## San Onofre Nuclear Generation Site Unit 1

# FAILURE ANALYSIS OF SWING CHECK VALVES

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## Failure Analysis of Check Valves

This paper summarizes the results of an analysis which identifies the root causes of check valve failures. The failures resulted in a water hammer in the horizontal feedwater line B and subsequently damage to the pipe supports and by-pass line check valve FWS-378. Additionally, analysis is performed to demonstrate that the replacement valves, manufactured by Atwood & Morrill Co., Inc., will not experience the identified failure mechanism and they are acceptable in terms of functionality and reliability.

#### Failure Mechanisms

Based on the results of the analysis, it is concluded that the root causes of the failure are:

- 1) The swing check valves are oversized such that their discs are dangling in the flow stream during the reduced power, reduced temperature operation since November, 1984. In a properly sized application, the disc is pushed firmly against the stop in the fully open position, thereby restraining it against vibrations.
- 2) The above condition permitted flow induced-vibrations to occur since November, 1984, resulting in accelerated wear and fatigue damage to the valve internals.

Since the Unit 1 started the prolonged reduced power, reduced RCS (Reactor Coolant System) temperature operation in November, 1984, the steady state flow velocity has been reduced by as much as 12.5%. (In the later part of 1981 and the early part of 1982, Unit 1 also operated at reduced power for about four and one half months.) The reduction in fluid velocity resulted in a 24% reduction in the lifting force on the discs of check valves FWS-345, 346, 398, 438 and 439. Consequently, the discs were able to dangle freely in the flow stream instead of being firmly pushed against the stop in the fully open position. The freely dangling disc responds to fluctuations in the fluid forces of the flowstream, resulting in an amplified displacement response at resonance. The disc studs were forced to repeatedly impact the stud stops at the resonant frequency. Consequently, the pin nut threads and the lock pins were sheared or worn and resulted in final failures as observed by the post-event inspection.

The design parameters for the main feedwater and feedwater pump discharge check valves are documented in Tables 1 and 2.

#### Bases of Failure Mechanism

The bases of the failure analysis are stated in this section. To reach the conclusion of the analysis, several critical areas are carefully studied. They are (1) the positions of the check valves during steady state operation, (2) the flow induced-vibrations (3) the post-event metallurgical analysis (4) role of turbulence in flow induced vibration, and (5) lack of adequate minimum velocity of full open for check valve sizing in 1965.

#### Position of Check Valve Discs

The position of the check valve disc can be determined by a balance of fluid lifting force and the gravitational force acting on the disc and hinge arm weights. However, due to the difficulty in accurately estimating the lift coefficients and the exact pressure distribution for a specific type of valve, the valve manufacturer usually resorts to a flow test to determine the minimum flow required to fully open the valve. This minimum flow is usually published by the vendor in its catalog manual. In the present failure analysis, two means are used to determine whether or not the valves are fully open. One is to use the minimum velocity recommended by the catalog manual from MCC Pacific Valves, the vendor of the failed valves. The other is to extrapolate the known test results for a check valve of similar design by properly considering the force balance equations.

#### Minimum Velocity Recommended by Vendor

In a recent MCC Pacific Valves catalog manual (Reference 1), the minimum velocity in ft/sec recommended is  $135 \sqrt{v}$  (where  $\overline{v}$  is the specific volume in cubic foot per 1b.) based on tests specifically on the Pacific valves. Table 3 summarizes the results of actual flow velocities at full and reduced power conditions. Detailed calculations are documented in Appendix A. Based on the reduced power and reduced temperature flow condition, the port seat velocity through the main feedwater check valves, FWS-346, FWS-345 and FWS-398 is 16.23 ft/sec. The port seat velocity through valves FWS-438 and FWS-439, is 16.78 ft/sec. Both of these values are below the minimum velocity of 18.48 ft/sec. required to fully open the swing check valve disc. It implies that all of the above check valves are partially open during the steady state, reduced power, and reduced temperature operation since November, 1984.

The analysis also shows that under full power conditions, the actual flow velocities through the 10" feedwater check valves was 18.54 ft/sec. and through the 12" check valves was 19.18 ft/sec. Both of these meet the minimum velocity criteria, even though with very little margin.

## Extrapolation of Existing Data Using Force Balance Method

Crane Co., another valve vendor which has performed numerous experiments on valves, has stated (Reference 3) that a Crane swing check valve (50 lb. disc weight, 25 lb. hinge arm weight, and 35° flow impinging angle on disc, as shown in figure 1), has been tested to fully open at average flow velocity of  $60\sqrt{\sqrt{3}}$ . To extrapolate this minimum velocity requirement to the valves in question, the force balance formula (Appendix B) is resorted to.

The force balance method equates the lifting force resulting from the impinging flow to the valve disc to the gravitational force from the disc and hinge arm weights. The impinging angle is defined in Figure 1.

It should be noted that this method does not consider any possible low pressure zone due to minor boundary layer separation at the tip of the disc (Reference 3) and the non-uniform pressure distribution on the surface of the disc. However, it is reasonable to be used to extropalate existing test data from one valve to other geometrically similar valves, accounting for the differences in impinging angle, disc and hinge arm weight, and seat port area. This is done by determining an emperical constant K from the available test data.

The results show that the minimum required velocity to the main feedwater line check valve is about 21.0 ft/sec and for the feedwater pump discharge check valves is 21.7 ft/sec. The results again imply that both valves are partially open because the actual flow velocities are 16.23 ft/sec and 16.78 ft/sec., respectively.

#### Partial Openness of All Check Valves

Based on the above discussion, it is reasonable to conclude that all of these five valves are partially open under the reduced flow operating condition. Based on MCC Pacific Valve's recent recommendation on minimum velocity (Reference 1) and the Crane data, it is estimated that the discs are dangling at about 2° from the fully open position (Appendix C).

