1.3 Limiter Plate Deletion

All Limitorque operators are supplied from the factory with limiter plates installed on the torque switches to prevent the switches from being set too high and potentially damaging the valve and/or operator. This plate served a purpose prior to the widespread use of diagnostic test equipment to establish MOV torque switch settings. With the advent of diagnostic test equipment the limiter plate ceased to be a useful device and, in fact, was sometimes a problem because switches could not be set high enough to obtain the desired thrust with the limiter plate in place. The setpoint DCPs provide for the removal of the limiter plate if the valve is being set-up utilizing diagnostic equipment.

6.1.2 Valve and/or Operator Modifications

As a result of the design review it was determined that a number of valves did not have sufficient margin when evaluated utilizing criteria which was more conservative than that utilized in the original plant design. A total of 53 motor-operated valves were subsequently modified to provide additional margin to support the conservative methodology utilized in the design review. DCP 92-V1N0165-0-1 was issued November 9, 1992 covering the Unit 1 modifications, and DCP 92-V2N0166-0-1 was issued February 25, 1993 covering the Unit 2 modifications. The modifications were implemented on each unit during the 1993 refueling outages. Table 6-1 identifies the MOVs which were upgraded and each MOV's respective modifications.

Table 6-1
Phase 1 MOV Modifications

Tag No.	Description	Modifications
1HV-2041	Thermal Barrier Isolation	Gear Set and Springpack
1HV-3009	Steam to AFW Turbine	Gear Set and Springpack
1HV-3019	Steam to AFW Turbine	Gear Set and Springpack
1HV-5106	Steam to AFW Turbine	Gear Set and Springpack
1HV-8471A	CCP Suction	Operator Replacement
1HV-8471B	CCP Suction	Operator Replacement
1HV-8716A	RHR Isolation	Springpack
1HV-8716B	RHR Isolation	Springpack
1HV-8801A	BIT Discharge Isolation	Motor, Gear Set and Springpack
1HV-8801B	BIT Discharge Isolation	Motor, Gear Set and Springpack
1HV-8804A	RHR to CCP Suction	Springpack
1HV-8804B	RHR to SI Pump Suction	Springpack
1HV-8806	SI Pump Suction Isolation	Gear Set and Springpack
1HV-8807A	SI Pump Suction	Operator Replacement
1HV-8807B	SI Pump Suction	Operator Replacement
1HV-8821A	SI Pump to RCS	Valve Stem, SB Conversion and Limit Switch Control Closing
1HV-8821B	SI Pump to RCS	Valve Stem, SB Conversion and Limit Switch Control Closing
1HV-8923A	SI Pump Suction	Operator Replacement
1HV-8923B	SI Pump Suction	Operator Replacement
1HV-8924	CCP Suction to SI Suction	Operator Replacement
1HV-19051	Thermal Barrier Isolation	Gear Set and Springpack
1HV-19053	Thermal Barrier Isolation	Gear Set and Springpack
1HV-19055	Thermal Barrier Isolation	Gear Set and Springpack
1HV-19057	Thermal Barrier Isolation	Gear Set and Springpack
1LV-0112D	RWST to CCP	Motor and Springpack
1LV-0112E	RWST to CCP	Motor and Springpack
2HV-2041	Thermal Barrier Isolation	Gear Set and Springpack
2HV-3009	Steam to AFW Turbine	Gear Set and Springpack
2HV-3019	Steam to AFW Turbine	Gear Set and Springpack
2HV-5106	Steam to AFW Turbine	Gear Set and Springpack
2HV-8111B	CCP Miniflow	Springpack
2HV-8471A	CCP Suction	Operator Replacement
2HV-8471B	CCP Suction	Operator Replacement
2HV-8716A	RHR Isolation	Springpack
2HV-8716B	RHR Isolation	Springpack

Tag No.	Description	Modifications
2HV-8801A	BIT Discharge Isolation	Motor and Gear Set
2HV-8801B	BIT Discharge Isolation	Motor and Gear Set
2HV-8804A	RHR to CCP Suction	Springpack
2HV-8804B	RHR to SI Pump Suction	Springpack
2HV-8806	SI Pump Suction Isolation	Gear Set and Springpack
2HV-8807A	SI Pump Suction	Operator Replacement
2HV-8807B	SI Pump Suction	Operator Replacement
2HV-8821A	SI Pump to RCS	Valve Stem, SB Conversion and Limit Switch Control Closing
2HV-8821B	SI Pump to RCS	Valve Stem, SB Conversion and Limit Switch Control Closing
2HV-8923A	SI Pump Suction	Operator Replacement
2HV-8923B	SI Pump Suction	Operator Replacement
2HV-8924	CCP Suction to SI Suction	Operator Replacement
2HV-19051	Thermal Barrier Isolation	Gear Set and Springpack
2HV-19053	Thermal Barrier Isolation	Gear Set and Springpack
2HV-19055	Thermal Barrier Isolation	Gear Set and Springpack
2HV-19057	Thermal Barrier Isolation	Gear Set and Springpack
2LV-0112D	RWST to CCP	Motor and Springpack
2LV-0112E	RWST to CCP	Motor and Springpack

6.2 Phase 2 Modifications

The Phase 2 modifications are being implemented as a result of the evaluation and reconciliation of the VEGP differential pressure test data and the subsequent re-evaluation of certain valves utilizing more conservative methodology. The re-evaluation process resulted in revised set-up requirements and reduced margins for certain valves. Revised Setpoint Data sheets were issued for all of the GL 89-10 valves on June 28, 1995 and DCRs are being developed to modify certain valves to provide additional margin and added insurance that the valves will be capable of performing their design-basis functions.

6.2.1 Butterfly Valve Torque Switch Bypass Modifications

The safety-related rotating stem valves at VEGP utilize a limit switch to trip the operator in both the opening and closing direction. The torque switches are not utilized to perform a primary control function in either the open or close direction, however, the torque switches are enabled and are capable of tripping the operator if the operator output torque reaches the switch setting. Although the torque switches do not perform a control function, with the switch enabled for the entire stroke, the switch settings are critical to the performance of the valve. Setting the switches requires the use of specialized test equipment and due to the extreme torque multiplication provided by the HB gear box, the torque switch is very sensitive and reaching a precise setting is very difficult. Since the torque switches do not perform a control function, these switches will be bypassed in conjunction with each butterfly valves next scheduled preventive maintenance or periodic test, which ever comes first. Deleting this switch from the control circuitry will provide additional assurance that these valves will be capable of performing their design-basis function. These modifications will be implemented under DCPs 95-V1N0035 and 95-V2N0036.

6.2.2 Valve and/or Operator Modifications

As a result of the evaluation of the differential pressure test data and the subsequent reconciliation of that data with the engineering calculations it was determined that a number of MOV's did not have sufficient margin. A total of 58 MOV's have been identified for modifications in order to increase the capability margins to acceptable levels. These modifications will be implemented under DCPs 95-V1N0022 and 95-V2N0023 during each units 1996 refueling outage. Table 6-2 identifies the MOV's to be modified and the respective modifications.

Table 6-2
Phase 2 MOV Modifications

Tag No.	Description	Modifications
1HV-2041	Thermal Barrier Isolation	SB Conversion, Limit Control
1HV-3009	Steam to AFW Turbine	SB Conversion, Limit Control
1HV-3019	Steam to AFW Turbine	SB Conversion, Limit Control
1HV-5113	CST to AFW Pump	Gear Set
1HV-8103A	RCP Seal Water	Replace Valve Stem
1HV-8103B	RCP Seal Water	Replace Valve Stem
1HV-8103C	RCP Seal Water	Replace Valve Stem
1HV-8103D	RCP Seal Water	Replace Valve Stem
1HV-8105	Charging Pump to RCS	Gear Set
1HV-8106	Charging Pump to RCS	Gear Set
1HV-8110	CCP Miniflow	Replace Valve Stem
1HV-8111A	CCP Miniflow	Replace Valve Stem
1HV-8111B	CCP Miniflow	Replace Valve Stem
1HV-8508A	CCP Miniflow	Gear Set and Springpack
1HV-8508B	CCP Miniflow	Gear Set and Springpack
1HV-8509A	CCP Miniflow	Springpack
1HV-8509B	CCP Miniflow	Springpack

Tag No.	Description	Modifications
1HV-8701A	RHR Pump Loop Suction	SB Conv., Limit Control, Gear Set
1HV-8701B	RHR Pump Loop Suction	SB Conversion, Limit Control
1HV-8702A	RHR Pump Loop Suction	SB Conversion, Limit Control
1HV-8702B	RHR Pump Loop Suction	SB Conv., Limit Control, Gear Set
1HV-8809A	RHR Discharge	Gear Set
1HV-8809B	RHR Discharge	Gear Set
1HV-9002A	Containment Spray Suction	Gear Set
1HV-9002B	Containment Spray Suction	Gear Set
1HV-9003A	Containment Spray Suction	Gear Set
1HV-9003B	Containment Spray Suction	Gear Set
1HV-9017A	Containment Spray Suction	Gear Set
1HV-9017B	Containment Spray Suction	Gear Set
2HV-2041	Thermal Barrier Isolation	SB Conversion, Limit Control
2HV-3009	Steam to AFW Turbine	SB Conversion, Limit Control
2HV-3019	Steam to AFW Turbine	SB Conversion, Limit Control
2HV-5113	CST to AFW Pump	Gear Set
2HV-8103A	RCP Seal Water	Replace Valve Stem
2HV-8103B	RCP Seal Water	Replace Valve Stem
2HV-8103C	RCP Seal Water	Replace Valve Stem
2HV-8103D	RCP Seal Water	Replace Valve Stem
2HV-8105	Charging Pump to RCS	Gear Set
2HV-8106	Charging Pump to RCS	Gear Set
2HV-8110	CCP Miniflow	Replace Valve Stem
2HV-8111A	CCP Miniflow	Replace Valve Stem
2HV-8111B	CCP Miniflow	Springpack
2HV-8508A	CCP Miniflow	Gear Set and Springpack
2HV-8508B	CCP Miniflow	Gear Set and Springpack
2HV-8509A	CCP Miniflow	Springpack
2HV-8509B	CCP Miniflow	Springpack
2HV-8701A	RHR Pump Loop Suction	SB Conv., Limit Control, Gear Set
2HV-8701B	RHR Pump Loop Suction	SB Conv., Limit Control, Gear Set
2HV-8702A	RHR Pump Loop Suction	SB Conv., Limit Control, Gear Set
2HV-8702B	RHR Pump Loop Suction	SB Conv., Limit Control, Gear Set
2HV-8809A	RHR Discharge	Gear Set
2HV-8809B	RHR Discharge	Gear Set
2HV-9002A	Containment Spray Suction	Gear Set
2HV-9002B	Containment Spray Suction	Gear Set
2HV-9003A	Containment Spray Suction	Gear Set
2HV-9003B	Containment Spray Suction	Gear Set
2HV-9017A	Containment Spray Suction	Gear Set
2HV-9017B	Containment Spray Suction	Gear Set

7.0 STATIC TEST PROGRAM

VEGP began performing static testing on MOVs prior to commercial operation. The I & E Bulletin 85-03 scope of MOVs was tested on both units prior to start-up and most of the remaining safety-related, rising stem valves on Unit 2 were tested prior to start-up. VEGP completed the initial static testing of all MOVs included in the GL 89-10 Program during the 1993 refueling outages.

7.1 Diagnostic Test Equipment

VEGP has utilized a variety of MOV test equipment to perform diagnostic testing. A tremendous amount of knowledge and experience has been gained since 1985 regarding MOV test equipment, and new and improved versions of this equipment have been released on a regular basis. VEGP has continually monitored the development of this equipment to ensure that the most accurate, technically advanced and cost effective diagnostic systems are employed at VEGP.

7.2.1 MOVATS Test Equipment

VEGP initially utilized the MOVATS 2000 Series of equipment which was an oscilloscope-based system utilizing the Thrust Measurement Device (TMD) as its primary thrust measurement sensor. The plant later upgraded to the MOVATS 3000 Series of equipment which is a computer-based system designed to utilize a variety of sensors. VEGP performed diagnostic testing on rising stem valves with the MOVATS 3000 Series equipment utilizing the TMD, the Stem Strain Transducer, the Stem Strain Ring, the Stem Load Sensor and the Torque Thrust Cell.

In addition, VEGP utilized the MOVATS Butterfly Analysis and Review Test System (BARTS) in conjunction with the 2000 and 3000 Series data acquisition systems to perform testing on rotating stem valves.

The use of the MOVATS test equipment has been discontinued at VEGP. The TMD methodology proved to be inaccurate and the Torque/Thrust Cell is an intrusive device which was not deemed to be practicable for long term use. The TMD is still used to measure spring pack displacement but is not utilized as a primary thrust measurement sensor.

7.1.2 VOTES Test Equipment

VEGP procured Valve Operation Test & Evaluation System (VOTES) from Liberty Technology and began using this equipment as its primary MOV test system during the Spring 1993 Unit 1 refueling outage. The VOTES test system is a computer-based system which utilizes the VOTES Force Sensor as its primary thrust measurement sensor. In addition, the VOTES test equipment is compatible with the VOTES Torque Cartridges, which are capable of measuring operator output torque. It is VEGPs intention to utilize the VOTES test system to perform all rising stem MOV testing in the future.

To perform testing on rotating stem valves the VOTES equipment may be utilized with Torque Plugs. The Torque Plugs are mounted on the HB gearbox and are utilized to provide an artificial stop for the purpose of loading the operator and measuring the available torque. In addition, strain gauge based stem mounted sensor have recently become available which allow running loads to be monitored on rotating stem valves. It is VEGPs intention to utilize the VOTES test system to perform all rotating stem MOV testing in the future utilizing Torque Plugs and/or stem mounted sensors.

7.2 Test Equipment Inaccuracy

There have been a number of issues raised regarding the accuracy of the various MOV diagnostic systems which have been utilized to set-up safety-related MOVs in nuclear plants. The NRC issued Information Notices 92-23, 92-83, 93-01, 93-54 and Supplement 5 to Generic Letter 89-10 dealing with various aspects of this problem. VEGP has evaluated all of the as-left switch settings on safety-related MOVs set-up utilizing diagnostic test equipment to ensure that no operability concerns exist relative to test equipment accuracy.

7.2.1 MOVATS TMD

A problem was identified relative to the use of the TMD to measure thrust in the closing direction based on a calibration performed in the opening direction. The entire scope of VEGP safety-related MOVs which were set-up utilizing this equipment were evaluated and dispositioned with respect to this problem. A summary of the results of this review were transmitted to the NRC in Letter No. LCV-0136 dated September 15, 1993. The use of the TMD to measure thrust has been discontinued at VEGP.

7.2.2 MOVATS Torque/Thrust Cell

A problem was identified concerning the use of the Torque/Thrust Cell in applications where there was minimal stem to stem nut engagement. The entire scope of VEGP safety-related MOVs which were set-up utilizing this equipment were evaluated and dispositioned with respect to this problem. A summary of the results of this review are contained in Letter No. SG-12779 dated September 24, 1993. The use of the Torque/Thrust Cell has been discontinued at VEGP.

7.2.3 VOTES Force Sensor

Liberty Technology identified several problems associated with the use of the VOTES Force Sensor. The problems involved the material constants being utilized in the calibration process and a torque effect which was not being addressed in the calibration process. Liberty addressed each of these issues in Release 2.3 of the VOTES software. VEGP did not perform testing utilizing the VOTES equipment prior to the release of the new software, therefore, the potential for this problem to have occurred at VEGP did not exist.

7.3 Static Test Procedures

Static testing at VEGP is performed on a 5 year/3 cycle interval as recommended in the Generic Letter. The VEGP static test program and associated procedures are outlined in the "Motor-Operated Valve Program Manual". This manual and the documents referenced therein provide a detailed description of the VEGP static test program.

8.0 DIFFERENTIAL PRESSURE TEST PROGRAM

VEGP implemented a differential pressure test program with the objective of validating the analytical methodology utilized in the engineering design review. A test scope was identified which included approximately 30% of the valves covered by the GL 89-10 program. The data collected in these tests was evaluated and compared to the design review calculations to ensure that the design methodology conservatively predicts the performance of the VEGP safety-related MOVs.

8.1 Differential Pressure Test Scope Identification

The identification of the MOVs to be differential pressure tested at VEGP was a two step process. The MOVs were first prioritized based on quantitative design criteria aimed at identifying those valves which would be most challenged in performing their design-basis function. The MOVs were then evaluated to identify those which could be tested under conditions close enough to design-basis to provide meaningful test data.

8.1.1 Valve Prioritization

A prioritization process was developed to identify the MOVs at VEGP which were candidates for differential pressure testing. The objective of the prioritization process was to rank the MOVs relative to the importance of performing differential pressure testing. The MOVs were evaluated based upon criteria aimed at identifying MOVs which must perform under severe service conditions and /or appear to be marginal based on an analytical evaluation.

