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Nuclear Main Steam Isolation Valves

Edward Vogt
Valve Company
BTR FLOW CONTROL SYSTEMS

Nuclear Main Stream Isolation Valves

by
E. B. POOL,
Supervising Research Engineer
Valve Engineering and Research, Flow Control Division
Rockwell International
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Introduction

Main steam isolation valves (MSIVs) are used on nuclear steam supply systems. Their sole function is to close in the improbable event of a rupture in the steam piping. The closing is triggered by an electrical signal (or loss thereof) which operates solenoid valves to release trapped fluid opposing the stored energy of springs or compressed gas. This stored energy is to close the valve within specified time limits, most commonly not less than 3 nor more than 6 seconds.

Nuclear main steam piping normally ranges from size 16 to size 36. Valve ratings are between ANSI Class 600 and 900. Because of nuclear quality control requirements, the larger of these valves can cost well over $200,000 each. Therefore there is great incentive to use the smallest possible size of valve to meet the pressure loss specification for normal flow. Also there is great incentive to use the smallest possible stored energy system to actuate the valve to minimize not only cost but space requirements and seismic problems. Since a line break, let alone a clean double ended one, has never occurred, no valve has ever had the opportunity to prove itself. One significant test was conducted in 1969 of one supplier's valve at the State Line Station of Commonwealth Edison Co.[1] This required the use of a full size central station boiler for a considerable period. Acceptance of valves made by other suppliers, other sizes and other designs have all been the result of mathematical extrapolation and analysis.

As a leading supplier of these valves, Rockwell has felt the need, at its own expense, to use its extensive flow testing facilities with reduced scale models (1) to optimize flow shape for normal flow, and (2) to measure loads which the actuator must overcome under the critical flow conditions which would prevail at a line break. The results of this project of several years' duration are the subject of this article.

Flow Shape Development

The original concept of an inclined bonnet globe valve was that of an angle valve to which two 45° elbows were attached to place the inlet and outlet in line. The lift was limited to half the seat port diameter or less as illustrated in the lower drawing of *Figure 1, pg.3*. The bottom of the overseat port was scarcely below the pipe centerline (B nearly zero). Three equally spaced guide ribs were used with one splitting the overseat port. When this design was tested using the modeling technique described below, the maximum flow coefficient C (defined in *Figure 1*) was measured as 0.72 for flow from overseat (curve 1). Pressure drop is proportional to $1/C^2$.

When Rockwell Edward adopted the inclined stem concept for non-nuclear stop and stop-check valves in the mid 1950s, an established modeling technique using clay and one symmetrical half of the shape with low pressure air (*Figure 2*) was focused on developing an optimum shape. Flow from under the seat was of prime interest and disk lift was cheap compared to seat diameter in handwheel or electric motor operated valves. The result was a valve with (1) an effective* lift of 7/8 of the seat diameter, (2) a much lower seat so the flow passage was more nearly centered on the pipe centerline (A = 2B in *Figure 1*) and (3) generous "checking" or bulging over the seat (normal to the section shown). All of these factors combined to give a flow coefficient of 1.1 at this high lift. Thus the pressure loss was $(0.72/1.1)^2$ or 43% of the original.

The requirement for main steam isolation valves in BWRs developed in the mid 1960s. By providing a central balance hole through the disk (*Figure 5*) the pressure over and under the disk-piston assembly could be statically balanced, thus greatly reducing the forces acting on the valve stem and reducing necessary operator size. This application introduced two new requirements for the flow shape. First the normal flow direction adopted was from over rather than under seat. Second the piping layout was so confining that the valve had to have a minimum end to end dimension. Further work with the half model air tests (seat port diameter 4 inches or 100 mm) showed that the flow coefficient could be maintained in both directions at 1.1 in spite of the reduced length with small modifications in shape.

As sizes increased to more than size 24, however, the spring and superstructure design showed a large cost increment for the high lift as well as space and seismic problems. Another program was undertaken to reduce lift as much as possible without sacrificing flow capacity. The result of this program has been curve 2 shown in

*Effective lift excludes any required recessing of the seat below the flow contours.

(1) Report APED 5750, March 1969, "Design and Performance of General Electric Boiling Water Reactor Main Steam Isolation Valves'' by D. A. Rockwell & E. H. VanZylstra. Available from Nat. Tech. Infor. Service, Springfield, Virginia 22161