It is interesting to note that when the plant power and the RCS coolant temperature are at their 100% nominal value, the average port seat velocities are 18.55 and 19.18 ft/sec for the 10" and 12" check valves, respectively. Both port seat velocities are greater than the minimum velocity 18.48 ft/sec., as recommended by the valve vendor, MCC Pacific Valves. This suggests that during the steady state 100% power operation prior to November 1984, all these valves were either fully open or very close to completely open. As such, the valve were stable and free of large amplitude vibrations. Therefore, accelerated wear on the valve internals is not expected to occur.

#### Flow-Induced Vibration

It is well known that a bluff object in a turbulent flow stream is subject to flow induced vibration. The exact mode of vibration depends on the flow velocity, natural frequency, and geometry of the object (References 4, 5 and 6).

A dangling disc of a swing check valve in a flow stream with high turbulence will not be an exception. Because of the vibration, excessive wear of the disc stud, stud nut, and the disc stop is to be expected. In fact, reference 7, a commonly used reference book for valves, states the following:

> "Lower velocities are not sufficient to lift the disc through its full stroke and hold it in a stable position against the stops, and can actually result in increase in pressure drop as indicated by the curves. Under these conditions, the disc fluctuates with each minor flow pulsation, causing noisy operation and rapid wear of the connecting moving parts."

As previously discussed, all five check valves under the reduced flow operating condition are partially closed with a free swing angle of about  $2^{\circ}$ from fully open position. This angle translates into a 0.2" free traveling distance between the tips of the stud and the stop. Any resonant vibration will certainly result in repeated contact between the stud and the stop.

Based on the analysis performed (see Appendix D), the natural frequency for the 10" and 12" swing discs are 2.0 and 1.2 Hz, respectively. Any external force acting on the dangling disc at a frequency in the vicinity of its natural frequency will excite the disc into the resonant vibration mode. Because of lack of mechanical damping, the amplification factor can be relatively large and can easily result in a displacement response of greater than 0.2" free traveling distance between the stud and the stop.

Two types of cyclic force can be responsible for the excitation. One is caused by the flow redistribution with a changing disc position. That is, when the free swing angle increases, the flow will redistribute itself and increases the lifting force exerted on the disc to decrease the free swing angle. As a result, a cyclic forcing function is generated. The other type of the cyclic force can be related to turbulence. Based on the fact the turbulence tends to have a much higher power spectral density distribution in the low frequency ( $1 \sim 4 \, \text{Hz}$ ) (references 3, 8), it is reasonable to assume that self-excitation of the discs can be induced by the turbulence. In fact, a very low power spectrum density is needed to excite the disc into a resonant mode and create enough displacement of the disc to result in repeated impact. Appendix E documents the derivation.

#### Post-Event Metallurgical Analysis

One of the five discs (FWS-346) was sent to Mettek Co. for Metallurgical analysis. The report documents the following major findings (reference 9):

- 1) Some material found in the thread roots of the disc stud is removed from the disc pin nut.
- 2) Repeated opening of the valve and impacting of the nut and disc pin on the stop created the forces which ultimately led to the shearing of the threads in the nut and was responsible for the ultimate loss of the nut in the assembly.
- 3) The dimensions taken of the nut and of the disc pin assembly indicated that the top of the nut was very close to the top end of the disc pin. Based on this information, it would at least be possible for the nut to impact the stop on the underneath side of the bonnet upon repeated use and opening and closing of the check valve.
- 4) The physical damage and mushrooming on the end of the disc pin proves that the end of the disc pin assembly had in fact impacted the stop on multiple occasions.

#### Failure Mode

Even though only one failed disc has been metallurgical analyzed, it is believed that the failure root causes for all valves are identical. The only differences among them is the degree of severity. Based on the physical appearance of all failed valves, it is reasonable to postulate the failure mode in following chronological steps.

 The discs are dangling in the flow stream due to the reduced flow operation.

- 2) The stud and/or nut are impacting the stop at a frequency about 1 to 3 Hz. It is worth noting that at this frequency, the number of the fatigue cycles can reach one million just within 12 days.
- 3) The shear force generated by the cyclic impact shears the lock pins due to fatigue stress (for FWS-346, 345 and 398) and loosens up the nuts (for all valves).
- 4) The disc starts spinning due to imbalance of fluid force exerted on a tilted disc. Because of loose nuts, the disc spins even with the anti-rotation lugs or bars installed on the disc.
- 5) The rotational movement and the vibration of the disc further loosen up the nut. Also, large amplitude vibration of the disc generates cyclic force on the disc stud.
- 6) Stud nut is disconnected from the stud. The stud is broken due to fatigue stress.

Based on the physical appearance of FWS-438, FWS-439, and FWS-398, they were probably at step (4) of the postulated failed mode at the time of the water hammer event. Meanwhile, FWS-435 and FWS-436 were probably at step (6) of the postulated failure mode.

The reasons for a more severe wear of valve internals for FWS-346, FWS-345 and FWS-398 than for FWS-438 and FWS-439 are judged to be as follows:

- 1) Because of the existence of high turbulence just upstream of FWS-345, 346 and 398, due to a large pressure drop in the upstream control valves (as shown in Figures 2 and 3), the turbulence intensity in these valves is much stronger than that in FWS-438 and FWS-439. Higher turbulence level will tend to excite the resonant vibration mode to a larger amplitude. The relationship is demonstrated by the proportionality of the acceleration response to the square root of the turbulence power spectrum density, as derived in reference 10.
- 2) Based on the minimum velocity requirement in reference 11, the actual seat port velocities in FWS-438 and FWS-439 have a larger margin to the minimum velocity of full open than those in FWS-345, FWS-346 and FWS-398.