8.1.1.1 Service Factor

The design-basis differential pressure was chosen as the criteria for evaluating the relative severity of each MOVs operating requirements. Differential pressure is the primary source of dynamic loads and the greater the differential pressure, the larger the difference in loads between static and dynamic conditions. Each MOV was assigned a service factor as outlined below:

Maximum Differential Pressure	Service Factor
DP ≤ 200 psi	1
200 psi < DP < 500 psi	2
DP ≥ 500 psi	3

Valves which are required to operate at design basis conditions of less than 200 psi differential pressure do not experience significant dynamic loading during operation. Industry testing has not identified valves in these applications as being anomalous performers regardless of valve type. These valves appear to exhibit predictable performance and torque/thrust requirements for these valves can be determined utilizing analytical techniques.

Valves which operate in the 200 psi to 500 psi range experience moderate dynamic loading. These valves warrant more attention than valves operating at less than 200 psi. However, these are not severe valve applications by any standard. In general, industry testing has not identified significant problems for valves operating under these conditions.

Valves operating against differential pressures of 500 psi and above experience the most significant loading increases under dynamic conditions. Virtually all of the concerns which have been identified within the industry relative to anomalous valve behavior involve testing performed at differential pressures greater than 500 psi. In reality, most of the problems have occurred at differential pressures considerably higher than 500 psi. However, 500 psi was chosen as a conservative value to categorize high differential pressure valves. Differential pressure test data for valves operating under these conditions will provide the most useful information regarding differences in valve performance between static and dynamic conditions.

8.1.1.2 Margin Factor

The percent difference in the calculated required torque/thrust versus the calculated available torque/thrust was chosen as the criteria for evaluating margins for each MOV. The greater the margin between the available torque/thrust and the required torque/thrust, the less critical the accuracy of the analytical methodology. Each MOV was assigned a margin factor as outlined below:

Calculated Margin	Margin Factor
Margin ≥ 200%	1
100% < Margin < 200%	2
Margin ≤ 100%	3

MOVs which are equipped with operators capable of producing greater than 200% of the required torque/thrust can be setup to ensure that the MOV will be capable of performing its design-basis function with a high degree of confidence. Even if an MOV in this category exhibited some type of anomalous behavior, the reserve capability inherent in the operator ensures that the MOV will be capable of performing its design-basis function.

MOVs which have margins between 100% and 200% warrant additional attention over those in the previous category. However, the operators on these MOVs still have substantial reserve capacity available should some type of anomalous behavior occur.

MOVs with margins of less than 100% require closer scrutiny to ensure that they are setup to operate within their design window. Diagnostic equipment accuracy, operator repeatability and the Load Sensitive Behavior phenomenon all serve to reduce the available margin. Differential pressure tests performed on MOVs in this category are useful in substantiating analytical methodologies which, by virtue of the reduced margins, must be inherently conservative.

It should be noted that the differential pressures and margins utilized in conjunction with the development of the original differential pressure test priorities were based on the original VEGP GL 89-10 calculations. The original differential pressure test priorities were revised in April 1994 to reflect a number of revisions to the original GL 89-10 calculations including the incorporation of the Limitorque A.C. motor elevated temperature Part 21, the use of pullout efficiency to calculate closing capability for D.C. operators and the implementation of DCPs on the CCP alternate miniflow piping. The final scope of differential pressure test valves was determined based on this revised priority matrix.

The differential pressure testing concluded in March 1995 and the reconciliation of the differential pressure test data with the design calculations resulted in a number of revisions to the original design calculations to ensure that the test results were adequately bounded by the analytical methodology. In addition, in certain cases, modifications will be implemented as a result of this revised methodology to provide additional margin. The revised methodology and the additional modifications each affect the calculated margins and may have changed the differential pressure test priorities for certain valves.

8.1.1.3 MOV Priority

The differential pressure test priority for each MOV was determined by adding the service factor and the margin factor. This resulted in a differential pressure test priority of between 2 and 6 for each valve with 6 being the highest priority and 2 being the lowest priority. An individual priority was determined for both the opening and closing direction. The differential pressure test priorities for each valve are discussed in Volume 2 of the VEGP close-out report entitled "Design-Basis Capability Verification Report".

8.1.2 Determination of Testable Valves

Each of the MOVs included in the GL 89-10 scope was evaluated by site engineering and operations personnel to determine the feasibility of performing a meaningful differential pressure test. For the purposes of this evaluation a differential pressure of 50% of the design-basis differential pressure was selected as a breakpoint to identify testable MOVs. In performing the evaluations, it was later determined that the maximum attainable differential pressure was typically either significantly higher or lower than 50% of the design-basis differential pressure. The 50% breakpoint was not a factor in most of the determinations. The testability of individual valves is discussed in more detail in the "Design-Basis Capability Verification Report".

8.1.3 Differential Pressure Test Scope

The scope of MOVs to be differential pressure tested in conjunction with the VEGP GL 89-10 Program was reviewed and discussed with NRC personnel during an audit conducted November 12 and 13, 1992. VEGP originally proposed to perform differential pressure tests on all priority 5 and 6 MOVs which were determined to be testable. This approach was selected to ensure that MOVs which must operate against high differential pressures and/or have marginally sized operators would be tested.

A matrix of the VEGP GL 89-10 valves grouped by manufacturer, type, size and ANSI rating was also reviewed with the NRC personnel. Organizing the GL 89-10 valves in this manner results in twenty-nine valve groups, eleven of which were represented in the original VEGP Differential Pressure Test Plan. The NRC personnel expressed concerns regarding VEGPs ability to develop acceptable operability justifications for the valves in groups in which a representative sample was not tested.

To address these concerns, the original VEGP Differential Pressure Test Plan was reviewed to determine if the scope of valves could be revised to reflect a more representative sampling of valves while continuing to adequately address the high priority valves. As a result of this review the following revisions were made to the original VEGP Differential Pressure Test Plan:

1. In the original test plan sixteen 4"-900 lb. Fisher globe valves were scheduled for testing. The scope of valves to be tested in this group was reduced to a total of four valves. The justification for the reduced test sample is that globe valves have been predictable performers and that a test scope of four valves is sufficient to identify any problems associated with these valves.

2. In the original test plan twelve 1.5"-1500 lb. Velan globe valves were scheduled for testing. The scope of valves to be tested in this group was reduced to a total of four valves. The justification for the reduced test scope is that globe valves have been predictable performers and that a test scope of four valves is sufficient to identify any problems with these valves.

3. In the original test plan only the testable priority 5 and 6 Westinghouse gate valves were scheduled for testing. The test scope of Westinghouse valves was increased to include all testable priority 4 through 6 valves. This provided a more comprehensive Westinghouse gate valve test scope and focused attention on the valve type which has typically been the most difficult to address analytically.

4. The original test plan did not include any butterfly valves. A total of four butterfly valves were added to the test scope. This provided baseline test data on this valve type.

The gate valve test scope was increased to include a more representative sample of the Westinghouse EMD gate valves. The majority of the problems that have been identified relative to MOVs have involved gate valves and the gate valve test scope was expanded to address this concern.

The globe valve test scope was reduced but still includes a representative sampling of the VEGP globe valve population. Minimal problems have been identified regarding globe valves and the revised scope, which includes valves from five of the six globe valve groups, was adequate to identify any potential problems.

A sample of butterfly valves was added to the test scope. Butterfly valves operate against relatively low differential pressures and typically have substantial margins. The relatively limited scope of butterfly valves included in the test scope was sufficient to verify the analytical methodology utilized to evaluate these valves.

A total of 83 MOVs were ultimately identified for differential pressure testing in support of the VEGP GL 89-10 program. The valves which were tested are identified in Section 9.0 of this document.

8.2 Differential Pressure Test Procedures

The differential pressure tests were performed under Temporary Engineering Procedures developed on-site by the Engineering Support Organization. These procedures specify the various system configurations required to perform each test and include data sheets to record the pertinent system data. The actual motor-operated valve test data taken during the test was controlled by Procedure 26866-C entitled "Dynamic Testing of Motor Operated Valves Using VOTES Analysis and Test System". This procedure outlined the steps necessary to obtain the required MOV data and to perform the necessary on-site evaluations of the data. The completed Temporary Engineering Procedure packages, including the applicable Procedure 26866-C data sheets, were transmitted to SCS for detailed evaluation following completion of the test.

8.3 Differential Pressure Test Data Evaluations

The data collected during the in-situ differential pressure tests was analyzed to validate the analytical methodology utilized in the design review. The evaluation process essentially involved two independent reviews. An initial evaluation was performed on site by the test personnel prior to returning the MOV to service. A more detailed evaluation was performed later off-site by SCS engineering personnel.

8.3.1 On-Site Evaluation

The MOV test personnel were responsible for performing a preliminary evaluation of the differential pressure test data on-site prior to returning the MOVs to service. The objective of this evaluation was to ensure that the valves would be capable of performing their design-basis function based on the results of the differential pressure test. Included in this evaluation was the extrapolation of the actual test data to design-basis conditions. The acceptance criteria for this evaluation is included in Procedure 26866-C.

The test data was evaluated to ensure that the MOV opened and closed acceptably, that no significant anomalies were present during the test and that the control switch trip point was within the values specified in the Setpoint Document. Any problems identified during this preliminary review were evaluated and resolved prior to returning the valves to service. Following the successful completion of this review the MOV was returned to service and the data was forwarded to SCS for detailed evaluation.

8.3.2 Engineering Evaluation

The evaluation performed by SCS was controlled by the Southern Company Services Plant Vogtle Operational Support Policy and Procedures Manual Appendix J entitled "Guidelines for Reviewing VEGP Generic Letter 89-10 Motor-Operated Valve (MOV) Differential Pressure Test Data". This guideline delineates the detailed evaluation to be performed to validate the methodology utilized in the GL 89-10 design review. Items which were reviewed in conjunction with this guideline included the valve factors, stem friction coefficients and Load Sensitive Behavior as well as any additional parameters potentially affecting the design review calculations. Any anomalies identified in conjunction with this review were evaluated and dispositioned. In addition, a broadness review was performed to address any additional valves which may have been affected. The detailed evaluation of the test data collected in conjunction with this testing is complete and the results of these evaluations are discussed in detail in Section 9.0 of this document.

9.0 DIFFERENTIAL PRESSURE TEST DATA EVALUATION AND RECONCILIATION

The differential pressure test data was evaluated in order to validate the empirical methodology utilized in the engineering design review. The evaluation included a detailed review of the test data to identify any apparent anomalies associated with a specific test as well as an analytical evaluation of the relevant test data in order to compare actual test values to design values. Differential pressure tests which did not attain at least 50% of the design differential pressure were not evaluated due to concerns regarding the extrapolation of result to design-basis conditions. The results of this review were utilized to confirm the design review methodology in cases were the test results were bounded by the methodology and as a basis for revising the methodology in cases where the test results were not bounded by the methodology.

The differential pressure test valves have been categorized based on valve groups. These valve groups are discussed in more detail in the "Design-Basis Capability Verification Report", but they are essentially groups of identical valves based on manufacturer, valve type, ANSI rating and size.

The initial discussions cover the reviews performed to evaluate the loads associated with opening and closing the valves in each valve group under differential pressure conditions. In many cases, the original design review methodology was revised, based on the differential pressure test results, to ensure that the methodology provided a conservative prediction of the opening and closing loads. In those instances where the methodology was revised, the test data was re-evaluated based on the revised methodology.

Also included in this section is a discussion of the evaluations which were performed to justify the stem friction coefficient values utilized in the engineering calculations. Data collected in conjunction with the differential pressure test program, as well as additional static test data, were utilized to evaluate this parameter.

9.1 Anchor Darling Gate Valves

A total of three Anchor-Darling gate valves, representing one of the four Anchor-Darling valve groups, were differential pressure tested. The valves which were tested are identified in Table 9-1.

Table 9-1
Anchor-Darling Gate Valve
Differential Pressure Test Scope

Group No.	Tag No.	Procedure No.	Test Date
AD-4	1HV-3009	T-ENG-93-01	4/24/93
AD-4	1HV-5106	T-ENG-94-15	10/14/94
AD-4	2HV-5106	T-ENG-93-31	10/19/93

The Anchor-Darling gate valve differential pressure tests were originally evaluated based on the Industry Standard Equation which is described in detail in Section 3 of this document. In reviewing the overall population of Anchor-Darling gate valves it became apparent that only a limit number of valves could be differential pressure tested in-situ under design-basis conditions. Therefore, the decision was made to re-evaluate the Anchor-Darling valves utilizing the EPRI Performance Prediction Program methodology.

The EPRI PPP methodology was developed based on an extensive analytical evaluation of the gate valve and is supported by extensive separate effects testing. The EPRI methodology was verified based on flow loop testing and, therefore, is a more defensible methodology than the industry standard equation when only limited in-situ differential pressure test data is available. The differential pressure test evaluations were revised to reflect this new methodology.

In performing the initial evaluation of the differential pressure test data, the test results were extrapolated to design-basis conditions utilizing linear extrapolation. The use of linear extrapolation was justified based on a review of the test reports issued by EPRI documenting the testing performed in conjunction with the Performance Prediction Program. The EPRI data indicated that as differential pressures increased, with a corresponding increase in contact stresses, friction coefficients tended to decrease. This trend was noted throughout the EPRI test data, which includes a number of Anchor-Darling gate valves similar to those installed at VEGP.

9.1.1 Anchor-Darling Group AD-4 Gate Valves

This group is composed of six, 4-0 inch, 900 lb, Anchor-Darling gate valves. The valves are located in the steam supply piping to the AFW pump turbines. Differential pressure tests were performed on a total of three valves in this group.

Differential Pressure Test Overview

1HV-3009 - Main Steam to Auxiliary Feedwater Pump Turbine

This valve is normally open and has a safety function to close to isolate a downstream break in the steam supply piping to the AFW pump turbines. The valve was tested in the closing direction only. This valve is not required to open under a differential pressure and opening the valve under a differential pressure is not compatible with the design and normal operation of the system. The valve closed successfully at the test differential pressure.

When the test results were evaluated it was determined that the differential pressure at flow isolation was much lower than originally expected. The lower than expected differential pressure was determined to be due to the large volume of piping downstream of the valve and the flow restriction caused by the throttled AFW pump turbine governor valve. Since steam is a compressible fluid the steam downstream of the valve expanded maintaining a substantial downstream pressure at the point of flow isolation rather than collapsing suddenly as would have been the case with an incompressible fluid such as water. The differential pressure was estimated to be approximately 500 psid at flow isolation, however, the differential pressure was changing rapidly at this point and an accurate correlation of the differential pressure to the flow isolation point is not possible. Since the differential pressure was less than 50% of the design-basis differential pressure and due to the uncertainties associated with determining this differential pressure at the point of flow isolation the test results for this valve were not evaluated.

1/2HV-5106 - Steam Supply to the Auxiliary Feedwater Pump Turbine

These valve are normally closed and have a safety function to open to admit steam to the AFW pump turbine. The valves were tested in the opening and closing direction. These valves are not required to close against a differential pressure but were tested in that direction to provide generic data for the remaining Anchor-Darling valves which were not differential pressure tested. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collect in each test.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Anchor-Darling Group AD-4 valves were performed utilizing the industry standard equation with a 0.5 valve factor. The Anchor-Darling differential pressure test data was originally evaluated based on this methodology. A valve factor was backed out of the test data and compared to the design valve factor to determine if the valves performance was bounded by the design methodology. Table 9-2 summarizes this evaluation for the group AD-4 valves.

Table 9-2
Anchor-Darling Group AD-4 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design VF Open	Test VF Open	Design DP Close psid	Test DP Close psid	Design VF Close	Test VF Close
1HV-3009	1185	0	0.5	N/A	1185	500	0.5	N/A
1HV-5106	1185	1075	0.5	0.44	1185	1075	0.5	0.23
2HV-5106	1185	1048	0.5	0.47	1185	1200	0.5	0.35

The results of the evaluation indicate that the tested valves are performing within the bounds of the design methodology. Both the opening and closing valve factors for the tested valves are less than the design valve factors.

To support the use of the EPRI methodology on the Anchor-Darling valves the differential pressure test data was re-evaluated based on the EPRI methodology. Since the EPRI gate valve model is a computer based methodology utilizing a series of complex algorithms to analyze the valves performance the methodology which was utilized to evaluate the differential pressure test data was considerably different than that used to evaluate the data based on the industry standard equation. To evaluate the test data based on the EPRI methodology, the EPRI model was re-run utilizing the actual test conditions and the required opening and closing thrusts were compared to the thrusts measured during the differential pressure test to determine if the valves performance was bounded by the EPRI methodology. Table 9-3 summarizes this re-evaluation for the group AD-4 valves.