#### Role of Turbulence in Flow Induced Vibration

As discussed above, white noise turbulence can excite a dangling check valve into a large amplitude vibration at its resonant frequency. Also, the higher the turbulence intensity (measured in terms of power spectral density), the larger the vibration amplitude. However, it should be noted that the necessary condition for the vibration is that the disc has to be under a free-motion condition, i.e., dangling in the flow stream. If the disc is firmly pushed by the fluild lifting force against the stop (i.e., the turbulence fluctuating force is less than the valve seating force of full open), it will not vibrate no matter how strong is the turbulence intensity. This is because if the disc is pushed against the stop, the spring constant k (as calculated in Appendix D) will become infinite. As a result, the natural frequency for



the disc is infinite. With this infinite natural frequency, the disc will not go into resonance since the turbulence intensity exponentially decreases with frequency and the turbulence frequency domain is finite (references 11, 12). In other words, it will not vibrate (reference 13) because the natural frequency of the disc does not coincide with the excitation frequency. Note that in a very strong turbulence condition, the fluctuating force is estimated to be no more than 10% of the disc weight.

In summary, high turbulence intensity will exacerbate the wear of valve internals <u>only</u> when the valve is oversized and dangling in the flow <u>stream</u>. Therefore, the root cause of the check valve failures is the oversizing, not the mislocation of these valves in a high turbulent flow stream.

#### Lack of Adequate Velocity Requirement for Check Valve Sizing in 1965

As a current practice, oversizing of a check valve is avoided by comparing the port seat velocity with the minimum velocity of full open recommended by the valve vendor. If the port seat velocity is greater than the recommended minimum velocity of full open with a comfortable margin, the valve is considered to be not oversized. The size of the margin which is deemed appropriate varies from one design engineer to another. Another practice which is somewhat antiquated is that sizing the valve according to the size of pipe. Even though it minimizes the efforts needed to fit the swing check valves into the pipe, this practice often results in oversizing of check valves. Following statements quoted from reference 16 provide caution to the valve selectors.

#### "SIZING OF CHECK VALVES

Care should be taken to avoid oversizing of conventional swing and lift check valves. To obtain the minimum velocity required to lift the disk to the full open and stable position it may be necessary on some applications to fit valves smaller in size than the pipe in which they are installed."

The minimum velocities of full open for those check valves manufactured by MCC Pacific and another valve vendor, Crane Company, are listed in Table 4 at various years. As can be seen in Table 4, there is no minimum velocity recommended by these two vendors before 1969 whereas sizing of the failed check valves were done sometime in 1965 or 1966.

Because of lack of appropriate documents on valve selection in 1965, it is not possible to know exactly how the valves were sized. However, based on the fact that the 10" valves (FWS-345, FWS-346, FWS-398) were on the 10" feedwater line and 12" valves (FWS-428, FWS-429) were on the 12" lines, it is reasonable to speculate that those failed valves were selected based on the line sizes.

#### Evolution of Minimum Velocity of Full Open

From table 1, it is interesting to note that the minimum velocity of full open recommended by MCC Pacific Valves has gone through several changes. The 1978 48  $\sqrt{v}$  minimum velocity was recommended by MCC Pacific Valves based on the minimum velocity recommended by Crane Company for its swing check valves (reference 14). It is well known to check valve designers that the minimum velocity strongly depends on the disc weight and flow impinging angle. Since the impinging angle and disc weights between Pacific valves and Crane valves are different (10° and 45 lbm for 10" valve versus 35° and 75 lbms), the 48  $\sqrt{v}$ for Pacific valves is erroneously low even though it is appropriate for the Crane valves.

In 1982, Crane Co. has updated its recommendation from 48  $\sqrt{v}$  to 60  $\sqrt{v}$  based on some recent test. In 1985, MCC Pacific Valves published two minimum velocities of full open without any cross-reference or special qualification in two separate parts of its general catalog. Based on a recent letter from MCC Pacific Valves to Southern California Edison (reference 14), the inconsistency is explained. The 55  $\sqrt{v}$  in reference 18 is meant to be a minimum velocity generally accepted by the industry for a reasonable service life whereas the 135  $\sqrt{v}$  in reference 19 is based on the test data on Pacific's swing check valves.

It is judged that the  $135 \sqrt{v}$  minimum velocity for the Pacific values is credible as it is consistent with the  $48 \sqrt{v}$  and  $60 \sqrt{v}$  for the Crane values, considering the differences in impinging angle and disc weight between these two types of values. An equivalent minimum velocity for the Pacific values based on extrapolation of Crane's data for the Pacific values ranges between 123  $\sqrt{v}$  and 153  $\sqrt{v}$  (as derived in Appendix B).

As stated in reference 14, MCC Pacific Valves changed its recommendation on minimum velocity of full open from  $48 \sqrt[7]{v}$  to  $135 \sqrt[7]{v}$  in 1985 by more than a factor of 2.5. No special notification has been received by SCE regarding this change. A special notification would have alerted SCE of potential oversizing problems of all the Pacific valves and prompted SCE into an evaluation program before the occurrence of the water hammer event.

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### Evaluation of Replacement Valve

Based on the understanding of the failure mode of the five valves, the replacement valve should meet the following criteria:

- 1) The seat port velocity should be greater than the minimum required velocity for full open valve over a wide range of power levels.
- 2) The disc and the hinge arm are integral parts of a single component such that they will not be disconnected even under the flow induced vibration condition, occurring during limited low power operation.

Among all the available valves on the market, five swing check valves manufactured by Atwood and Morrill Co., Inc., are selected. The design data of these valves are tabulated in Table 5. Based on the force balance method extrapolating test data from Crane Co. (reference 2), the minimum velocity of full open is calculated to be 13.85 ft/sec. (Appendix F). The actual velocity at 92% power for the main feedwater line is 22.68 ft/sec and for the feedwater discharge line is 34.02 ft/sec. The power level at which the actual velocity for the main feedwater line check valves falls below the minimum required velocity is 61%. Below this power level, flow induced vibration will occur. However, because of the integral design of the hinge arm and the disc, the previously mentioned failure mode is judged not to occur.

#### Conclusion

The major conclusions of this study are:

- The actual flow velocities for all the failed valves were below the minimum velocity required to keep the valves fully open during the reduced power, reduced RCS temperature operation since November, 1984. This resulted in accelerated wear and fatigue damage due to disc vibrations causing failure of the valves. They, however, would have been fully open if the power and RCS temperature were at their 100% nominal values.
- 2) The design of the replacement valve manufactured by Atwood & Morrill Co., Inc., is judged to be acceptable with good margin above the minimum requirements.

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## 10" MCC PACIFIC CHECK VALVE DATA FWS-346, 345, and 398

VALVE SIZE:	10", WELD-END
Class	600
Seat Port I.D.	9.75 in
Seat Port Area	74.7 in <sup>2</sup> (0.52 ft <sup>2</sup> )
Disc O.D.	11.25 in
Disc Area	99.4 in $^{2}$ (0.69 ft <sup>2</sup> )
Disc Weight	33 lbs
Arm Weight	12 lbs
Total Impinging Angle	10°
Valve Cv	5440
Flow Rate/Valve	1,850,000 lb/hr (100% power) (4,317 GPM)
	1,618,750 lb/hr (92% power) (3,780 GPM)
Seat Port Velocity	18.54 FPS (100% power)
	16.23 (92% power)

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## 12" MCC PACIFIC CHECK VALVE DATA FWS-438, 439

VALVE SIZE:	12", WELD-END
Class	600
Seat Port I.D.	11.75 in
Seat Port Area	108.4 in <sup>2</sup> (0.75 ft <sup>2</sup> )
Disc O.D.	13.1 in
Disc Area	$134.8 \text{ in}^2$ (0.94 ft <sup>2</sup> )
Disc Weight	52.5 lbs
Arm Weight	13 1bs
Total Impinging Angle	10°
Valve Cv	7045
Flow Rate/Valve	2,775,000 lb/hr (100% power) (6,475 GPM)
	2,428,125 1b/hr (92% power) (5,670 GPM)
Seat Port Velocity	19.18 FPS (100% power)
	16.78 (92% power)

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## COMPARISON OF

## MINIMUM REQUIRED VELOCITY & ACTUAL VELOCITIES

## FOR MCC PACIFIC CHECK VALVES

VALVE SIZE	OPERATING UNDER % POWER	MIN. VEL.* FOR FULL OPEN (FPS)	ACTUAL SEAT PORT VELOCITY (FPS)	REMARKS
10"	100%	18.48	18.54	O.K. (Just above minimum)
	92%	18.48	16.23	Not O.K.
12"	100%	18.48	19.18	0.K.
	92%	18.48	16.78	Not O.K.



\* The minimum required flow velocity is based on the most recent Pacific Technical Data (Ref. 11)

$$V_{\min} = 135 \sqrt[4]{v}$$

 $\sqrt{\overline{v}}$  = Specific Volume at Operating Conditions

<u>Table 4</u>

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Year	MCC Pacific	Crane Company
1965-1969	none (reference 14)	none (reference 15)
1976		48 <del>v</del>
1978	48 / 🔽	48 VV
1982	48 <del>√</del> v	60 🛛 🔽 (reference 17)
1985	55 <mark>√v</mark> (reference 18)	60 H V
1985	135 √ v̄ (reference 19)	60 🗸 🔽

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10" 900# ATWOOD & MORRILL CO.

## CHECK VALVE DATA

FWS-346, 345, 398, 438 and 439

VALVE SIZE:	10" WELD END
Class	600
Seat Port I.D.	8.25"
Seat Port Area	53.4 in <sup>2</sup> (0.37 ft <sup>2</sup> )
Disc O.D.	9.25"
Disc Area	$67.2 \text{ in}^2 (0.47 \text{ ft}^2)$
Disc Weight	120 lb
Drum Weight	(included in 120 lb total)
Total Impinging Angle*	35°
Valve C <sub>v</sub>	

Note:

This information is preliminary based on the visual measurement performed on a non-qualified design drawing. However, the analysis will still be valid if the impinging angle for the actual design is greater than 35°

#### References

1 "Cast Steel - Bolted Bonnet Valves - Bulletin 400", pages 34 and 80 MCC Pacific Valves Form PCB-8-78 (1978).

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- 2 Personal Communication between C. Chiu (SCE) and K. Asher (Crane Co.), January 24 and 25, 1986.
- 3 H. Sehlichting, Boundary Layer Theory, McGraw Hill Book Company (1968).
- 4 R. D. Belevins, <u>Flow Induced Vibration</u>, Van Nostrand Keinhold Company, New York (1984).
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- 7 "Flow of Fluids through Valves, Fittings and Pipe", Crane Co., Technical Paper No. 410 (1976).
- 8 A. J. Yule, "Large Scale Structure in the Mixing Layer of a Round Jet", Journal of Applied Mechanics, Vol 89, pp. 413 - 432 (1978).
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- 10 W. T. Thomson, Vibration Theory and Application, Prentice Hall, Inc. (1963)
- 11 M. K. Au-Yang and K. B. Jordan, "Dynamic Pressure Inside a PWR A Study Based on Laboratory and Field Test Data," J. Nucl. Eng. Design, Vol. 58, pp 113-125 (1980)
- 12 A. Powell, "On the Fatigue Failure of Structures Due to Vibration's Exerted by Random Pressure Fields," J. Acoust. Soc. Am., Vol 30, (12), pp 1130-1135 (1958)
- 13 "Basic Vibration Theory," Chapter 1, <u>Shock and Vibration Handbook</u>, Edited by C. M. Harris and C. E. Crede, McGraw-Hill Book Co. (1976)
- 14 Letter from R. I. Armstrong to C. Chiu (SCE), dated February 21, 1986
- 15 "Flow of Fluids Through Valves, Fittings and Pipe," Crane Company, Technical Paper 410 (1969)
- 16 <u>Valve Users Manual</u>, Edited by J. Kemplay, Mechanical Engineering Publications LTD, London (1980)
- 17 "Flow of Fluids through Valves, Fittings and Pipe", Crane Co., Technical Paper No. 410 (1982).