Table 9-3
Anchor-Darling Group AD-4 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	EPRI Opening Thrust lb	Test Opening Thrust lb	Test/ EPRI Open	EPRI Closing Thrust lb	Test Closing Thrust lb	Test/ EPRI Close
1HV-5106	8997	4878	54%	9008	4564	51%
2HV-5106	8811	4557	52%	9876	6297	64%

The results of the re-evaluation indicate that the tested valves are performing within the bounds of the EPRI methodology. Both the opening and closing thrust for the tested valves was less than the required thrust predicted by the EPRI methodology. In fact, for the valves tested in-situ at VEGP, the EPRI methodology substantially over predicted the required opening and closing thrusts.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that both the industry standard equation utilizing a 0.5 valve factor and the EPRI Performance Prediction Program methodology conservatively predict thrust requirements for Anchor-Darling 4 inch, 900 lb, gate valves.

9.2 Westinghouse Gate Valves

A total of 46 Westinghouse gate valves, representing seven of the fourteen Westinghouse valve groups, were differential pressure tested. The valves which were tested are identified in Table 9-4.

Table 9-4
Westinghouse Gate Valve
Differential Pressure Test Scope

Group No.	Tag No.	Procedure No.	Test Date
W-2A	1FV-0610	T-ENG-94-21	10/31/94
W-2A	1FV-0611	T-ENG-94-22	11/01/94
W-2A	1HV-8146	T-ENG-94-16	09/29/94
W-2A	1HV-8147	T-ENG-94-16	09/29/94
W-2A	2FV-0610	T-ENG-95-03	02/14/95
W-2A	2FV-0611	T-ENG-95-04	01/24/95
W-2A	2HV-8146	T-ENG-95-05	03/23/95
W-2A	2HV-8147	T-ENG-95-05	03/23/95
W-2B	1HV-8105	T-ENG-94-10	09/23/94
W-2B	1HV-8106	T-ENG-94-09	10/06/94
W-2B	2HV-8105	T-ENG-93-30	09/30/93
W-2B	2HV-8106	T-ENG-93-30	09/30/93
W-4	1HV-8821A	T-ENG-93-11	04/13/93
W-4	1HV-8821B	T-ENG-93-11	04/12/93
W-4	2HV-8821A	T-ENG-93-27	09/29/93
W-4	2HV-8821B	T-ENG-93-32	10/07/93
W-5	1HV-8801A	T-ENG-93-11	03/27/93
W-5	1HV-8801B	T-ENG-93-11	03/27/93
W-5	1HV-8802A	T-ENG-93-11	04/12/93
W-5	1HV-8802B	T-ENG-94-13	09/29/94
W-5	1HV-8835	T-ENG-93-11	04/12/93
W-5	2HV-8801A	T-ENG-93-26	09/24/93
W-5	2HV-8801B	T-ENG-95-05	03/23/95
W-5	2HV-8802A	T-ENG-93-28	09/29/93
W-5	2HV-8802B	T-ENG-95-06	02/23/95
W-5	2HV-8835	T-ENG-95-06	03/23/95
W-6	1HV-8807A	T-ENG-94-17	10/05/94
W-6	1HV-8807B	T-ENG-94-17	10/05/94
W-6	1HV-8923A	T-ENG-94-17	10/05/94
W-6	1HV-8923B	T-ENG-94-08	09/30/94
W-6	1HV-8924	T-ENG-93-11	04/12/93

Group No.	Tag No.	Procedure No.	Test Date
W-6	2HV-8807A	T-ENG-95-08	04/03/95
W-6	2HV-8807B	T-ENG-95-08	04/03/95
W-6	2HV-8923A	T-ENG-95-08	03/02/95
W-6	2HV-8923B	T-ENG-95-12	03/19/95
W-6	2HV-8924	T-ENG-95-13	03/19/95
W-7	1HV-8806	T-ENG-94-08	09/30/94
W-7	2HV-8806	T-ENG-95-12	03/19/95
W-8	1HV-8716A	T-ENG-94-12	09/21/94
W-8	1HV-8716B	T-ENG-94-12	09/20/94
W-8	1HV-8804A	T-ENG-93-11	04/17/93
W-8	1HV-8804B	T-ENG-94-08	09/30/94
W-8	2HV-8716A	T-ENG-95-13	03/19/95
W-8	2HV-8716B	T-ENG-93-29	10/09/93
W-8	2HV-8804A	T-ENG-95-13	03/19/95
W-8	2HV-8804B	T-ENG-95-12	03/19/95

The Westinghouse differential pressure test data was originally evaluated based on the Westinghouse methodology which is discussed in detail in Section 3 of this document. The methodology is similar to the Industry Standard Equation, however, the methodology utilizes valve specific disc coefficients. The use of multiple disc coefficients makes it difficult to apply the available test data to valve groups in which in-situ differential pressure testing could not be performed at VEGP. In addition, due to the many uncertainties associated with performing in-situ differential pressure testing, the relatively minor differences in disc coefficients were not apparent based on the VEGP in-situ differential pressure test results.

Due to the factors outlined above the decision was made to revise the methodology being utilized to calculated the Westinghouse gate valve thrust requirements to a uniform methodology which would bound the Westinghouse methodology but which did not utilize valve specific disc coefficients. The EPRI NMAC equation was selected for this purpose utilizing a 0.55 friction coefficient for steam service and a 0.60 friction coefficient for water service. The differential pressure test evaluations were revised to reflect this new methodology.

Consideration was given to utilizing the EPRI PPP methodology for Westinghouse gate valves, however, the methodology had not yet been released when this work was being performed in the Spring of 1995. In addition, a preliminary review of the EPRI methodology indicated that it would be extremely difficult to utilize on a large scope of valves due to the overall complexity of the methodology and the fact that it was not computer based.

In performing the initial evaluation of the differential pressure test data, the test results were extrapolated to design-basis conditions utilizing linear extrapolation. The use of linear extrapolation was justified based on a review of the test reports issued by EPRI documenting the testing performed in conjunction with the Performance Prediction Program. The EPRI data indicated that as differential pressures increased, with a corresponding increase in contact stresses, friction coefficients tended to decrease. This trend was noted throughout the EPRI test data. In addition, the VEGP in-situ differential pressure test data for Westinghouse gate valves indicated that the friction coefficients tended to be lower for valves tested at relatively high differential pressures as opposed to valves test at relatively low differential pressures.

9.2.1 Westinghouse Group W-2A Gate Valves

This group is composed of eight 3 inch, 2035 lb, Westinghouse gate valves. The valves are located in the Chemical and Volume Control System (CVCS) and the Residual Heat Removal System (RHR). Differential pressure tests were performed on each valve in this group.

<u>Differential Pressure Test Overview</u>

1/2FV-0610 & 1/2FV-0611 - RHR Pump Miniflow

These valves are normally open to provide RHR pump miniflow. The valves have a safety function to close on increasing RHR pump flow and to re-open on decreasing RHR pump flow. All of the valves with the exception of 2FV-0611 were tested in the opening and closing direction. The valves opened and closed successfully at differential pressures approaching design-basis conditions and data was collected in each test.

When testing valve 2FV-0611 the closing stroke was inadvertently omitted from the test. Therefore, closing data for this valve was not available and a closing evaluation could not be performed.

1/2HV-8146 & 1/2HV-8147 - Normal and Alternate Charging to RCS

The function of these valves is alternated over the life of the plant with the normal charging valve open and the alternate charging valve closed. The valves have a safety function to open to provide an emergency boration flow path. The valves were tested in the opening direction only since these valves do not have an active safety function to close. Each of the valves opened successfully at the test differential pressures and data was collected in each test.

When the test results were evaluated for 2HV-8146 and 2HV-8147 a problem was identified relative to the differential pressure data recorded during these tests. It was determined that the differential pressures associated with these two tests could not be accurately determined from the available data, therefore, these tests were not evaluated further.

<u>Differential Pressure Test Data Evaluation</u>

The initial GL 89-10 thrust calculations for the Westinghouse Group W-2A valves were performed utilizing Westinghouse equations and Westinghouse supplied disc coefficients of 0.431 in the opening direction and 0.479 in the closing direction. The Westinghouse differential pressure data was originally evaluated based on this methodology. Disc coefficients were backed out of the test data and compared to the design disc coefficients to determine if the valves performance was bounded by the design methodology. Table 9-5 summarizes this evaluation for the group W-2A valves.

Table 9-5
Westinghouse Group W-2A Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Disc Coef. Open	Test Disc Coef. Open	Design DP Close psid	Test DP Close psid	Design Disc Coef. Close	Test Disc Coef. Close
1FV-0610	202	179	.431	.850	202	179	.479	.519
1FV-0611	202	192	.431	.596	202	189	.479	.366
1HV-8146	490	294	.431	.966	0	0	.479	N/A
1HV-8147	490	307	.431	.803	0	0	.479	N/A
2FV-0610	202	193	.431	.725	202	197	.479	.679
2FV-0611	202	192	.431	.568	202	0	.479	N/A
2HV-8146	490	N/A	.431	N/A	0	N/A	.431	N/A
2HV-8147	490	N/A	.431	N/A	0	N/A	.431	N/A

A preliminary review of the differential pressure test evaluation results indicated that the valves did not appear to be performing within the bounds of the Westinghouse methodology. In many cases the apparent opening and/or closing disc coefficient were greater than the Westinghouse specified disc coefficients.

To support the use of the NMAC methodology on the Westinghouse gate valves the differential pressure data was re-evaluated based on this methodology. Friction coefficients were backed out of the test data and compared to the design friction coefficients to determine if the valves performance was bounded by the NMAC methodology. Table 9-6 summarizes this evaluation for the group W-2A valves.

Table 9-6
Westinghouse Group W-2A Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Fric. Coef. Open	Test Fric. Coef. Open	Design DP Close psid	Test DP Close psid	Design Fric. Coef. Close	Test Fric. Coef. Close
1FV-0610	202	179	.6	1.23	202	179	.6	.67
1FV-0611	202	192	.6	.81	202	189	.6	.50
1HV-8146	490	294	.6	1.49	0	0	.6	N/A
1HV-8147	490	307	.6	1.21	0	0	.6	N/A
2FV-0610	202	193	.6	1.08	202	197	.6	.8
2FV-0611	202	192	.6	.83	202	0	.6	N/A
2HV-8146	490	N/A	.6	N/A	0	N/A	.6	N/A
2HV-8147	490	N/A	.6	N/A	0	N/A	.6	N/A

Here again, the results of the evaluation indicated that the valves did not appear to be performing within the bounds of the NMAC methodology. Many of the apparent opening and/or closing friction coefficients are higher than the friction coefficients being utilized in the design methodology.

To determine the reason for this apparent discrepancy between the design methodology and the test results, the test packages for these valves were reviewed in detail. The system configurations, the VOTES and differential pressure data as well as the overall use of this data in the evaluation process were all reviewed to determine if there was a reasonable explanation for the valves performance.

Valves 1/2HV-0610 and 1/2HV-0611 are 3 inch valves which were tested at differential pressures of less than 200 psid. Valves 1HV-8146 and 1HV-8147 are also 3 inch valves and were tested at differential pressures of approximately 300 psid. Due the size of these valves and the associated test pressures the total opening and closing loads are quite small. In the case of the 1/2HV-0610 and 1/2HV-0611 valves the maximum thrust after disc pullout ranges from approximately 1,100 lb to 1,600 lb and the thrust at flow cutoff ranges from approximately 1,300 lb to 1,900 lb. In the case of the 1HV-8146 and 1HV-8147 valves the maximum thrust after disk pullout ranges from approximately 1700 lb to 2400 lb. As a point of reference, the thrust calculations for each of these valves includes a 1500 lb load to account for packing and in many cases the total loads measured in the tests were less than the design packing load.

It is apparent that the loads associated with operating these valves are small and that the evaluation of the test results for these valves are extremely sensitive to even the smallest of errors. In evaluating the test data to determine the apparent friction coefficients, the total loads referenced above are adjusted to eliminate stem rejection loads and packing loads. This results in a differential pressure load which is the basis for determining the apparent friction coefficient. The actual differential pressure loads for these valves are in some cases well under 1000 lb.

The overall accuracy of the VOTES thrust measurement is a function of the valve specific calibration, but is generally on the order of ± 10%. To assume, however, that this is the overall accuracy of the thrust measurement associated with a differential pressure test would be very misleading. In reality, there are a host of potential uncertainties associated with differential pressure tests which can effect the overall accuracy of the test data and the subsequent evaluation of that data.

Fundamental to the use of the VOTES data acquisition system is the requirement that a zero load point be manually identified on each trace. This is a relatively simple task when performing static testing but becomes quite complex and subjective when dealing with differential pressure tests. In the case of a differential pressure test, a stem rejection load is always present which can make identifying an accurate zero extremely difficult. The criticality of identifying the zero accurately is of paramount importance when dealing with the relatively small loads associated with these valves.

Identifying the flow cutoff point and the maximum thrust after disc pullout are also largely subjective tasks with each test presenting its own unique difficulties in these areas. In some tests obvious plateaus are present which make the determination of these points relatively simple. However, in many tests these points are not so obvious and in those cases the points must be selected conservatively based on an analysis of the best available data. The use of accelerometers later in the test program substantially aided in the determination of these points, however, in many cases, the uncertainties associated with noise are sufficient to significantly effect the test results for these valves.

In summary, to the extent that an accurate zero can be determined and the relevant points on the opening and closing traces can be identified, the thrust measurement is accurate to within approximately ± 10%. However, errors in determining the zero or in determining the maximum thrust after disk pullout or the flow cutoff thrust, from the opening and closing traces respectively, can lead to errors of a much larger magnitude. For example, a 500 lb error is establishing a zero would result in an additional 50% error in the calculated friction coefficient for a valve with a differential pressure load of 1000 lb. A 500 lb error would be relatively insignificant on tests performed on larger valves or at higher differential pressures but on these small, low pressure valves it has a dramatic impact on the test results.

Conclusions

The test data for these valves was extrapolated to design-basis conditions and all of the valves are capable of performing their design-basis function in spite of the higher than design apparent friction coefficients. The fact that the apparent friction coefficients for these valves are not bounded by the NMAC methodology is not an indication that this methodology does not provide a conservative prediction of the loads associated with operating the Westinghouse EMD gate valve. Rather, the results of these tests reflect the limitations associated with performing in-situ differential pressure tests on small valves at relatively low differential pressures.

The apparent friction coefficients which were calculated for these valves are not considered to be credible. The loads associated with opening and closing these valves are so small that the sensitivity of the evaluation to the uncertainties inherent in the overall test process render the results meaningless. The results tend to be skewed due to the fact that the overall evaluation process is performed in a manner which ensures, to the extent possible, that uncertainties result in additional conservatism rather than reduced conservatism.

9.2.2 Westinghouse Group W-2B Gate Valves

This group is composed of eight 3 inch, 2035 lb, Westinghouse gate valves. The valves are located in the Reactor Coolant System (RCS) and the Chemical and Volume Control System (CVCS). Differential pressure tests were performed on a total of four valves in this group.

<u>Differential Pressure Test Overview</u>

1/2HV-8105 & 1/2HV-8106 - Charging Pump to RCS Isolation

These valves are normally open to provide charging flow to the regenerative heat exchanger and have a safety function to close on an SI signal to isolate normal charging. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching or exceeding the design-basis differential pressure and valid data was collected in each test.

<u>Differential Pressure Test Data Evaluation</u>

The initial GL 89-10 thrust calculations for the Westinghouse Group W-2B valves were performed utilizing Westinghouse equations and Westinghouse supplied disc coefficients of 0.365 in the opening direction and 0.443 in the closing direction. The Westinghouse differential pressure data was originally evaluated based on this methodology. Disc coefficients were backed out of the test data and compared to the design disc coefficients to determine if the valves performance was bounded by the design methodology. Table 9-7 summarizes this evaluation for the group W-2B valves.

The results of the evaluation indicate that the valves are performing within the bounds of the Westinghouse methodology with the exception of valve 1HV-8105. Valve 1HV-8105 has an apparent opening disc coefficient which is higher than the Westinghouse specified disc coefficient.

To support the use of the NMAC methodology on the Westinghouse valves the differential pressure data was re-evaluated based on this methodology. Friction coefficients were backed out of the test data and compared to the design friction coefficients to determine if the valves performance was bounded by the NMAC methodology. Table 9-8 summarizes this evaluation for the group W-2B valves.

Table 9-7
Westinghouse Group W-2B Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Disc Coef. Open	Test Disc Coef. Open	Design DP Close psid	Test DP Close psid	Design Disc Coef. Close	Test Disc Coef. Close
1HV-8105	460	2441	.365	.466	2681	2441	.443	.411
1HV-8106	460	2611	.365	.213	2681	1216	.443	.211
2HV-8105	460	2669	.365	.232	2681	2669	.443	.129
2HV-8106	460	2650	.365	.250	2681	2650	.443	.386

Table 9-8
Westinghouse Group W-2B Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Fric. Coef. Open	Test Fric. Coef. Open	Design DP Close psid	Test DP Close psid	Design Fric. Coef. Close	Test Fric. Coef. Close
1HV-8105	460	2441	.6	.66	2681	2441	.6	.50
1HV-8106	460	2611	.6	.40	2681	1216	.6	.27
2HV-8105	460	2669	.6	.32	2681	2669	.6	.16
2HV-8106	460	2650	.6	.34	2681	2650	.6	.47

The results of the re-evaluation indicate that the valves are performing within the bounds of the NMAC methodology with the exception of the opening friction coefficient for valve 1HV-8105. The apparent opening friction coefficient for this valve is approximately 10% higher than the value utilized in the design calculations. However, given the overall accuracy of the differential pressure test process, this apparent discrepancy with respect to one valve is not considered to be significant and does not indicate a problem with this valve group. In addition, these valves were tested at differential pressures which were approximately 5 times the design differential pressures. All of the closing friction coefficients are bounded by the methodology as are the opening friction coefficients for the remaining valves.

<u>Conclusions</u>

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that the NMAC equation utilizing a 0.6 friction coefficient provides a reasonable prediction of the thrust requirements for Westinghouse 3 inch, 2035 lb, gate valves. The apparent opening friction coefficient for one valve is approximately 10% above the design value, however, this difference is within the overall accuracy of the test process.

9.2.3 Westinghouse Group W-4 Gate Valves

This group is composed of four 4 inch, 900 lb, Westinghouse gate valves. The valves are located in the Safety Injection System (SI). Differential pressure tests were performed on each valve in this group.

Differential Pressure Test Overview

1/2HV-8821A/B - SI Pump to RCS Cold Leg Isolation

These valves are normally open to provide an SI flow path to the cold-legs and have a safety function to close when transferring to hot-leg recirculation. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressure approaching design-basis conditions and valid data was collected in each test.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Westinghouse Group W-4 valves were performed utilizing Westinghouse equations and Westinghouse supplied disc coefficients of 0.350 in the opening direction and 0.394 in the closing direction. The Westinghouse differential pressure data was originally evaluated based on this methodology. Disc coefficients were backed out of the test data and compared to the design disc coefficients to determine if the valves performance was bounded by the design methodology. Table 9-9 summarizes this evaluation for the group W-4 valves.

Table 9-9
Westinghouse Group W-4 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Disc Coef. Open	Test Disc Coef. Open	Design DP Close psid	Test DP Close psid	Design Disc Coef. Close	Test Disc Coef. Close
1HV-8821A	1750	1460	.350	.196	1750	1460	.394	.255
1HV-8821B	1750	1482	.350	.137	1750	1490	.394	.076
2HV-8821A	1750	1496	.350	.091	1750	1504	.394	.164
2HV-8821B	1750	1510	.350	.177	1750	1464	.394	.171

The results of the evaluation indicate that the valves are performing within the bounds of the Westinghouse methodology. Both the opening and closing disc coefficients for the tested valves are less than the Westinghouse specified disc coefficients.

To support the use of the NMAC methodology on the Westinghouse valves the differential pressure data was re-evaluated based on this methodology. Friction coefficients were backed out of the test data and compared to the design friction coefficients to determine if the valves performance was bounded by the NMAC methodology. Table 9-10 summarizes this evaluation for the group W-4 valves.

Table 9-10
Westinghouse Group W-4 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Fric. Coef. Open	Test Fric. Coef Open	Design DP Close psid	Test DP Close psid	Design Fric. Coef. Close	Test Fric. Coef. Close
1HV-8821A	1750	1460	.6	.26	1750	1460	.6	.32
1HV-8821B	1750	1482	.6	.18	1750	1490	.6	.10
2HV-8821A	1750	1496	.6	.12	1750	1504	.6	.21
2HV-8821B	1750	1510	.6	.24	1750	1464	.6	.22

The results of the re-evaluation indicate that the valves are performing within the bounds of the NMAC methodology. Both the opening and closing friction coefficients for the tested valves are less than the design friction coefficients.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it can be concluded that the NMAC equation utilizing a 0.6 friction coefficient conservatively predicts thrust requirements for Westinghouse 4 inch, 900 lb, gate valves.

9.2.4 Westinghouse Group W-5 Gate Valves

This group is composed of sixteen 4 inch, 1525 lb, Westinghouse gate valves. The valves are located in the Safety Injection (SI) system and the Chemical and Volume Control System (CVCS). Differential pressure tests were performed on a total of ten valves in this group.

Differential Pressure Test Overview

1/2HV-8801A/B - BIT Discharge Isolation

These valves are normally closed and have a safety function to open on an SI signal to provide a flow path from the CCPs to the RCS cold legs. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching or exceeding design-basis conditions and valid data was collected in each test.

1/2HV-8802A/B - SI Pump to RCS Hot Leg Isolation

These valves are normally closed and have a safety function to open to provide an SI flow path to the hot-legs when transferring to hot leg recirculation. The valves were tested in the opening and closing direction. Each valve opened successfully against differential pressures approaching design-basis conditions and valid data was collected in each test.

The 1/2HV-8802A/B valves do not have an active safety function to close and the design-basis closing differential pressure for these valves is 0 psid. In an effort to collect generic data on Westinghouse valves, these valves were closed with the system configured for a test designed to simulate the opening design-basis differential pressure of 1735 psid. Valves 1HV-8802B and 2HV-8802A/B did not attain flow cutoff at the test differential pressure prior to the operation of the torque switches, therefore, the closing data for these valves was not valid and was not evaluated.

In addition, the test sequence for valve 2HV-8802A required that the valve be closed under a differential pressure and then re-opened under the differential pressure. Since the valve did not make hard seat contact on the closing stroke the opening data was determined to be invalid and was not evaluated. The test sequence for valves 1HV-8802B and 2HV-8802B required that the valve be closed statically and then re-opened under a differential pressure. The opening data for these valves was valid and was evaluated.

1/2HV-8835 - SI Pump to Cold Leg Isolation

These valves are normally open to provide an SI flow path to the RCS cold-legs and have a safety function to close when transferring to hot-leg recirculation. The valves were tested in the opening and closing direction. Each valve opened successfully against differential pressures approaching design-basis conditions.

The 1/2HV-8835 valves do not have an active safety function to close and the design-basis closing differential pressure for these valves is 0 psid. In an effort to collect generic data on Westinghouse valves, these valves were closed with the system configured for a test designed to simulate the opening design-basis differential pressure of 1728 psid. The valves did not attain flow cutoff at the test differential pressure prior to the operation of the torque switches, therefore, the closing data was not valid and was not evaluated.

In addition, the test sequence for valve 1HV-8835 required that the valve be closed under a differential pressure and then re-opened under the differential pressure. Since the valve did not make hard seat contact on the closing stroke the opening data was determined to be invalid and was not evaluated. The test sequence for valve 2HV-8835 required that the valve be closed statically and then re-opened under a differential pressure. The opening data for this valve was valid and was evaluated.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Westinghouse Group W-5 valves were performed utilizing Westinghouse equations and Westinghouse supplied disc coefficients of 0.350 in the opening direction and 0.394 in the closing direction. The Westinghouse differential pressure data was originally evaluated based on this methodology. Disc coefficients were backed out of the test data and compared to the design disc coefficients to determine if the valves performance was bounded by the design methodology. Table 9-11 summarizes this evaluation for the group W-5 valves.

The results of the evaluation indicate that the valves are generally performing within the bounds of the Westinghouse methodology. The apparent opening disk coefficient for valve 2HV-8802B is slightly higher than the design disk coefficient.

To support the use of the NMAC methodology on the Westinghouse valves the differential pressure data was re-evaluated based on this methodology. Friction coefficients were backed out of the test data and compared to the design friction coefficients to determine if the valves performance was bounded by the NMAC methodology. Table 9-12 summarizes this evaluation for the group W-5 valves.

Table 9-11
Westinghouse Group W-5 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Disc Coef. Open	Test Disc Coef. Open	Design DP Close psid	Test DP Close psid	Design Disc Coef. Close	Test Disc Coef. Close
1HV-8801A	2674	2688	.350	.193	483	2736	.394	.148
1HV-8801B	2674	2700	.350	.183	483	2676	.394	.115
1HV-8802A	1735	1474	.350	.188	0	1474	.394	.267
1HV-8802B	1735	1475	.350	.346	0	1475	.394	N/A
1HV-8835	1728	1500	.350	N/A	0	1500	.394	N/A
2HV-8801A	2674	2637	.350	.235	483	2637	.394	.214
2HV-8801B	2674	1871	.350	.211	483	2017	.394	.142
2HV-8802A	1735	1392	.350	N/A	0	1536	.394	N/A
2HV-8802B	1735	1497	.350	.369	0	1347	.394	N/A
2HV-8835	1728	1468	.350	.343	0	1302	.394	N/A

Table 9-12
Westinghouse Group W-5 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Fric. Coef. Open	Test Fric. Coef Open	Design DP Close psid	Test DP Close psid	Design Fric. Coef. Close	Test Fric. Coef. Close
1HV-8801A	2674	2688	.6	.26	483	2736	.6	.19
1HV-8801B	2674	2700	.6	.25	483	2676	.6	.15
1HV-8802A	1735	1474	.6	.25	0	1474	.6	.33
1HV-8802B	1735	1475	.6	.48	0	1475	.6	N/A
1HV-8835	1728	1500	.6	N/A	0	1500	.6	N/A
2HV-8801A	2674	2637	.6	.32	483	2637	.6	.27
2HV-8801B	2674	1871	.6	.28	483	2017	.6	.18
2HV-8802A	1735	1392	.6	N/A	0	1536	.6	N/A
2HV-8802B	1735	1497	.6	.51	0	1347	.6	N/A
2HV-8835	1728	1468	.6	.47	0	1302	.6	N/A

The results of the re-evaluation indicate that the valves are performing within the bounds of the NMAC methodology. Both the opening and closing friction coefficients for the tested valves are less than the design friction coefficients.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that the NMAC equation utilizing a 0.6 friction coefficient conservatively predicts thrust requirements for Westinghouse 4 inch, 1525 lb, gate valves.

9.2.5 Westinghouse Group W-6 Gate Valves

This group is composed of fourteen, 6 inch, 150 lb, Westinghouse gate valves. The valves are located in the Safety Injection (SI) system and the Chemical and Volume Control System (CVCS). Differential pressure tests were performed on a total of ten valves in this group.

<u>Differential Pressure Test Overview</u>

1/2HV-8807A/B - RHR to SI Pump Suction Isolation

These valves are normally closed and have a safety function to open when transferring to recirculation. The valves have a safety function to close to isolate a passive failure leak of less than 50 gpm. These valves are in the suction path to the SI pumps, therefore, to perform a differential pressure test under flow conditions without risking damage to the pumps, the internals were removed from a check valve to provide a return flow path to the RWST. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

1/2HV-8923A/B - SI Pump Suction Isolation

These valves are normally open to provide a suction flow path to the SI pumps. The valves have a safety function to close to isolate a passive failure leak of less than 50 gpm. These valves are in the suction path to the SI pumps, therefore, to perform a differential pressure test under flow conditions without risking damage to the pumps, the internals were removed from a check valve to provide a return flow path to the RWST. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

1/2HV-8924 - RHR to SI Pump Suction Isolation

These valves are normally open to provide a flow path between the RHR pump discharge and the SI pumps. The valves have a safety function to close to isolate a passive failure leak of less than 50 gpm. These valves are in the suction path to the SI pumps, therefore, to perform a meaningful differential pressure test under flow conditions without risking damage to the pumps, the internals must be removed from a check valve to provide a return flow path to the RWST. The valves were tested in the opening and closing direction and each of the valves opened and closed successfully at the test differential pressure. However, valve 1HV-8924 was tested in a system configuration which did not involve the disassemble of a check valve to provide a return flow path. This test resulted in a differential pressure of much less than 50% of the design-basis differential pressure. Due to the low differential pressure the data for this valve was considered invalid and was not evaluated. Valve 2HV-8924 was tested in the configuration with the check valve disassembled with a test differential pressure approaching design-basis conditions and valid data was collected in this test.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Westinghouse Group W-6 valves were performed utilizing Westinghouse equations and Westinghouse supplied disc coefficients of 0.368 in the opening direction and 0.433 in the closing direction. The Westinghouse differential pressure data was originally evaluated based on this methodology. Disc coefficients were backed out of the test data and compared to the design disc coefficients to determine if the valves performance was bounded by the design methodology. Table 9-13 summarizes this evaluation for the group W-6 valves.

The results of the evaluation indicate that the valves are generally performing within the bounds of the Westinghouse methodology. The apparent disk coefficients for valves 1HV-8923B and 2HV-8807A are slightly higher than the design disk coefficients.

To support the use of the NMAC methodology on the Westinghouse valves the differential pressure data was re-evaluated based on this methodology. Friction coefficients were backed out of the test data and compared to the design friction coefficients to determine if the valves performance was bounded by the NMAC methodology. Table 9-14 summarizes this evaluation for the group W-6 valves.

Westinghouse Group W-6 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Disc Coef. Open	Test Disc Coef. Open	Design DP Close psid	Test DP Close psid	Design Disc Coef. Close	Test Disc Coef. Close
1HV-8807A	220	196	.368	.290	220	174	.433	.209
1HV-8807B	220	171	.368	.225	220	174	.433	.264
1HV-8923A	220	161	.368	.268	220	172	.433	.259
1HV-8923B	220	158	.368	.414	220	173	.433	.273
1HV-8924	0	29	.368	N/A	220	29	.433	N/A
2HV-8807A	220	179	.368	.391	220	174	.433	.367
2HV-8807B	220	163	.368	.314	220	172	.433	.235
2HV-8923A	220	163	.368	.345	220	174	.433	.221
2HV-8923B	220	156	.368	.327	220	168	.433	.273
2HV-8924	0	161	.368	.331	220	171	.433	.156

Table 9-14

Westinghouse Group W-6 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Fric. Coef. Open	Test Fric. Coef. Open	Design DP Close psid	Test DP Close psid	Design Fric. Coef Close	Test Fric. Coef. Close
1HV-8807A	220	196	.6	.38	220	174	.6	.28
1HV-8807B	220	171	.6	.29	220	174	.6	.34
1HV-8923A	220	161	.6	.35	220	172	.6	.33
1HV-8923B	220	158	.6	.56	220	173	.6	.34
1HV-8924	0	29	.6	N/A	220	29	.6	N/A
2HV-8807A	220	179	.6	.54	220	174	.6	.44
2HV-8807B	220	163	.6	.43	220	172	.6	.29
2HV-8923A	220	163	.6	.47	220	174	.6	.27
2HV-8923B	220	156	.6	.37	220	168	.6	.33
2HV-8924	0	161	.6	.45	220	171	.6	.19

The results of the re-evaluation indicate that the valves are performing within the bounds of the NMAC methodology. Both the opening and closing friction coefficients for the tested valves are less than the design friction coefficients.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it can be concluded that the NMAC equation utilizing a 0.6 friction coefficient conservatively predicts thrust requirements for Westinghouse 6 inch, 150 lb, gate valves.

9.2.6 Westinghouse Group W-7 Gate Valves

This group is composed of six, 8 inch, 150 lb, Westinghouse gate valves. The valves are located in the Safety Injection (SI) system and the Chemical and Volume Control System (CVCS). Differential pressure tests were performed on a total of two valves in this group.

Differential Pressure Test Overview

1/2HV-8806 - RWST to SI Pump Suction

These valves are normally open to provide a suction flow path from the RWST to the SI pumps. The valves have a safety function to close when transferring to recirculation. These valves are in the suction path to the SI pumps, therefore, to perform a differential pressure test under flow conditions without risking damage to the pumps, the internals were removed from a check valve to provide a return flow path to the RWST. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Westinghouse Group W-7 valves were performed utilizing Westinghouse equations and Westinghouse supplied disc coefficients of 0.361 in the opening direction and 0.424 in the closing direction. The Westinghouse differential pressure data was originally evaluated based on this methodology. Disc coefficients were backed out of the test data and compared to the design disc coefficients to determine if the valves performance was bounded by the design methodology. Table 9-15 summarizes this evaluation for the group W-7 valves.

Table 9-15
Westinghouse Group W-7 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Disc Coef. Open	Test Disc Coef. Open	Design DP Close psid	Test DP Close psid	Design Disc Coef. Close	Test Disc Coef. Close
1HV-8806	210	186	.361	.255	210	192	.424	.251
2HV-8806	210	154	.361	.276	210	165	.424	.197

The results of the evaluation indicate that the tested valves are performing within the bounds of the Westinghouse methodology. Both the opening and closing disc coefficients for the tested valves are less than the design disc coefficients.