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References (Continued)



18 "Cast Steel - Bolted Bonnet Valves", Undated Catalog, (obtained in 1985), page BB-36, MCC Pacific Valves.

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19 "Technical Data" page TD-27, Undated Catalog (obtained in 1985), MCC Pacific Valves.

#### APPENDIX A

## COMPARISON OF MINIMUM REQUIRED VELOCITY TO ACTUAL VELOCITIES UNDER FULL AND REDUCED POWER OPERATION

#### 10" Pacific Check Valve (Full Power)

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Recommended minimum velocity to open the valve fully is given by MCC-Pacific in their most recent Technical Data Catalog (Page TD-27, Reference 11).

Required  $V_{min} = 135 \sqrt{v}$   $\sqrt{v} = \text{specific volume (ft^3/lb)}$ for SONGS - 1, under original full operating conditions, T operating = 407°F  $\sqrt{v} = .01875 \text{ ft}^3/lb}$   $V_{min}$  required = 135 x  $\sqrt{.01875}$ = 18.48 FPS.

Actual flow rate under full power, through each 10" valve:

$$W = \frac{5,550,000 \text{ lb/hr}}{3}$$
  
= 1,850,000 lb/hr  
= 4317 gpm ( $\overline{v}$  = .0187)

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Valve Seat Port area, for 10" Swing Check Valve (Reference 19, page TD-26)

$$A_{seat} = 74.7 \text{ in}^2$$

Therefore, actual seat port velocity under full power:



Since the  $V_{actual}$  is just above the minimum required velocity  $(V_{min})$  to keep the disc fully open, the original conditions were just marginally sufficient.

FPS

Any reduction in flow rate would unseat the disc from the full open stop, and expose it to fluttering and flow induced vibration which can cause rapid wear of the disc support pin, nut, and hinge pin assembly.



## 10" MCC Pacific Valve Operation Under 92% Power

The actual mass flow rate under 92% Power operation is reduced to 0.875 times the full power flow rate.

New flow rate:

 $Q = 0.875 \times 5,550,000 = 4,856,250$  lb/hr

Flow rate through each 10" valve

$$Q = \frac{Q}{3} = 1,618,750$$
 lb/hr (= 3780 gpm)

Actual seat port velocity under 92% Power operation

Vactual	= .3208 x	<u>Q gpm</u> A in <sup>2</sup>	FPS
	= .3208 x	<u>3780</u> 74.7	
	= 16.23 FP	2	
	< 18.48 FP	S V <sub>min</sub> requir	ed

Since the actual flow velocity is less than the <u>minimum</u> required to keep the disc fully open, accelerated wear will be expected. This can dramatically reduce the life of the disc support components due to their being subjected to continuous vibrations, fluttering under actual operation.



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Recommended min flow velocity to open the valve fully (MCC Pacific Catalog, Page TD-27)

 $V_{min}$  required = 135  $\sqrt{v}$ = 135  $\sqrt{0.01875}$ = 18.48 FPS.

Actual flow rate under full power: (2, 12" valves)

 $W = \frac{5,550,000}{2} = 2,775,000 \text{ lbs/hr.}$ = 6480 gpm ( $\overline{v} = .0187$ )

Valve seat port area for 12", 600# (Ref: MCC Pacific, Page TD-26)  $A_{seat} = 108.4 \text{ in}^2$ 

Actual seat port velocity,

Vactual =  $0.3208 \times \frac{Q \text{ gpm}}{A \text{ in}^2}$ =  $0.3208 \times 6480/108.4$ = 19.18 FPS.

The  $V_{actual}$  of 19.18 FPS is greater than the minimum required  $V_{min} = 18.48$  FPS by the manufacturer, to keep the valve fully open. Thus, the original 100% power conditions fulfill the minimum requirements.



## 12" MCC Pacific Valve Operation Under 92% Power

Actual min flow rate under reduced power,

₩ = 4,856,250 lbs/hr. (See 10" Calculations)

Flow rate through each 12" valve,

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 $\frac{W}{2}$  = 2,428,125 lbs/hr. = 5670 gpm ( $\overline{v}$  = .01875)

Actual seat port velocity, under reduced power operations

V<sub>actual</sub> = 0.3208 X 5670/108.4 = 16.78 FPS < 18.48 FPS V<sub>min</sub> required

#### APPENDIX - B

#### CRANE TEST DATA & FORCE BALANCE METHOD TO ESTIMATE MINIMUM FLOW VELOCITY REQUIREMENT FOR MCC PACIFIC CHECK VALVE

Actual tests performed by Crane Valve Company (Ref: Crane Technical Paper 410, page A-27), along with pertinent test data for a 10" 600# Swing Check valve provided by Crane Engineering in January 1986 are summarized below:

Minimum pipe velocity for full disc lift =  $60 \sqrt{\overline{v}}$  FPS

Where  $\overline{v}$  = specific volume, ft<sup>3</sup>/lb.

Crane Test Valve details: 10" 600 Class

Disc weight = 50 lb

Arm weight = 25 lb

Angle wrt. horizontal, when fully open  $= 30^{\circ}$ 

Seat plane tilt w.r.t. vertical (approx.) = 5°

Total Angle change of the fluid impinging on the disc (fully open) = 30° + 5°

Disc O.D. ≈ 11.25"

Disc Force Balance Relationship

In order to be able to extrapolate the available Crane test data to analyze the effect of changes in disc weight, port size and the disc angle at the fully open position, the following relations are developed.