To support the use of the NMAC methodology on the Westinghouse valves the differential pressure data was re-evaluated based on this methodology. Friction coefficients were backed out of the test data and compared to the design friction coefficients to determine if the valves performance was bounded by the NMAC methodology. Table 9-16 summarizes this evaluation for the group W-7 valves.

Table 9-16
Westinghouse Group W-7 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Fric. Coef. Open	Test Fric. Coef. Open	Design DP Close psid	Test DP Close psid	Design Fric. Coef. Close	Test Fric. Coef. Close
1HV-8806	210	186	.6	.34	210	192	.6	.31
2HV-8806	210	154	.6	.36	210	165	.6	.25

The results of the re-evaluation indicate that the valves are performing within the bounds of the NMAC methodology. Both the opening and closing friction coefficients for the tested valves are less than the design friction coefficients.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it can be concluded that the NMAC equation utilizing a 0.6 friction coefficient conservatively predicts thrust requirements for Westinghouse 8 inch, 150 lb, gate valves.

9.2.7 Westinghouse Group W-8 Gate Valves

This group is composed of twelve, 8 inch, 316 lb, Westinghouse gate valves. The valves are located in the Residual Heat Removal (RHR) system and the Containment Spray (CS) system. Differential pressure tests were performed on a total of eight valves in this group.

Differential Pressure Test Overview

1/2HV-8716A/B - RHR to RCS Hot Leg Isolation

These valves are normally open to crosstie the discharge of the two trains of the RHR system. The valves have a safety function to close when transferring to cold-leg recirculation and to open when transferring to hot-leg recirculation. The valves were tested in the opening and closing direction. The valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

1/2HV-8804A/B - RHR to CCP or SI Pump Suction

These valves are normally closed and have a safety function to open to establish a flow path between the discharge of the RHR pumps and the suction of the CCPs or SI pumps when transferring to recirculation. The valves have a safety function to close to isolate a passive failure leak of less than 50 gpm. These valves are in the suction path to the SI or CCP pumps, therefore, to perform a meaningful differential pressure test under flow conditions without risking damage to the pumps the internals must be removed from a check valve to provide a return flow path to the RWST. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at the test differential pressure. However, valve 1HV-8804A was tested in a system configuration which did not involve the disassemble of a check valve to provide a return flow path. This test resulted in a differential pressure of much less than 50% of the design-basis differential pressure. Due to the low differential pressure the data for this valve was considered invalid and was not evaluated. Valves 1HV-8804B and 2HV-8804A/B were tested in the configuration with the check valve disassembled with a test differential pressures approaching design-basis conditions and valid data was collected in each test.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Westinghouse Group W-8 valves were performed utilizing Westinghouse equations and Westinghouse supplied disc coefficients of 0.361 in the opening direction and 0.424 in the closing direction. The Westinghouse differential pressure data was originally evaluated based on this methodology. Disc coefficients were backed out of the test data and compared to the design disc coefficients to determine if the valves performance was bounded by the design methodology. Table 9-17 summarizes this evaluation for the group W-8 valves.

Table 9-17
Westinghouse Group W-8 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Disc Coef. Open	Test Disc. Coef. Open	Design DP Close psid	Test DP Close psid	Design Disc. Coef. Close	Test Disc. Coef. Close
1HV-8716A	253	193	.361	.240	273	223	.424	.233
1HV-8716B	253	198	.361	.360	273	223	.424	.256
1HV-8804A	425	21	.361	N/A	267	31	.424	N/A
1HV-8804B	425	169	.361	.289	267	186	.424	.426
2HV-8716A	253	161	.361	.472	273	196	.424	.328
2HV-8716B	253	192	.361	.335	273	215	.424	.289
2HV-8804A	425	155	.361	.422	267	167	.424	.520
2HV-8804B	425	165	.361	.593	267	164	.424	.529

A preliminary review of the differential pressure test data indicated that a number of the valves did not appear to be performing within the bounds of the Westinghouse methodology. In a number of cases the apparent opening and/or closing disc coefficients were greater than the Westinghouse specified disc coefficients.

To support the use of the NMAC methodology on the Westinghouse valves the differential pressure test data was re-evaluated based on this methodology. Friction coefficients were backed out of the test data and compared to the design friction coefficients to determine if the valves performance was bounded by the NMAC methodology. Table 9-18 summarizes this evaluation for the group W-8 valves.

Table 9-18
Westinghouse Group W-8 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Fric. Coef. Open	Test Fric. Coef. Open	Design DP Close psid	Test DP Close psid	Design Fric. Coef. Close	Test Fric. Coef. Close
1HV-8716A	253	193	.6	.3	273	223	.6	.29
1HV-8716B	253	198	.6	.48	273	223	.6	.32
1HV-8804A	425	21	.6	N/A	267	31	.6	N/A
1HV-8804B	425	169	.6	.39	267	186	.6	.51
2HV-8716A	253	161	.6	.65	273	196	.6	.40
2HV-8716B	253	192	.6	.45	273	215	.6	.35
2HV-8804A	425	155	.6	.58	267	167	.6	.61
2HV-8804B	425	165	.6	.84	267	164	.6	.62

The results of the re-evaluation indicate that the valves are generally performing within the bounds of the NMAC methodology. However, several valves have apparent opening and/or closing friction coefficients which are higher than the values utilized in the calculations. Valves 2HV-8804A/B have apparent closing friction coefficients of 0.61 and 0.62 and valves 2HV-8716A and 2HV-8804B have apparent opening disc coefficients of 0.65 and 0.84. With the exception of the opening friction coefficient for valve 2HV-8804B the differences in the apparent test friction coefficients and the design friction coefficients are less than 10% and are not considered significant.

The apparent opening friction coefficient for valve 2HV-8804B is approximately 40% above the value utilized in the calculations. The differential pressure test data for this valve and the subsequent detailed evaluation of the data were reviewed in an effort to explain the higher than expected friction coefficient. Several anomalies were noted in this review. The average running load was substantially higher in the dynamic test than in the static test. Typically, the running load in the opening direction would be expected to be lower in the dynamic test since the stem rejection force assists in opening the valve. In addition, the friction coefficient in the closing direction was significantly lower than in the opening direction. The opening and closing friction coefficients are generally expected to be similar in both directions although apparent differences have been noted on other valves tested at VEGP. In summary, it appears that the results of this particular test are inconclusive. Several anomalies were noted which raise questions as to the overall accuracy of the evaluation, although none are significant enough to warrant disregarding the data completely.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP is was concluded that the NMAC equation utilizing a 0.6 friction coefficient provides a reasonable prediction of the thrust requirements for Westinghouse 8 inch, 316 lb, gate valves. The apparent friction coefficients for several of the valves were higher than the design value, however, with the exception of one valve, the differences were within the overall accuracy of the test process.

The opening friction coefficient for valve 2HV-8804B appeared high based on the results of the evaluation. However, a detailed review of the differential pressure test evaluation for this valve identified inconsistencies which may indicate problems associated with the test data collected in conjunction with this test. Considering the fact that a total of eight valves in this group were differential pressure tested and that the apparent opening friction coefficient for this valve was the only friction coefficient which was significantly higher than design, the results of this test are not considered significant. The test data for this valve was extrapolated to design-basis conditions, and the valve is capable of performing its design-basis function in spite of the higher than design apparent friction coefficient. This valve will be re-evaluated and appropriate actions taken to resolve the discrepancy prior to startup of Unit 2 following the refueling outage scheduled for the Fall of 1996.

9.3 Fisher Globe Valves

A total of eight Fisher globe valves, representing two of the three Fisher valve groups, were differential pressure tested. The valves which were tested are identified in Table 9-19.

Table 9-19
Fisher Globe Valve
Differential Pressure Test Scope

Group No.	Tag No.	Procedure No.	Test Date
FG-2	1FV-5154	T-ENG-93-02	04/21/93
FG-2	1FV-5155	T-ENG-93-03	04/21/93
FG-2	2FV-5154	T-ENG-95-09	03/11/95
FG-2	2FV-5155	T-ENG-95-10	03/27/95
FG-3	1HV-5137	T-ENG-94-14	10/13/94
FG-3	1HV-5139	T-ENG-94-14	10/13/94
FG-3	2HV-5120	T-ENG-93-31	10/19/93
FG-3	2HV-5122	T-ENG-93-31	10/19/93

The Fisher globe valves which were differential pressure tested at VEGP include two distinctly different globe valve designs. The valves in group FG-2 are of the unbalanced disk design and the valves in group FG-3 are of the balanced disk design. The thrust requirements for each of these valve types were determined by Fisher utilizing proprietary methodology which is applicable to the specific valve type. The evaluation of the VEGP test results based on this methodology is described in more detail in the following sections.

9.3.1 Fisher Group FG-2 Globe Valves

This group is composed of four, 4 inch, 900 lb, unbalanced disk Fisher globe valves. The valves are located in the Auxiliary Feedwater (AFW) system. Differential pressure tests were performed on each valve in this group.

<u>Differential Pressure Test Overview</u>

1/2FV-5154 & 1/2FV-5155 - Turbine-Driven AFW Pump Miniflow

These valves are normally open to provide turbine-driven AFW pump miniflow to the Condensate Storage Tank (CST) and have a safety function to close on high AFW pump flow and to re-open on low AFW pump flow. These valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

<u>Differential Pressure Test Data Evaluation</u>

The GL 89-10 thrust calculations for the Fisher Group FG-2 valves were performed by the manufacturer utilizing proprietary methodology. Since the detailed methodology was not available the test data was evaluated against the design calculations by comparing the total measured test thrust to the calculated required thrust. The test thrust was adjusted to the design-basis conditions by multiplying the measured thrust by the ratio of the design differential pressure to the test differential pressure. This methodology of extrapolation is conservative because it includes static loads which do not vary as a function of differential pressure. Table 9-20 and 9-21 summarizes this evaluation for the group FG-2 valves.

Table 9-20
Fisher Group FG-2 Valves
DP Test Data Evaluation Summary - Opening

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Thrust Open lb	Test Thrust Open lb	Extrapolated Test Thrust Open lb	Extrapolated Thrust/ Design Thrust
1FV-5154	1715	1560	4641	1792	1970	42%
1FV-5155	1715	1488	4641	68	78	2%
2FV-5154	1715	1648	4641	213	222	5%
2FV-5155	1715	1664	4641	1166	1202	26%

Table 9-21
Fisher Group FG-2 Valves
DP Test Data Evaluation Summary - Closing

Valve Tag No.	Design DP Close psid	Test DP Close psid	Design Thrust Close lb	Test Thrust Close lb	Extrapolated Test Thrust Close lb	Extrapolated Thrust/ Design Thrust
1FV-5154	1715	1480	4641	2079	2409	52%
1FV-5155	1715	1548	4641	999	1107	24%
2FV-5154	1715	1648	4641	1487	1547	33%
2FV-5155	1715	1664	4641	2186	2253	49%

The results of the evaluation indicate that the tested valves performed within the bounds of the design methodology. The extrapolated opening and closing thrusts are less than the design thrusts predicted by the Fisher methodology.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that the Fisher methodology conservatively predicts thrust requirements for Fisher 2 inch, 900 lb, unbalanced disk globe valves.

9.3.2 Fisher Group FG-3 Globe Valves

This group is composed of sixteen, 4 inch, 900 lb, balanced disk Fisher globe valves. These valves are located in the Auxiliary Feedwater System (AFW). Differential pressure tests were performed on a total of four valves in this group.

Differential Pressure Test Overview

1HV-5137 and 1HV-5139 - Motor-Driven AFW Pump Discharge

These valves are normally open to provide a flow path from the motor-driven AFW pumps to the steam generators and the valves have a safety function to open and close as required to control flow to the steam generators. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

2HV-5120 and 2HV-5122 - Turbine-Driven AFW Pump Discharge

These valves are normally open to provide a flow path from the turbine-driven AFW pump to the steam generators and the valves have a safety function to open and close as required to control flow to the steam generators. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at the test differential pressures.

When the test results for these valves were evaluated it was determined that the differential pressures at flow isolation were much lower than expected. This was due to the fact that a substantial amount of steam generator pressure was required to operate the turbine-driven AFW system. The steam generator pressure was present on the downstream side of the discharge valves and thereby limited the differential pressures attainable in this test. The differential pressures measured in the tests were less than 50% of the design-basis differential pressure, however, since these valves are of the balanced disk design the data was evaluated. The thrust requirements for balanced disk globe valves do not vary significantly as a function of differential pressure, therefore, the data collect at the reduced differential pressures was considered to be valid and indicative of the valves performance at design-basis conditions.

Differential Pressure Test Data Evaluation

The GL 89-10 thrust calculations for the Fisher Group FG-3 valves were performed by the manufacturer utilizing proprietary methodology. Since the detailed methodology was not available the test data was evaluated against the design calculations by comparing the total measured test thrust to the calculated required thrust. It was not necessary to extrapolate the test thrust values to design-basis conditions because these are balanced disk globe valves and the thrust requirements do not vary as a function of differential pressure. The actual thrusts measured at the test conditions are representative of the thrust requirements at design-basis conditions. Table 9-22 and 9-23 summarize this evaluation for the group FG-3 valves.

Table 9-22
Fisher Group FG-3 Valves
DP Test Data Evaluation Summary - Opening

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design Thrust Open lb	Test Thrust Open lb	Test/ Design Open
1HV-5137	1767	1368	3860	716	19%
1HV-5139	1767	1384	3860	1425	37%
2HV-5120	1776	480	3860	2133	55%
2HV-5122	1776	480	3860	1780	46%

Table 9-23
Fisher Group FG-3 Valves
DP Test Data Evaluation Summary - Closing

Valve Tag No.	Design DP Close psid	Test DP Close psid	Design Thrust Close psid	Test Thrust Close psid	Test/ Design Close psid
1HV-5137	1767	1528	3860	2901	75%
1HV-5139	1767	1608	3860	3075	80%
2HV-5120	1776	560	3860	3316	86%
2HV-5122	1776	700	3860	2755	71%

The results of the evaluation indicate that the tested valves are performing within the bounds of the design methodology. The opening and closing thrusts are less than the design thrusts predicted by the Fisher methodology.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that the Fisher methodology conservatively predicts thrust requirements for Fisher 4 inch, 900 lb, balanced disk globe valves.

9.4 Velan Globe Valves

A total of 22 Velan globe valves, representing each of the three groups of Velan valves, were differential pressure tested. The valves which were tested are identified in Table 9-24.

Table 9-24
Velan Globe Valve
Differential Pressure Test Scope

Group No.	Tag No.	Procedure No.	Test Date
V-1	1HV-8116	T-ENG-93-10	04/20/93
V-1	2HV-8116	T-ENG-95-07	03/20/95
V-2	1HV-8103B	T-ENG-94-11	10/11/94
V-2	1HV-8103C	T-ENG-94-11	10/11/94
V-2	1HV-8920	T-ENG-93-12	04/13/93
V-2	2HV-8814	T-ENG-93-28	09/23/93
V-3	1HV-8110	T-ENG-94-06	10/07/94
V-3	1HV-8111A	T-ENG-93-10	04/20/93
V-3	1HV-8111B	T-ENG-94-06	10/07/94
V-3	1HV-8508A	T-ENG-94-10	09/23/94
V-3	1HV-8508B	T-ENG-94-09	10/06/94
V-3	1HV-8509A	T-ENG-94-09	10/06/94
V-3	1HV-8509B	T-ENG-94-10	09/23/94
V-3	1HV-8813	T-ENG-93-12	04/13/93
V-3	2HV-8110	T-ENG-93-26	09/24/93
V-3	2HV-8111A	T-ENG-95-11	03/20/95
V-3	2HV-8111B	T-ENG-93-30	09/30/93
V-3	2HV-8508A	T-ENG-95-11	03/20/95
V-3	2HV-8508B	T-ENG-95-07	03/20/95
V-3	2HV-8509A	T-ENG-95-07	03/20/95
V-3	2HV-8509B	T-ENG-95-11	03/20/95
V-3	2HV-8813	T-ENG-93-27	09/29/93

The Velan globe valve differential pressure tests were originally evaluated based on the Industry Standard Equation with a 1.1 valve factor. In reviewing the test data it was determined that in certain cases the test data was not bounded by this methodology. Therefore, the decision was made to re-evaluate the Velan globe valves utilizing the EPRI PPP methodology.

The EPRI PPP methodology was developed based on an extensive analytical evaluation of the globe valve and was subsequently validated based on flow loop testing. The large scope of Velan globe valves differential pressure test at VEGP served to further validate the methodology. The differential pressure test evaluations were revised to reflect this new methodology.

9.4.1 Velan Group V-1 Globe Valves

This group is composed of two, 1 inch, 1500 lb Velan globe valves. The valves are located in the Chemical Volume and Control System (CVCS). Differential pressure tests were performed on each valve in this group.