F, Lifting force from the fluid momentum change +  $F_h$ , Lifting force exerted on the hinge

= Total disc and arm weight, W.

F = (mass flow rate) x change in velocity (1)

 $F_h \approx \frac{1}{2} \bullet$  arm weight (obtained by balancing the moment around the axis passing the center of gravity of the disc)

 $A_{proj} = A \sin \theta$ , where  $A = \frac{\pi}{4} \cdot D^2$ 

## APPENDIX - B (Continued)

Change in velocity,  $\Delta V \approx V \sin \theta$  (For small  $\theta$ 's) (2)

Mass flow rate undergoing change in direction of  $\Delta V$ 

 $M = \rho \cdot V \cdot A_{\text{proj}}$   $M = \rho V A \sin \theta$ From (1), (2) & (3),

•...

 $F = M \times \Delta V$  $F \simeq \rho VA Sin \theta. V Sin \theta$ 

The above fluid dynamic force exerted on the disc can be equated to the total disc and arm weight minus one half of the arm weight through an emperically determined constant K =

(3)

(5)

 $W = K_{. \rho}AV^{2} Sin^{2}\theta + 0.5W_{a} \qquad (W_{a} = arm weight)$ (4)

The minimum flow velocity, V, required to keep crane valve fully open in the SCE SONGS-1 operating conditions can be calculated:

 $V = 60 \ 1^{\prime} \overline{v}$   $\overline{v} = 0.01875 \ ft^{3}/1b$  for water at = 400°F  $V = 60 \ r^{\prime} \overline{0.01875}$ = 8.2 FPS

Using the Crane 10" 600# valve data given earlier, and V = 8.2 FPS, we can now evaluate the emperial constant K:

 $K = \frac{W - 0.5W_{a}}{PAV^{2}Sin^{2}\theta}$ =  $\frac{(50 + 25 - 12.5) \times 0.9 (Bouyancy Factor)}{\frac{53.4}{32.2} \times \frac{\pi}{4} \times \frac{11.25^{2}}{144} \times 8.2^{2} \times Sin^{2}35^{\circ}}$ K = 2.21

\_B-2

## Extrapolation to MCC Pacific Valve

In order to estimate the minimum flow velocity required to fully open the MCC Pacific valve, we will use the above emperical constant K, and equation (4).

$$W - W_{a} = K_{P}AV^{2}Sin^{2}\theta$$

$$V = \frac{W - 0.5W_{a}}{K_{o}A \sin^{2}\theta}$$

for 10" 600# Pacific Check Valve,

 $W = 0.9 \times (39)$  accounting for bouyancy

= 35.1#

 $\theta$  = total change in fluid direction

= 10° at fully open position

$$V^{2} = \frac{35.1}{2.22 \times \frac{53.4}{32.2} \times \frac{\pi}{4} \times \frac{11.25^{2}}{144} \times \text{Sin}^{2}(10^{\circ})} = 458$$

V = 21 FPS.

This shows that the minimum flow velocity required to keep the 10" Pacific Check Valve is much higher than that for Crane Check Valve. The major reason for this is the very small angle (5° with respect to horizontal) of the disc at fully open position used in Pacific design.

#### Minimum Flow Velocity Formula (General) for Pacific Check Valve

The above results can be used to derive a general formula for minimum flow velocity required to keep the valve fully open.

 $V = C \sqrt{\overline{v}}$ 

Using results from the Pacific Valve case analysis,

V = 21 FPS

 $\overline{\mathbf{v}} = 0.01875 \text{ ft}^3/\text{lb}$  for a temp  $\simeq 400^\circ\text{F}$ 

$$C = \frac{21}{\sqrt{0.01875}}$$

= 153

Therefore, the minimum flow velocity relationship for Pacific Check Valve can be written as:

$$V = 153 l' \overline{v}$$
 (6)



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\* APPENDIX - B (Continued)

It is interesting to compare this with published catalog information by Pacific over the years. We find the following three chronological changes:

1978 Catalog, (Bulletin 400) page 34

Minimum V = 48  $\sqrt{\overline{v}}$ 

Also, repeated on page 80 of the same catalog

 $V = 48 \sqrt[7]{v}$ 

Undated Catalog, Acquired from Pacific in 1985

Page BB-36

Minimum V = 55  $\sqrt{v}$ 

Page TD-27

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Minimum V = 135  $\sqrt{v}$ 

The following observations pertaining to their minimum velocity formulas are important:

1. Their original formula of  $48 \sqrt{v}$ , as well as  $55\sqrt{v}$  appears to be in direct conflict with the Crane data, which states  $60\sqrt{v}$  for a disc that only opens to 30° from horizontal. To open it to 5° w.r.t. horizontal would obviously require a much higher velocity due to the assumption variation in lifting force as the disc approaches horizontal position.

No test data to support their recommendations of 48  $\overline{v}$  or 55  $\overline{v}$  are available, in spite of repeated request for this information from Pacific by SCE.

2. The most recently available technical data section shows a minimum

B-4

velocity requirement of 135  $\sqrt{v}$ .

This is quite a dramatic jump from their previous recommended valves. However, it appears to be in general agreement with our extrapolations of Crane test data, and the proper use of force balance relationships

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for the value disc (=  $153 \sqrt{\overline{v}}$ ).