Differential Pressure Test Overview

1/2HV-8116 - CCP A Discharge

These valves are normally closed and receive a close signal on an SI signal. The valves have a safety function to open to provide a safety grade charging path on a loss of instrument air. There is a normally closed air-operated valve located upstream of each of these valves, therefore, the design-basis closing differential pressure is zero. During the differential pressure tests the valves were tested in both the opening and closing direction at a differential pressure approaching the design-basis opening differential pressure. Each of the valves opened and closed successfully and valid data was collected in each test.

These valves are unique with respect to the scope of globe valves included in the VEGP GL 89-10 program in that they are installed in a flow over the seat configuration. In this configuration flow assists in closing the valves and resists the opening of the valves. All of the remaining globe valves in the GL 89-10 program are installed in a flow under the seat configuration.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Velan Group V-1 valves were performed utilizing the industry standard equation with a 1.1 valve factor. The Velan differential pressure test data was originally evaluated based on this methodology. A valve factor was backed out of the test data and compared to the design valve factor to determine if the valves performance was bounded by the design methodology. Table 9-25 summarizes this evaluation for the group V-1 valves.

Table 9-25
Velan Group V-1 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design VF Open	Test VF Open	Design DP Close psid	Test DP Close psid	Design VF Close	Test VF Close
1HV-8116	2674	2490	1.1	0.95	0	2490	1.1	0.47
2HV-8116	2674	2637	1.1	0.58	0	2637	1.1	0.33

The results of the evaluation indicate that these valves are performing within the bounds of the design methodology. Both the opening and closing valve factors for the tested valves are less than the design valve factors.

To support the use of the EPRI methodology on the VEGP Velan globe valves the differential pressure data was re-evaluated based on the EPRI methodology. The EPRI methodology does not utilize a valve factor, therefore, a comparison of design versus test valve factors was not possible. To evaluate the test data based on the EPRI methodology, the EPRI model was re-run utilizing the actual test conditions and the EPRI predicted thrust was compared to the thrust measured during the differential pressure test to determine if the valve's performance was bounded by the methodology. Table 9-26 summarizes this re-evaluation for the group V-1 valves.

Table 9-26
Velan Group V-1 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	EPRI Opening Thrust lb	Test Opening Thrust lb	Test/EPRI Opening
1HV-8116	7035	6483	92%
2HV-8116	7602	4861	64%

The results of the re-evaluation indicate that the valves are performing within the bounds of the EPRI methodology. The opening thrust for these valves was less than the required thrust predicted by the EPRI methodology. The closing thrust was not evaluated since these are flow ove: the seat valves and flow assists in the closing of the valves.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that the EPRI Performance Prediction Program methodology conservatively predicts th ust requirement for Velan 1 inch, 1500 lb, globe valves.

9.4.2 Velan Group V-2 Globe Valves

This group is composed of twelve, 1-1/2 inch, 1500 lb Velan globe valves. The valves are located in the Chemical and Volume Control System (CVCS). Differential pressure tests were performed on a total of four valves in this group.

Differential Pressure Test Overview

1HV-8103A/B - RCP Seal Water Inlet

These valves are normally open to provide RCP seal injection flow and have a safety function to close to isolate a line break inside containment. These valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at the test differential pressure.

When the test results were evaluated it was determined that the test differential pressure was much less than 50% of the design-basis differential pressure. A review of tests indicated that test differential pressures approaching design-basis would not be possible with the system configured as it was for the differential pressure tests. Due to the low differential pressures, the data for these valves was considered invalid and was not evaluated.

1HV-8920 & 2HV-8814 - SI Pump Miniflow

These valves are normally open to provide A and B train SI pump miniflow respectively to the reactor water storage tank (RWST) and have a safety function to close when transferring to cold-leg recirculation. The valves were tested in the opening and closing direction at differential pressures approaching design-basis conditions and each valve operated successfully.

When the test results were evaluated it was determined that the VOTES Force Sensor had been improperly calibrated prior to the 1HV-8814 differential pressure test. Therefore, the test data for this valve could not be evaluated. The test data for the valve 2HV-8920 was valid.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Velan Group V-2 valves were performed utilizing the industry standard equation with a 1.1 valve factor. The Velan differential pressure test data was originally evaluated based on this methodology. A valve factor was backed out of the test data and compared to the design valve factor to determine if the valves performance was bounded by the design methodology. Table 9-27 summarizes this evaluation for the group V-2 valves.

Table 9-27
Velan Group V-2 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design VF Open	Test VF Open	Design DP Close psid	Test DP Close psid	Design VF Close	Test VF Close
1HV-8103B	2715	294	1.1	N/A	2715	282	1.1	N/A
1HV-8103C	2715	384	1.1	N/A	2715	166	1.1	N/A
1HV-8920	1521	1404	1.1	N/A	1521	1404	1.1	1.14
2HV-8814	1521	1440	1.1	N/A	1521	1440	1.1	N/A

The results of the evaluation, for the valve with valid data, indicated that the test closing valve factor was not bounded by the design valve factor. Therefore, the decision was made to utilize the EPRI Performance Prediction Program methodology to re-evaluate the thrust requirements for the Velan globe valves.

The EPRI methodology does not utilize a valve factor, therefore, a comparison of design versus test valve factors was not possible. To evaluate the test data based on the EPRI methodology, the EPRI model was re-run utilizing the actual test conditions and the EPRI predicted thrust was compared to the thrust measured during the differential pressure test to determine if the valve's performance was bounded by the methodology. Table 9-28 summarizes this re-evaluation for the group V-2 valves.

Table 9-28
Velan Group V-2 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	EPRI Closing Thrust lb	Test Closing Thrust lb	Test/EPRI Closing
1HV-8920	6856	5037	73%

The results of the re-evaluation indicate that this valve is performing within the bounds of the EPRI methodology. The closing thrust for the valve was less than the required thrust predicted by the EPRI methodology. The opening thrust was not evaluated since this is a flow under the seat valve and flow assists in the opening of the valve.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that the EPRI Performance Prediction Program methodology conservatively predicts thrust requirement for Velan 1-1/2 inch, 1500 lb, globe valves.

9.4.3 Velan Group V-3 Globe Valves

This group is composed of twenty-two, 2 inch, 1500 lb Velan globe valves. These valves are located in the Safety Injection (SI) system and the Chemical and Volume Control System (CVCS). Differential pressure tests were performed on a total of sixteen valves in this group.

<u>Differential Pressure Test Overview</u>

1/2HV-8110 - CCP Miniflow Isolation

These valves are normally open to provide RCP seal water return flow and have a safety function to close on a containment isolation signal. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

1/2HV-8111A/B - CCP Miniflow Isolation

These valves are normally open to provide CCP miniflow to the seal water heat exchanger and have a safety function to close on a safety injection signal to isolate normal CCP miniflow. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions.

When the test results were evaluated it was determined that the VOTES test equipment was incorrectly zeroed when testing valve 1HV-8111A. Therefore, the test data for this valve could not be evaluated. The test data for the remaining three valves was valid.

1/2HV-8508A/B and 1/2HV-8509A/B - CCP Miniflow Isolation

These valves are normally closed and have safety function to open on a safety injection signal concurrent with high CCP discharge pressure to provide CCP miniflow to the RWST. The valves have a safety function to close when transferring to cold leg recirculation. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

1/2HV-8813 - SI Pump Common Miniflow

These valves are normally open to provide SI pump miniflow to the RWST and have a safety function to close when transferring to cold-leg recirculation. The valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at differential pressures approaching design-basis conditions and valid data was collected in each test.

Differential Pressure Test Data Evaluation

The initial GL 89-10 thrust calculations for the Velan Group V-3 valves were performed utilizing the industry standard equation with a 1.1 valve factor. The Velan differential pressure test data was originally evaluated based on this methodology. A valve factor was backed out of the test data and compared to the design valve factor to determine if the valves performance was bounded by the design methodology. Table 9-29 summarizes this evaluation for the group V-3 valves.

The results of the evaluation indicate that in certain cases the test closing valve factors were not bounded by the design valve factors. Therefore, the decision was made to utilize the EPRI Performance Prediction Program methodology to re-evaluate the thrust requirements for the Velan globe valves.

The EPRI methodology does not utilize a valve factor, therefore, a comparison of design versus test valve factors was not possible. To evaluate the test data based on the EPRI methodology, the EPRI model was re-run utilizing the actual test conditions and the EPRI predicted thrust was compared to the thrust measured during the differential pressure test to determine if the valve's performance was bounded by the methodology. Table 9-30 summarizes this re-evaluation for the group V-3 valves.

The results of the re-evaluation indicate that the valves are performing within the bounds of the EPRI methodology. The closing thrust for the tested valves was in all cases less than the required thrust predicted by the EPRI methodology. The opening thrust was not evaluated since these are flow under the seat valves and flow assists in the opening of the valves.

Conclusions

Based on the results of the in-situ differential pressure testing performed at VEGP it was concluded that the EPRI Performance Prediction Program methodology conservatively predicts thrust requirement for Velan 2 inch, 1500 lb, globe valves.

Table 9-29
Velan Group V-3 Valves
Original DP Test Data Evaluation Summary

Valve Tag No.	Design DP Open psid	Test DP Open psid	Design VF Open	Test VF Open	Design DP Close psid	Test DP Close psid	Design VF Close	Test VF Close
1HV-8110	2644	2632	1.1	0.91	2676	2640	1.1	1.15
1HV-8111A	2643	2293	1.1	N/A	2676	2293	1.1	N/A
1HV-8111B	2643	2614	1.1	0.73	2676	2464	1.1	0.91
1HV-8508A	2641	2547	1.1	0.83	2641	2547	1.1	0.97
1HV-8508B	2641	2560	1.1	N/A	2641	2573	1.1	1.11
1HV-8509A	2641	2547	1.1	N/A	2620	2509	1.1	1.03
1HV-8509B	2641	2534	1.1	N/A	2620	2534	1.1	1.0
1HV-8813	0	1504	1.1	N/A	1521	1504	1.1	1.39
2HV-8110	2644	2586	1.1	0.66	2676	2599	1.1	1.27
2HV-8111A	2643	2573	1.1	N/A	2676	2522	1.1	1.08
2HV-8111B	2643	2534	1.1	0.30	2676	2547	1.1	1.08
2HV-8508A	2641	2522	1.1	N/A	2641	2496	1.1	1.22
2HV-8508B	2641	2522	1.1	N/A	2641	2470	1.1	0.90
2HV-8509A	2641	2522	1.1	N/A	2620	2355	1.1	1.07
2HV-8509B	2641	2522	1.1	N/A	2620	2496	1.1	1.14
2HV-8813	0	1320	1.1	0.46	1521	1328	1.1	1.27

Table 9-30
Velan Group V-3 Valves
Revised DP Test Data Evaluation Summary

Valve Tag No.	EPRI Closing Thrust lb	Test Closing Thrust lb	Test/EPRI Closing
1HV-8110	11123	9602	86%
1HV-8111B	10580	7530	71%
1HV-8508A	10383	7490	72%
1HV-8508B	10471	9038	86%
1HV-8509A	10182	8890	87%
1HV-8509B	10326	8678	84%
1HV-8813	7089	6705	95%
2HV-8110	10941	9231	84%
2HV-8111A	10670	7421	70%
2HV-8111B	10310	8473	82%
2HV-8508A	10138	9081	90%
2HV-8508B	10051	6933	69%
2HV-8509A	9663	7490	78%
2HV-8509B	10138	8397	83%
2HV-8813	6470	5244	81%

9.5 Fisher Butterfly Valves

A total of four Fisher butterfly valves, representing one of the five Fisher valve groups, were differential pressure tested. The valves which were tested are identified in Table 9-31.

Table 9-31
Fisher Butterfly Valve
Differential Pressure Test Scope

Group No.	Tag No.	Procedure No.	Test Date
FB-2	1HV-1823	T-ENG-94-07	8/13/94
FB-2	1HV-1831	T-ENG-94-07	8/13/94
FB-2	2HV-2135	T-ENG-95-14	6/12/95
FB-2	2HV-2139	T-ENG-95-18	6/12/95

A limited scope of Fisher butterfly valves were differential pressure tested to evaluated the Fisher methodology utilized to calculate torque requirements for these valves. The test results were evaluated by SCS, although a formal evaluation procedure was not developed due to the limit scope of valves to be tested. The evaluations are documented in the attachment to letter SG-14279 dated June 28, 1995.

9.5.1 Fisher Group FB-2 Butterfly Valves

This group is composed of twenty-eight, 8 inch, 150 lb Fisher butterfly valves. The valves are located in the NSCW and AFW systems. Differential pressure tests were performed on a total of four valves in this group.

Differential Pressure Test Overview

1HV-1823 & 1HV-1831 - NSCW to Containment Cooler

These valves are normally open to provide NSCW flow to the Containment Coolers. The valves have a safety function to close on a manual actuation to provide containment isolation. These valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at the test differential pressure and valid data was collected in each test.

2HV-2135 & 2HV-2139 - NSCW to Reactor Cavity Coolers

These valves are normally open to provide NSCW flow to the Reactor Cavity Coolers. The valves have a safety function to close on a loss of offsite power or an SI signal to divert NSCW flow to more critical loads. These valves were tested in the opening and closing direction. Each of the valves opened and closed successfully at the test differential pressure and valid data was collected in each test.

Differential Pressure Test Data Evaluation

The GL 89-10 calculations for the Fisher Group FB-2 valves were performed utilizing the manufacturers proprietary methodology. The methodology utilizes a number of empirical factors to account for the various loads encountered by a butterfly valve under differential pressure conditions. To evaluate the test results the loads measured in the differential pressure tests were divided into static and dynamic components. The static component represents the torque due to the packing load and was assumed to be equal to the average running load which was measured in the most recent static test. The dynamic component was determined by subtracting the packing load from the maximum opening and closing torque measured in the differential pressure tests. The dynamic component was then compared to the dynamic torque predicted by the Fisher methodology at the test differential pressure.

Table 9-32 compares the Fisher packing loads to the average running loads measured in static tests performed prior to the differential pressure tests. It is apparent, based on a review of this data, that the measured running loads are substantially higher than the loads utilized by Fisher to account for packing loads. The higher than design packing loads were unexpected, therefore, to provide additional insight concerning this issue additional butterfly valves were statically tested during 1R5 and 2R4 to collect data relative to packing loads. The 1R5 and 2R4 testing confirmed the preliminary results and it was concluded that the Fisher packing loads were not conservative with respect to the Fisher butterfly valves in service at VEGP.

Table 9-32
Fisher Group FB-2 Valve
Static Packing Loads

Tag No.	Fisher Design Packing Load (ft-lb)	Opening Packing Load (ft-lb)	Closing Packing Load (ft-lb)
1HV-1823	9	90	94
1HV-1831	9	72	79
2HV-2135	9	20	41
2HV-2139	9	94	74

It is worth mentioning at this point that previous butterfly diagnostic testing which was performed at VEGP utilized either the MOVATS BARTS equipment or the Liberty VOTES equipment utilizing Torque Plugs. This equipment is installed on the HB gearbox and allows for certain diagnostics testing to be performed. Unfortunately, this equipment does not provide a means of measuring running loads. Therefore, although extensive diagnostic testing had been performed on rotating stem valves at VEGP, the data collected in these tests did not allow packing loads to be evaluated. At the onset of the butterfly valve differential pressure testing a new sensor manufactured by Liberty was utilized to measure torque. This new sensor is attached directly to the stem of the butterfly valve and provides an accurate measure of the torque being input into the valve stem. Data collected with this sensor was the first indication that the packing loads for these valves were higher than assumed in the Fisher calculations. As a result of this testing a standardized packing configuration has been developed for rotating stem valves and the packing loads utilized in the design calculations have been revised to more realistically reflect the actual loads. The revised packing configuration and resulting loads are discussed in detail in Section 3 of this document.

Table 9-33 and 9-34 compare the Fisher opening and closing dynamic loads to the loads measured in the differential pressure tests. The opening dynamic loads measured in the tests ranged from 62% to 105% percent of the Fisher predicted values. The closing dynamic loads measured in the tests ranged from 96% to 105% of the Fisher predicted values. The Fisher predicted values were exceeded in certain cases by approximately 5%, however, this apparent disagreement is insignificant relative to the overall accuracy of the test process. A minor error in the torque measurement or the differential pressure measurement would be sufficient to account for this slight discrepancy. Overall, the dynamic torque requirements for the tested valves were relatively consistent and in line with the predicted values.