## APPENDIX C

## Change in Disc Position Angle From Full Power to Reduced Power

Equation (4) from Apendix B relates disc weight to fluid force balance:

$$W - 0.5W_{a} = K_{P}AV^{2} Sin^{2}\theta$$
$$Sin^{2} \theta = \frac{W - 0.5W_{a}}{K_{P}AV^{2}}$$

K was evaluated from available test data to be 2.66 for the 10" Pacific Valve, under 100% Power.

$$Sin^{2} \theta = \frac{(0.9 \times 39) \times 32.2}{2.21 \times (53.4) \times 0.69 \times (18.55)^{2}}$$
  
= 0.0403  
Sin \theta = 0.20  
\theta = 11.5^{\theta}

For reducer power operation

≃ 2°

Sin<sup>2</sup> 
$$\theta$$
 = 0.0403 x  $\left[\frac{18.55}{16.23}\right]^2$  = 0.0526  
Sin  $\theta$  = 0.0229  
 $\theta$  = 13.3°  
 $\Delta \theta$  between full power and reducer power  
= 13.3 - 11.5 = 1.8°

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C-1

#### APPENDIX D

Natural Frequency Calculations For The Check Valve Disc/Arm Assembly Fluid dynamic force on the disc was shown to be equal to (Eq 4)

 $F = K.\rho AV^2 Sin^2 \theta$ 

Therefore, the disc assembly stiffness can be calculated

$$K_{stiff} = \frac{dF}{d\theta} = K.\rho.A.V.^{2} 2 \sin \theta.Cos \theta.$$

=  $K_{P}AV^{2}$  Sin 20

Therefore, the natural frequency of oscillations of the disc arm assembly

fn =  $\frac{1}{2\pi}$   $\frac{K_{stiff}}{M}$ 

$$= \frac{1}{2\pi} \qquad \frac{K.pAV^2. Sin2\theta}{M}$$

For the 10" 600 # PACIFIC VALVE

K = 2.21 for all conditions p = 53.4 lb/ft<sup>3</sup> A =  $\frac{\pi}{4} \times \frac{11.25^2}{144} = 0.69 \text{ ft}^2$ V = 18.55 FPS under 100% power M = 0.9 x 45 = 40.5 lb<sub>m</sub>, considering buoyancy factor θ = 10°

Therefore, for 10" Pacific Check Valve

fn = 
$$\frac{1}{2\pi}$$
   
= 2.21 x 53.4 x 0.69 x 18.55<sup>2</sup> x Sin 20°  
40.5  
= 2.2 hz. under 100% power  
fn<sup>1</sup> = 2.2 x  $\frac{16.23}{18.55}$   
= 2.0 hz under 92% power

D-1

#### APPENDIX D (Continued)

## For 12" Pacific Check Valve

Under 100% Power,

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fn	= 1	2.66 x 53.4 x 0.94 x 19.18 <sup>2</sup> x Sin 20
	2π	(0.9 x 65.5)
	= 1.3 hz	z. under 100% power
fn¹	= 1.3 x	<u>16.80</u> 19.18
	= 1.2 hz	under 92% power

Note: Because the valve discs are firmly seated at 100% power, the frequency determined here is done only for purpose of illustration. That is, if the discs were not fully open at 100% power, the natural frequency would have been the values calculated above. The disc will not vibrate when it is firmly seated.

#### APPENDIX E

#### VALVE DISC RESPONSE TO RANDOM VIBRATION POWER SPECTRUM INPUT

The response of a single degree of freedom system to vibration spectrum having a power spectral density (PSD) is given by (Ref: Vibration Theory and Application by W. T. Thomson, Prentice Hall, Inc., 1963):

$$XRMS = \left[\frac{\pi}{2} \times (PSD) \times fn \times Q\right]^{\frac{1}{2}}$$

Where  $X_{RMS}$  = Acceleration response in g PSD = Power spectral density in g<sup>2</sup>/hz fn = Natural frequency Q = Transmissibility at resonance, =  $\frac{1}{2 \text{ Cc}}$ 

Сc

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The RMS displacement response can be computed from the above:

$$Y_{RMS} = \frac{386 \times X_{RMS}}{(2\pi f_n)^2}$$
 (386 = 32.2 • 12.0)

critical dampling factor = 5%

$$= \frac{9.78 \text{ x}^{\text{X}} \text{RMS}}{(\text{fn})^2}$$
$$= \frac{9.78}{\text{fn}^2} \left[ \frac{\pi}{2} \cdot \text{PSD} \cdot \text{fn} \cdot \text{Q} \right]^{\frac{1}{2}}$$

This is the 1  $\sigma$  response, which occurs 66% of the time. The maximum response of 3  $\sigma$  occurs 2.8% of the time, and is usually more damaging from a fatigue damage standpoint.

$$Y_{max} = \frac{3 \times 9.78}{fn^2} \left[ \frac{\pi}{2} \cdot PSD \cdot fn \cdot Q \right]^{\frac{1}{2}}$$

E-1

#### APPENDIX E

One can rewrite the above to determine the PSD necessary to create a certain magnitude of  $Y_{max}$  response:

PSD = 
$$\begin{bmatrix} Y \\ 3 \\ \hline{x} \\ 9.78 \end{bmatrix}^2 \times \frac{1}{\frac{\pi}{2} \times fn \times Q}$$
  
=  $.00074 \quad \frac{Y \\ \frac{max^2}{Q} \\ fn^3}$   
=  $.000064 \quad g^2/hz \qquad Y_{MAX} = 0.2 \quad fn = 1.4$ 

Assuming a wide band white noise type power spectrum from 0-50 hz, the overall RMS'g level corresponding to the above PSD is only

$$G_{RMS} = \sqrt{.000064 \ \frac{g^2}{hz}} \times (50-0)$$

= .0566 g's

This 0.0566 g's is the equivalent force generated by turbulence needed to move the disc stud by 0.2". This value is considered reasonable with the high turbulent condition experienced by FWS-345, FWS-346 and FWS-398. Also, the actual critical damping factor is judged to be greater than 0.05. Therefore,  $G_{\rm RMS}$  calculated is conservative.