Table 9-33
Fisher Group FB-2 Valve
Opening Dynamic Loads

Tag No.	Design DP (psid)	Test DP (psid)	Fisher Dynamic Torque (ft-lb)	Test Dynamic Torque (ft-lb)	Test/Fisher Dynamic Torque
1HV-1823	70	87	117	100	85%
1HV-1831	70	93	124	100	81%
2HV-2135	73	68	97	102	105%
2HV-2139	73	68	97	60.6	62%

Table 9-34
Fisher Group FB-2 Valve
Closing Dynamic Loads

Tag No.	Design DP (psid)	Test DP (psid)	Fisher Dynamic Torque (ft-lb)	Test Dynamic Torque (ft-lb)	Test/Fisher Dynamic Torque
1HV-1823	70	87	117	112	96%
1HV-1831	70	93	124	125	101%
2HV-2135	73	68	97	94.5	97%
2HV-2139	73	68	97	102.3	105%

Conclusions

Based on the results of the in-situ differential pressure testing, as well as additional static testing, it was concluded that the original static loads utilized in the Fisher methodology did not conservatively predict the actual packing loads for rotating stem valves at VEGP. As a result of this testing the packing configuration and resulting packing loads for these valves will be modified as required to ensure that the packing loads are predictable and bounded by the analytical methodology.

With respect to the dynamic loads, it was concluded that the Fisher methodology provided a reasonable approximation of the actual loads required to operate the valves under differential pressure conditions. Although the measured loads exceeded the Fisher predicted loads by approximately 5% in certain cases, this apparent disagreement was not considered to be significant given the overall accuracy of the test process.

9.6 Stem Friction Coefficient

In conjunction with the differential pressure test program data was taken to evaluate the stem friction coefficient. The stem friction coefficient is an empirical term which is used to calculate the stem factor. The stem factor is important because it is utilized to convert operator output torque to stem thrust for rising stem valves. This conversion is the basis for determining operator capability in terms of thrust, and as such, is an integral part of the design review calculations.

9.6.1 Test Methodology

Prior to discussing the results of the testing it is relevant to review the methodology utilized to determine the actual stem friction coefficient from test data. The stem friction coefficient is not a directly measurable term, therefore, it must be calculated from the test data. To perform this calculation it is necessary to have measurements of both stem thrust and operator output torque at some specified point in the valve stroke. This data is then utilized to calculate the stem factor as follows:

$$FS = \frac{Torque}{Thrust}$$

Where: FS = Stem Factor

The stem factor can also be calculated based on the geometry of the valve stem and the stem friction coefficient utilizing the following equation:

$$FS = \frac{d}{24}\left[\frac{(\cos\alpha)(\tan a) + \mu_{stem}}{\cos\alpha - (\mu_{stem})\tan a}\right]$$

Where: μ_{stem} = Stem Friction Coefficient
d = Stem outside diameter - Stem pitch/2 (for standard ACME threads)
α = Stem thread half angle
a = Stem thread lead angle

$$\tan a = \frac{L}{\pi d}$$

L = Lead

Rearranging this equation and utilizing the stem factor which was calculated based on the measured torque and thrust values allows the actual stem friction coefficient to be determined.

$$\mu_{stem} = \frac{(24FS\cos\alpha - d\cos\alpha\tan a)}{24FS\tan a + d}$$

When calculating the stem friction coefficient in this manner the accuracy of the torque and thrust measurements become very important. Relatively minor errors in these measurements can result in significant errors in the calculated stem friction coefficient when utilizing this methodology.

The thrust measurements were obtained utilizing a VOTES Force Sensor. This test equipment is described in more detail in Section 7.0 of this document. The accuracy of the thrust measurement utilizing the VOTES Force Sensor is specific to each valve and associated calibration, but is generally on the order of ± 10%.

Some of the initial torque measurements were taken utilizing the VOTES Torque Cartridge (VTC). The VTC measures spring pack displacement as well as the applied force allowing operator output torque to be calculated by the VOTES software. However, due to the intrusiveness of this device, this torque measurement technique was abandoned early in the test program.

The majority of the torque measurements were taken utilizing a methodology which employs both a B&W Spring Pack Tester and the MOVATS TMD. The spring packs were removed from the operators and torque versus displacement curves were developed utilizing the Spring Pack Tester. During the actual differential pressure testing spring pack displacement was monitored utilizing the TMD with the output of this device being feed to a spare channel in the VOTES data acquisition system. Combining the data developed on the Spring Pack Tester with the measured spring pack displacement during the test allowed operator output torque to be determined. Although the accuracy of this hybrid system has not been formally documented, the measurement technique is comparable to the VOTES VTC which has an accuracy of ± 10%. For the purposes of this evaluation it will be assumed that the Spring Pack Tester/TMD torque measurement system also has an accuracy of ± 10%.

Since the stem factor is calculated by dividing the thrust into the operator output torque, each of which have an accuracy of ± 10%, the inaccuracy of the resulting stem factor may be significantly greater than ± 10%. To illustrate this point assume a stem with standard ACME threads and the following properties:

Stem Diameter	1.25"
Pitch	.333
Lead	.666
Stem Friction Coefficient	0.15
Stem Factor	0.0163

The stem diameter, pitch and lead are simply a function of the stem geometry. The stem friction coefficient is an assumed value and the stem factor was calculated based on the geometric properties of the stem, which are constants, and the assumed friction coefficient. Knowing the stem factor it is now possible to relate operator output torque to stem thrust for this stem. If an operator output torque of 200 ft-lb is assumed the thrust corresponding to a stem factor of 0.0163 becomes 12,270 lb. Therefore, if field measurements were taken on a valve with this stem configuration and the stem friction coefficient were actually 0.15 then for a 200 ft-lb operator output torque a corresponding 12,270 lb of stem thrust would be expected.

The determination of a stem friction coefficient based on test data is essentially the reverse of this process. Torque and thrust measurements are taken and converted to a stem factor which is then utilized to calculate a stem friction coefficient. To evaluate the accuracy of this process, bounding errors were assumed for both the torque and thrust measurement. A stem factor was calculated for each case based on the measurement errors and a stem friction coefficient was subsequently calculated based on this stem factor.

<u>Base Case</u>

$$FS = \frac{200}{12{,}270} = 0.0163\,; \qquad \mu_{stem} = 0.15$$

<u>Test Case 1</u>

Torque Error = +10%, Thrust Error = -10%

$$FS = \frac{220}{11{,}043} = 0.0199\,; \qquad \mu_{stem} = 0.219,\ \text{Error} = \frac{0.219 - 0.15}{0.15} = 46\%$$

<u>Test Case 2</u>

Torque Error = -10%, Thrust Error = +10%

$$FS = \frac{180}{13{,}497} = 0.0133\,; \qquad \mu_{stem} = 0.091,\ \text{Error} = \frac{0.091 - 0.15}{.15} = -39\%$$

As can be seen from this analysis a 10% error in the torque and thrust measurement can result in very large errors when calculating the stem friction coefficient based on test data. Obviously, the likelihood of having concurrent errors of 10% in opposite directions is remote, but this example illustrates the worst case for this particular valve stem. Even with measurement errors of only 5%, errors on the order of 20% could occur when calculating the stem friction coefficient. It is important to recognize that the potential for errors of this magnitude exist when evaluating stem friction coefficient values calculated from test data.

The problems associated with test equipment inaccuracy are further substantiated by reviewing the stem friction coefficient data in Table 4-3 which taken in conjunction with the stem lubrication evaluation program. The 2R2 test data was taken utilizing a MOVATS Torque/Thrust Cell which is capable of measuring both thrust and torque with an accuracy of approximately 5%. A total of 20 sets of data were taken in 2R2 and only one set of data resulted in a calculated stem friction coefficient over 0.15 and none of the data sets resulted in a calculated stem friction coefficient of less than 0.08. By contrast, the data taken in 2R4 was collected utilizing a VOTES Force Sensor to measure thrust and a MOVATS TMD in conjunction with a spring pack curve to measure torque. The accuracy of each of these measurements is approximately 10%. A total of 16 data sets were taken in 2R4 resulting in three calculated stem friction coefficients over 0.15 and seven calculated stem friction coefficients of less than 0.08. This provides a clear indication of the data scatter associated with the less accurate test equipment and its impact on calculated stem friction coefficient values. The 2R2 lubrication data was the only stem friction coefficient data taken with the Torque/Thrust Cell and it is substantially more consistent than the remaining data which was taken with the VOTES Force Sensor and the TMD torque measurement.

9.6.2 Static and Differential Pressure Test Data

To evaluate the actual stem friction coefficient at VEGP both thrust and torque data were taken during the differential pressure tests. Similar data was also taken during the static testing which was performed prior to the differential pressure test in order to evaluate any changes in the stem friction coefficient which occurred when going from static to dynamic conditions.

The original intent was to evaluate the stem friction coefficient at flow cutoff on the closing stroke of the differential pressure test. However, this proved to be impractical due to the fact that in many cases the torque at flow cutoff could not be measured because there was little or no spring pack compression at this point. Therefore, the stem friction coefficient was evaluated at control switch trip during both the static and dynamic tests.

In performing a preliminary review of the data it was determined that the difference in the calculated stem friction coefficient values between the static and dynamic tests were not significant. While the mean of the dynamic stem friction coefficient data was higher than the mean of the static data, the differences were not substantial. Therefore, the decision was made to base the evaluations on the static test data for the following reasons:

1. Load Sensitive Behavior is theoretically characterized by variations in the stem friction coefficient from static to dynamic conditions. Since switch settings at VEGP are adjusted to account for this effect the static value is the relevant stem friction coefficient with respect to the engineering calculations. Any degradation in the stem friction coefficient which occurs when going from static to dynamic conditions is accounted for by the 20% adjustment which is included in switch settings for Load Sensitive Behavior.

2. Static test data was available for additional valves which had not been differential pressure tested. This allowed the overall population of valves being evaluated relative the stem friction coefficient to be increased to a more representative sample size.

Finally, the rising stem valves at VEGP employ two different stem thread configurations. The Fisher, Velan and Westinghouse valves utilize standard ACME threads while the Anchor-Darling valves utilize stub ACME threads. The preliminary evaluation of the test data indicated a significant difference in the stem friction coefficient for the two thread types, therefore, the standard ACME and stub ACME data were each evaluated independently.

Table 9-35
Standard ACME Thread
Stem Friction Coefficient

Valve Tag No.	Stem Friction Coefficient		Valve Tag No.	Stem Friction Coefficient
Unit 1 DP Test Valves Static Stem Fric. Coef.			Unit 2 DP Test Valves Static Stem Fric. Coef.	
1FV-5155	0.14		2FV-5154	0.15
1FV-0610	0.16		2FV-5155	0.10
1FV-0611	0.09		2FV-0610	0.15
1HV-5137	0.10		2HV-5120	0.13
1HV-5139	0.14		2HV-8110	0.15
1HV-8103B	0.16		2HV-8111A	0.12
1HV-8103C	0.06		2HV-8116	0.11
1HV-8105	0.08		2HV-8146	0.07
1HV-8106	0.07		2HV-8147	0.07

Valve Tag No.	Stem Friction Coefficient		Valve Tag No.	Stem Friction Coefficient
1HV-8110	0.07		2HV-8508A	0.09
1HV-8111B	0.10		2HV-8508B	0.16
1HV-8116	0.15		2HV-8509A	0.13
1HV-8146	0.12		2HV-8509B	0.15
1HV-8147	0.05		2HV-8716A	0.07
1HV-8508A	0.10		2HV-8716B	0.13
1HV-8508B	0.11		2HV-8801B	0.09
1HV-8509A	0.14		2HV-8802B	0.11
1HV-8509B	0.16		2HV-8804A	0.10
1HV-8716A	0.10		2HV-8804B	0.11
1HV-8716B	0.15		2HV-8806	0.10
1HV-8801A	0.13		2HV-8807A	0.12
1HV-8802A	0.09		2HV-8807B	0.22*
1HV-8802B	0.04		2HV-8813	0.18
1HV-8804B	0.17		2HV-8835	0.07
1HV-8806	0.11		2HV-8923A	0.15
1HV-8807A	0.15		2HV-8923B	0.12
1HV-8807B	0.12		2HV-8924	0.20
1HV-8813	0.12			
1HV-8821B	0.17			
1HV-8920	0.15			
1HV-8923A	0.08			
1HV-8923B	0.06			
Lubrication Study Valves Static Friction Coefficients				
2FV-0611	0.15		2HV-8485B	0.08
2HV-3548	0.12		2HV-8701B	0.1
2HV-5125	0.09		2HV-8702A	0.16
2HV-5132	0.09		2HV-8811B	0.1
2HV-5137	0.11		2HV-8840	0.07
2HV-8110	0.16		2LV-0112C	0.1

* Valve 2HV-8807B was determined to have a defective stem nut which was causing the high friction coefficient. Since this friction coefficient was not representative of the overall valve population this value was not factored into the evaluation.

Table 9-35 outlines the calculated static stem friction coefficient data for valves equipped with standard ACME threads. Included in this table are the differential pressure test valves with valid test data, as well as those additional valves which were tested in conjunction with the stem lubrication test program discussed in section 4.3.1 of this document.

The data contained in Table 9-35 was originally analyzed utilizing statistical techniques. The results of this analysis indicated that there was a significant degree of scatter associated with the sample data and that the standard deviation associated with this data was quite large. A 95% confidence value, which would correspond to the sample mean plus two standard deviations, resulted in a stem friction coefficient of approximately 0.18. However, considering the inaccuracies inherent in the process of calculating a stem friction coefficient based on test data, it was determined that this would not be a reasonable approach to evaluating this data. Due to the potential inaccuracies associated with the process, and the corresponding data scatter, an evaluation of this type would result in an unrealistically high stem friction coefficient.

The mean of the data, however, is meaningful because the uncertainties associated with the determination of the stem friction coefficient are random in nature and do not significantly alter the mean. The mean of the standard ACME thread stem friction coefficient data is 0.116 which is substantially less than the 0.15 value which is being utilized in the design review calculations. As is readily apparent, there are a number of valves which have calculated stem friction coefficients which are greater than the 0.15 value. However, given the inaccuracies associated with the determination of the stem friction coefficient from test data, this data scatter would be expected. What must be recognized is that these values are an inevitable result of the inaccuracies inherent in the test process and are not an indication of high actual stem friction coefficients.

Therefore, based on the results of this review, it was concluded that the use of a 0.15 stem friction coefficient on valves with standard ACME threads was supported by the test data. The Fisher, Velan and Westinghouse design review calculations will continue to utilize a stem friction coefficient of 0.15. It should be noted that stem friction coefficients are routinely monitored in conjunction with the periodic static test program and that any apparent stem friction coefficients which are outside the design value can be readily identified and evaluated.

As previously stated, the stem friction coefficient data for the Anchor-Darling valves, which utilize stems with stub ACME threads, appeared to be higher than the stem friction coefficient data for the standard ACME threads. Since there was only limited Anchor-Darling valve data acquired in conjunction with the differential pressure test program, existing static test packages were also reviewed to acquire additional stem friction coefficient data for this group of valves. Table

9-36 outlines the calculated static stem friction coefficient values for the valves which were differential pressure tested as well as those valves which had been statically tested and which had both torque and thrust data available.

Table 9-36
Stub ACME Thread
Stem Friction Coefficient

Valve Tag No.	Stem Friction Coefficient		Valve Tag No.	Stem Friction Coefficient
1HV-5106	.15		2HV-5106	.21
1HV-19051	.12		2HV-19055	.16
1HV-19053	.13		2HV-19057	.14
1HV-19055	.17		2HV-3009	.19
1HV-19057	.13			
1HV-3009	.16			
1HV-3019	.19			

Evaluating this stem friction coefficient data results in a mean of 0.159 which is slightly higher than the 0.15 value which was being utilized in the design review calculations. The mean for this data is significantly higher than for the standard ACME thread data. There is not an obvious explanation for the apparent differences in the stem friction coefficient between the standard ACME and stub ACME thread valves. The valves are maintained and lubricated in the same manner and differences of this magnitude would not have been expected. However, based on the results of this analysis, the stem friction coefficient being utilized in the design calculations for the Anchor-Darling valves has been increased from 0.15 to 0.20.

10.0 POST CLOSE-OUT ACTIVITIES

As a result of the differential pressure test program and the subsequent reconciliation of the differential pressure test data with the design review calculations, many of the MOVs were re-evaluated utilizing revised methodology. The re-evaluation process resulted in the development of revised minimum required and maximum allowable thrust and/or torque values for many of the MOVs.

In some cases the MOVs as-left configuration was not within the revised design range. In these cases Operability Determinations were developed to support the continued operation of the MOV in its present configuration until the appropriate steps can be taken to bring the MOV into compliance with the revised calculations. The MOVs which are currently set-up outside their respective revised design ranges are identified in Section 5.0 of this document.

In many cases the revised calculations, which were typically more conservative than the original design review methodology, resulted in reduced available margins. Therefore, in a number of instances, although the MOVs are currently set-up within the revised design range, modifications have been identified to provide additional margin.

10.1 Hardware Modifications

Valve and/or operator modifications will be implemented on a total of 58 MOVs to provide additional margin. The MOVs to be modified and their respective modifications are identified in Section 6 of this document. These modifications will be implemented during the 1996 refueling outages and are identified as having completion dates of 1R6 or 2R5 in Table 10-1.

10.2 Butterfly Valve Torque Switch Bypass and Repacking

The opening and closing torque switch for all butterfly valves will be bypassed in conjunction with each valves next scheduled periodic static test or preventive maintenance, which ever comes first. In addition, the packing loads will be evaluated in conjunction with each valves next scheduled periodic test and the valves will be repacked on an as-required basis. The completion dates outlined in Table 10-1 coincide with each butterfly valves current periodic test and/or preventive maintenance schedule.

10.3 Miscellaneous Activities

In addition to the hardware modifications outlined in Section 10.1, the following activities will be implemented in the 1996 refueling outages and have completion dates of 1R6 or 2R5 in Table 10-1:

1. The torque switch on butterfly valve 1HV-11612 will be bypassed due to the fact that the current open torque setting is less than the calculate required opening torque. An Operability Determination for this valve is included in Section 5 of this document.

2. The torque switches on butterfly valves 1HV-1807, 1HV-2135 and 1HV-2139 will be bypassed due to the fact that physical interferences prevent mounting the sensors necessary to set the switches utilizing diagnostic equipment.

3. The torque switches on valves 1HV-8716A and 2HV-8920 will be reset due to the fact that the current close torque switch settings are less than the calculated required closing torque. Operability Determinations for these valves are included in Section 5.0 of this document.

4. The open torque switch on valve 2HV-9017A and 2LV-0112C will be bypassed, thereby, completing the implementation of the DCPs which bypassed the open torque switches on all of the VEGP GL 89-10 rising stem valves.

Table 10-1
Post Close-Out Activities

Valve Tag No.	Post Close-Out Activity	Completion Date
1FV-0610	N/A	N/A
1FV-0611	N/A	N/A
1FV-5154	N/A	N/A
1FV-5155	N/A	N/A
1HV-11600	Bypass Torque Switch Repack as Required	09/96 03/99
1HV-11605	Bypass Torque Switch Repack as Required	08/96 03/99
1HV-11606	Bypass Torque Switch Repack as Required	07/96 02/99
1HV-11607	Bypass Torque Switch Repack as Required	06/96 01/99
1HV-11612	Bypass Torque Switch Repack as Required	1R6 03/99
1HV-11613	Bypass Torque Switch Repack as Required	09/96 04/99
1HV-1668A	Bypass Torque Switch Repack as Required	1R6 1R8
1HV-1668B	Bypass Torque Switch Repack as Required	1R6 1R8
1HV-1669A	Bypass Torque Switch Repack as Required	1R6 1R8
1HV-1669B	Bypass Torque Switch Repack as Required	1R6 1R8
1HV-1806	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-1807	Bypass Torque Switch Repack as Required	1R6 1R7
1HV-1808	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-1809	Bypass Torque Switch, Repack as Required	1R7
1HV-1822	Bypass Torque Switch, Repack as Required	1R7
1HV-1823	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-1830	Bypass Torque Switch Repack as Required	1R7 1R8

Valve Tag No.	Post Close-Out Activity	Completion Date
1HV-1831	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-19051	N/A	N/A
1HV-19053	N/A	N/A
1HV-19055	N/A	N/A
1HV-19057	N/A	N/A
1HV-1974	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-1975	Bypass Torque Switch, Repack as Required	1R7
1HV-1978	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-1979	Bypass Torque Switch, Repack as Required	1R7
1HV-2041	SB Conversion, Limit Switch Control, Verify Valve Guide Clearance > 1/16 Inch	1R6
1HV-2134	Bypass Torque Switch, Repack as Required	1R7
1HV-2135	Bypass Torque Switch Repack as Required	1R6 1R7
1HV-2138	Bypass Torque Switch, Repack as Required	1R7
1HV-2139	Bypass Torque Switch Repack as Required	1R6 1R7
1HV-2624A	Bypass Torque Switch, Repack as Required	1R7
1HV-2624B	Bypass Torque Switch, Repack as Required	1R7
1HV-2626A	Bypass Torque Switch Repack as Required	1R6 1R8
1HV-2627A	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-2628A	Bypass Torque Switch Repack as Required	1R7 1R8
1HV-2629A	Bypass Torque Switch Repack as Required	7 R8
1HV-3009	Convert to SB & Limit Switch Control, Verify Valve Guide Clearance > 1/16 Inch	1R6
1HV-3019	Convert to SB & Limit Switch Control, Verify Valve Guide Clearance > 1/16 Inch	1R6
1HV-3548	N/A	N/A
1HV-5106	N/A	N/A
1HV-5113	Replace Gears, Bypass Torque Switch, Repack as Required	1R6
1HV-5118	Bypass Torque Switch, Repack as Required	1R7

Valve Tag No.	Post Close-Out Activity	Completion Date
1HV-5119	Bypass Torque Switch Repack as Required	1R6 1R7
1HV-5120	N/A	N/A
1HV-5122	N/A	N/A
1HV-5125	N/A	N/A
1HV-5127	N/A	N/A
1HV-5132	N/A	N/A
1HV-5134	N/A	N/A
1HV-5137	N/A	N/A
1HV-5139	N/A	N/A
1HV-8000A	N/A	N/A
1HV-8000B	N/A	N/A
1HV-8100	N/A	N/A
1HV-8103A	Replace Valve Stem & Stem Nut	1R6
1HV-8103B	Replace Valve Stem & Stem Nut	1R6
1HV-8103C	Replace Valve Stem & Stem Nut	1R6
1HV-8103D	Replace Valve Stem & Stem Nut	1R6
1HV-8104	N/A	N/A
1HV-8105	Replace Gear Set	1R6
1HV-8106	Replace Gear Set	1R6
1HV-8110	Replace Valve Stem & Stem Nut	1R6
1HV-8111A	Replace Valve Stem & Stem Nut	1R6
1HV-8111B	Replace Valve Stem & Stem Nut	1R6
1HV-8112	N/A	N/A
1HV-8116	N/A	N/A
1HV-8146	N/A	N/A
1HV-8147	N/A	N/A
1HV-8438	N/A	N/A
1HV-8471A	N/A	N/A
1HV-8471B	N/A	N/A
1HV-8485A	N/A	N/A
1HV-8485B	N/A	N/A
1HV-8508A	Replace Gear Set and Spring Pack	1R6
1HV-8508B	Replace Gear Set and Spring Pack	1R6
1HV-8509A	Replace Spring Pack	1R6
1HV-8509B	Replace Spring Pack	1R6
1HV-8701A	Convert to SB, Limit Control, Replace Gear Set	1R6
1HV-8701B	Convert to SB, Limit Control	1R6

Valve Tag No.	Post Close-Out Activity	Completion Date
1HV-8702A	Convert to SB, Limit Control	1R6
1HV-8702B	Convert to SB, Limit Control, Replace Gear Set	1R6
1HV-8716A	Adjust Close Torque Switch	1R6
1HV-8716B	N/A	N/A
1HV-8801A	N/A	N/A
1HV-8801B	N/A	N/A
1HV-8802A	N/A	N/A
1HV-8802B	N/A	N/A
1HV-8804A	N/A	N/A
1HV-8804B	N/A	N/A
1HV-8806	N/A	N/A
1HV-8807A	N/A	N/A
1HV-8807B	N/A	N/A
1HV-8809A	Replace Gear Set	1R6
1HV-8809B	Replace Gear Set	1R6
1HV-8811A	N/A	N/A
1HV-8811B	N/A	N/A
1HV-8812A	N/A	N/A
1HV-8812B	N/A	N/A
1HV-8813	N/A	N/A
1HV-8814	N/A	N/A
1HV-8821A	N/A	N/A
1HV-8821B	N/A	N/A
1HV-8835	N/A	N/A
1HV-8840	N/A	N/A
1HV-8920	N/A	N/A
1HV-8923A	N/A	N/A
1HV-8923B	N/A	N/A
1HV-8924	N/A	N/A
1HV-8994A	Abandon in Place, Delete From GL 89-10 Scope	1R6
1HV-8994B	Abandon in Place, Delete From GL 89-10 Scope	1R6
1HV-9001A	N/A	N/A
1HV-9001B	N/A	N/A
1HV-9002A	Replace Gear Set	1R6
1HV-9002B	Replace Gear Set	1R6
1HV-9003A	Replace Gear Set	1R6
1HV-9003B	Replace Gear Set	1R6
1HV-9017A	Replace Gear Set	1R6

Valve Tag No.	Post Close-Out Activity	Completion Date
1HV-9017B	Replace Gear Set	1R6
1HV-9380A	N/A	N/A
1HV-9380B	N/A	N/A
1LV-0112B	N/A	N/A
1LV-0112C	N/A	N/A
1LV-0112D	N/A	N/A
1LV-0112E	N/A	N/A
2FV-0610	N/A	N/A
2FV-0611	N/A	N/A
2FV-5154	N/A	N/A
2FV-5155	N/A	N/A
2HV-11600	Bypass Torque Switch, Repack as Required	07/96
2HV-11605	Bypass Torque Switch, Repack as Required	06/96
2HV-11606	Bypass Torque Switch, Repack as Required	06/96
2HV-11607	Bypass Torque Switch Repack as Required	08/96 02/98
2HV-11612	Bypass Torque Switch Repack as Required	09/96 03/99
2HV-11613	Bypass Torque Switch Repack as Require	11/96 05/99
2HV-1668A	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-1668B	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-1669A	Bypass Torque Switch Repack as Required	2R5 2R7
2HV-1669B	Bypass Torque Switch Repack as Required	2R5 2R7
2HV-1806	Bypass Torque Switch, Repack as Required	2R5
2HV-1807	Bypass Torque Switch, Repack as Required	2R5
2HV-1808	Bypass Torque Switch, Repack as Required	2R5
2HV-1809	Bypass Torque Switch, Repack as Required	2R5
2HV-1822	Bypass Torque Switch, Repack as Required	2R5
2HV-1823	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-1830	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-1831	Bypass Torque Switch, Repack as Required	2R5
2HV-19051	N/A	N/A

Valve Tag No.	Post Close-Out Activity	Completion Date
2HV-19053	N/A	N/A
2HV-19055	N/A	N/A
2HV-19057	N/A	N/A
2HV-1974	Bypass Torque Switch, Repack as Required	2R5
2HV-1975	Bypass Torque Switch, Repack as Required	2R5
2HV-1978	Bypass Torque Switch, Repack as Required	2R5
2HV-1979	Bypass Torque Switch Repack as Required	2R5 2R7
2HV-2041	Convert to SB, Limit Switch Control, Verify Valve Guide Clearance > 1/16 Inch	2R5
2HV-2134	Bypass Torque Switch, Repack as Required	2R5
2HV-2135	Bypass Torque Switch Repack as Required	2R5 2R6
2HV-2138	Bypass Torque Switch, Repack as Required	2R5
2HV-2139	Bypass Torque Switch, Repack as Required	2R6
2HV-2624A	Bypass Torque Switch, Repack as Required	2R5
2HV-2624B	Bypass Torque Switch, Repack as Required	2R5
2HV-2626A	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-2627A	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-2628A	Bypass Torque Switch Repack as Required	2R5 2R6
2HV-2629A	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-3009	Convert to SB & Limit Switch Control, Verify Valve Guide Clearance > 1/16 Inch	2R5
2HV-3019	Convert to SB & Limit Switch Control, Verify Valve Guide Clearance > 1/16 Inch	2R5
2HV-3548	N/A	N/A
2HV-5106	N/A	N/A
2HV-5113	Replace Gears, Bypass Torque Switch, Repack as Required	2R5
2HV-5118	Bypass Torque Switch, Repack as Required	2R5
2HV-5119	Bypass Torque Switch Repack as Required	2R6 2R7
2HV-5120	N/A	N/A
2HV-5122	N/A	N/A
2HV-5125	N/A	N/A

Valve Tag No.	Post Close-Out Activity	Completion Date
2HV-5127	N/A	N/A
2HV-5132	N/A	N/A
2HV-5134	N/A	N/A
2HV-5137	N/A	N/A
2HV-5139	N/A	N/A
2HV-8000A	N/A	N/A
2HV-8000B	N/A	N/A
2HV-8100	N/A	N/A
2HV-8103A	Replace Valve Stem & Stem Nut	2R5
2HV-8103B	Replace Valve Stem & Stem Nut	2R5
2HV-8103C	Replace Valve Stem & Stem Nut	2R5
2HV-8103D	Replace Valve Stem & Stem Nut	2R5
2HV-8104	N/A	N/A
2HV-8105	Replace Gear Set	2R5
2HV-8106	Replace Gear Set	2R5
2HV-8110	Replace Valve Stem & Stem Nut	2R5
2HV-8111A	Replace Valve Stem & Stem Nut	2R5
2HV-8111B	Replace Spring Pack	2R5
2HV-8112	N/A	N/A
2HV-8116	N/A	N/A
2HV-8146	N/A	N/A
2HV-8147	N/A	N/A
2HV-8438	N/A	N/A
2HV-8471A	N/A	N/A
2HV-8471B	N/A	N/A
2HV-8485A	N/A	N/A
2HV-8485B	N/A	N/A
2HV-8508A	Replace Gear Set and Spring Pack	2R5
2HV-8508B	Replace Gear Set and Spring Pack	2R5
2HV-8509A	Replace Spring Pack	2R5
2HV-8509B	Replace Spring Pack	2R5
2HV-8701A	Convert to SB, Limit Control, Replace Gear Set	2R5
2HV-8701B	Convert to SB, Limit Control, Replace Gear Set	2R5
2HV-8702A	Convert to SB, Limit Control, Replace Gear Set	2R5
2HV-8702B	Convert to SB, Limit Control, Replace Gear Set	2R5
2HV-8716A	N/A	N/A
2HV-8716B	N/A	N/A
2HV-8801A	N/A	N/A

Valve Tag No.	Post Close-Out Activity	Completion Date
2HV-8801B	N/A	N/A
2HV-8802A	N/A	N/A
2HV-8802B	N/A	N/A
2HV-8804A	N/A	N/A
2HV-8804B	Re-evaluate and identify actions	2R5
2HV-8806	N/A	N/A
2HV-8807A	N/A	N/A
2HV-8807B	Replace Stem Nut	2R5
2HV-8809A	Replace Gear Set	2R5
2HV-8809B	Replace Gear Set	2R5
2HV-8811A	N/A	N/A
2HV-8811B	N/A	N/A
2HV-8812A	N/A	N/A
2HV-8812B	N/A	N/A
2HV-8813	N/A	N/A
2HV-8814	N/A	N/A
2HV-8821A	N/A	N/A
2HV-8821B	N/A	N/A
2HV-8835	N/A	N/A
2HV-8840	N/A	N/A
2HV-8920	Adjust Close Torque Switch	2R5
2HV-8923A	N/A	N/A
2HV-8923B	N/A	N/A
2HV-8924	N/A	N/A
2HV-9001A	N/A	N/A
2HV-9001B	N/A	N/A
2HV-9002A	Replace Gear Set	2R5
2HV-9002B	Replace Gear Set	2R5
2HV-9003A	Replace Gear Set	2R5
2HV-9003B	Replace Gear Set	2R5
2HV-9017A	Replace Gear Set, Bypass Open Torque Switch	2R5
2HV-9017B	Replace Gear Set	2R5
2HV-9380A	N/A	N/A
2HV-9380B	N/A	N/A
2LV-0112B	N/A	N/A
2LV-0112C	Bypass Open Torque Switch	2R5
2LV-0112D	N/A	N/A
2LV-0112E	N/A	N/A

11.0 PERMANENT PLANT PROGRAMS

There are a number of programmatic recommendations included in the generic letter which are essentially on-going in nature. These on-going activities are generally aimed at ensuring that the MOVs will be maintained in an operable condition for the life of the plant. Included in this category are the following recommendations:

1. Establishment of procedures to ensure that the appropriate switch settings are maintained for the life of the plant.

2. Implementation of a periodic testing program in conjunction with the preventive maintenance program to monitor MOV performance and to detect degradation in a timely manner.

3. Establishment of post-maintenance test guidelines to ensure that switch settings are adequately maintained following preventive and/or corrective maintenance.

4. Establishment of guidelines for the performance of root cause analysis to ensure that MOV failures are adequately addressed and that corrective actions are appropriate.

5. Implementation of a trending program to monitor MOV performance and to provide a basis for justifying extended periodic test intervals.

6. Implementation of a training program which addresses all aspects of MOV set-up and maintenance with particular emphasis on diagnostic testing.

Programs are in place at VEGP to implement each of the recommendations outlined above. The programs have been refined and enhanced as the overall MOV program progressed and additional data became available. An outline of the permanent plant programs is contained in the Maintenance Department maintained "Motor-Operated Valve Program Manual", GEN-34, Revision 2.