#### APPENDIX - F

## NEW 10" X 900# ATWOOD & MORRILL SWING CHECK VALVE CALCULATIONS

Using the disc force balance relationship derived for the MCC Pacific check valve flow calculations we have

$$Vmin = / \frac{W - 0.5W_{a}}{K_{p}A \sin^{2}\theta}$$
 ft/sec

where K = constant = 2.21 from test data

The Atwood & Morrill swing check valve has the following design data:

= 120 1b. = 8.25 in.	
= 55°	Approx. 20° from vertical in closed position
= 35°	Approx. 15° from horizontal
	= 120 1b. = 8.25 in. = 55° = 35°

Flow condition at SCE

Temperature = 405°F Specific Volume = 0.01875 ft<sup>3</sup>/lb. W = 120 lb. (W<sub>a</sub> = 0, for reason of conservatism)  $\rho = \frac{53.4}{32.2}$ A =  $\frac{1}{144} \times \frac{\pi}{4} (8.25 + 1)^2$ est. disc dia = 0.467 ft<sup>2</sup>. Vmin =  $\sqrt{\frac{120 \times 0.9}{2.21 \times 0.467 \times \sin^2(35^\circ) \times 53.4/32.2}}$ = 13.85 ft/sec

Therefore, the minimum required flow velocity at the check valve port is 13.85 ft/sec to keep the disc in fully open position.

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F-1 🧠

Flow Velocity for 100% Power Condition

$$W = \frac{5,550,000}{3} \ 1b/hr$$
  
= 1,850,000 1b/hr  
$$V = \frac{1,850,000}{53.4 \times \frac{\pi}{4}(8.25^2)} \times \frac{3600}{144}$$

V = 25.92 ft/sec > 13.85 ft/sec

Flow Velocity for 92 % Power Condition

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35241

 $V = .875 \times 25.92$  ft/sec

= 22.68 ft/sec > 13.85 ft/sec

Therefore, this new check value has the proper size to satisfy the 100% and 92% power flow conditions.

F-2



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 $\lambda$  = SEAT PLANE TILT WITH RESPECT TO VERTICAL

 $\beta$  = DISC ANGLE WITH RESPECT TO HORIZONTAL

 $\Theta = (\alpha + \beta) = \text{TOTAL IMPINING ANGLE}$ 

## FIGURE 1 CHECK VALVE CONFIGURATION





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- NOTES: 1. MAJOR TURBULENCE COMPONENTS (PRESSURE REDUCING CONTROL VALVE, AND AN EXPANDER) IMMEDIATELY UPSTREAM.
  - 2. ONLY IXD STRAIGHT PIPE LENGTH AFTER HIGH TURBULENCE COMPONENTS.

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## FIGURE 3: 10" 600# CHECK VALVE INSTALLATION (FWS - 345, 346, 398)





Response to I&E Question No. 2 NRC Request for Information of February 19, 1986 - San Onofre Unit 1

#### Question

- 2. Provide a determination and supporting documentation of the adequacy of the various check valve designs for application in the feedwater and other safety-related systems. This determination should include, as appropriate:
  - a. Evaluation of check valves that do not have integral internals to:
    - Assess existing inspection or test results.
    - Assess existing maintenance histories.
    - Perform and evaluate the results of disassembly and visual inspection of selected check values based on the results of selected check values based up on results of the above reviews.
    - Provide a description of any modifications required from these evaluations.
  - b. For check valves with integral internals:
    - Determine which valves are only partially open during nominal operating conditions.
    - Assess existing inspection and test results.
    - Review existing maintenance histories.
    - Perform and evaluate the results of disassembly and visual inspection of selected check valves based up on results of the above reviews.
  - c. Provide a description of any modifications required from these evaluations.

#### Response

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A complete response to this question cannot be provided at this time due to the fact that the major efforts in this area are ongoing. The attached action plan provides a description of the ongoing evaluations and a completion schedule. The scope of the action plan is limited to those check valves of a swing check valve design. The complete results of the action plan evaluations and our review of non-swing check valve designs will be provided in the March 28, 1986 final report to the NRC regarding the November 21, 1985 event.

## SAN ONOFRE UNIT 1

#### Introduction

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As a result of the failure of five Pacific Valve check valves at San Onofre Unit 1 and the evaluated failure mechanism, concerns over the generic applicability of this failure mechanism to other valve manufacturers of swing check valves at San Onofre Unit 1 were reviewed. Approximately 170 check valves have been identified in Unit 1 as having a swing check valve design. A representative sample (17 valves) of the various valve sizes, normally having flow through them, was selected for disassembly and inspection to evaluate the degree of degradation (if any) and generic implications. The manufacturers of these valves included Pacific Valve, Crane, Powell-William, Westinghouse, and Aloyco. The valves inspected were found to be operable. All valves would function properly if called upon to do so.

#### Action Plan

The following additional evaluations are being performed on the swing check valves described above:

- Maintenance will disassemble and inspect those Pacific swing check valves that have not been inspected to date (there are currently 21 Pacific swing check valves identified). This inspection is planned to be complete by March 21, 1986. 10 of the 21 valves have been inspected and all 10 have been determined to be operable. Some minor wear has been observed on the stud nuts of the 8" condensate check valves.
- 2. Maintenance records for the past 5 years are being reviewed to determine if any swing check valves have a history that would indicate a recurring problem that requires design resolution. To date no swing check valve has demonstrated a symptom of excessive wear. This review will be completed by March 7, 1986.
- 3. The Nuclear Plant Reliability Data System (NPRDS) is being reviewed to identify potential generic check valve problems or manufacturers with generic problems. Results of this effort will be utilized to identify further action as appropriate. Completion is forecast by March 7, 1986.
- 4. Flow calculations to determine if any swing check valve has been "dangling" will be completed by March 7, 1986, for check valves subjected to continuous flow. Inspection may be required based on the calculations.
- 5. A final, integrated report will be issued by March 28, 1986, that will summarize the results of Actions 1 through 4 above and any required corrective action.