Nuclear Main Stream Isolation Valves

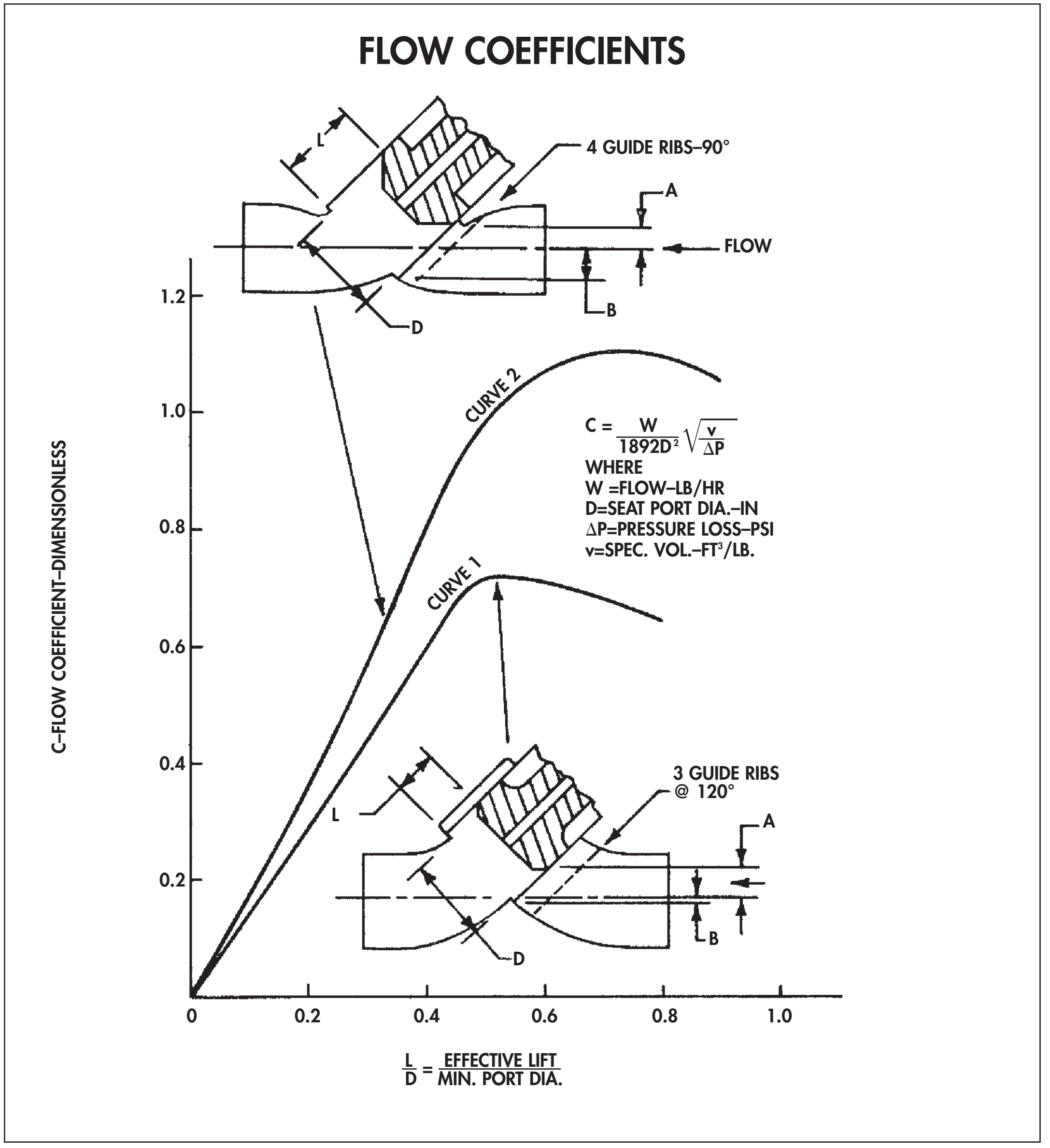

Figure 1: Flow Coefficients

Figure 1. Note that the flow coefficient has been maintained at 1.1 with a reduction of lift to 0.70 seat port diameters. Also note that the flow coefficient at any lift has been increased substantially so that the design may be used at lower lifts with minimum penalty in pressure drop.

This superior performance is due to several features. First the seat was further lowered with respect to the pipe centerline until the latter is midway in the overseat passage (A = B in *Figure 1*). Second a frustoconical disk nose extends the inlet "nozzle" to a point directly over the seat, at least near the center plane, preventing an upward flow diversion into a dead end. Third, the underseat outlet has been shaped like a shallow cone as nearly as possible to make a good diffuser.

Both curves of *Figure 1* have been confirmed by air testing a full 6" (150 mm) model consisting of two symmetrical halves molded in clay and bolted together. Many other shapes have been tried. Rockwell has been unable to attain a flow coefficient of 1.1 at any lift less than 0.7 seat port diameters with a body guided disk.

One of the first production valves of this shape (size 26) was also flow tested on water in Rockwell's water test loop at Kearney, Nebraska where up to 16,800 GPM were put through the valve, confirming the flow coefficient.

Disk Forces at Critical Flow

Rockwell has a facility at Murrysville, Pa. which has an underground storage capacity of over 300 ft^3 (8.5 M^3) of air at pressures up to 1500 psi (103 bars). This air

Figure 2: Half section models are molded of clay with care and precision.

can be discharged through a test valve into a muffler. The smallest available MSIV (size 16) has been tested at this facility several times with various modifications. (See *Figure 3* taken before installation of the muffler.) The results have been useful but with a field discharge pipe limited to size 8 and the rapid decline of the storage pressure, significant force measurements would have been difficult. Therefore a reduced scale model was constructed. A size 6 ANSI Class 2500 Flite-Flow valve was selected and modified for these force measurements. The seat port diameter of 4-3/8 inches (111 mm) makes it possible to hold inlet pressure fairly steady at 800 psi (55 bars) for at least 4 seconds with the valve fully open. All force and pressure measurements can be recorded on an 8 channel oscillograph, so that the force coefficients are based on simultaneous readings of force and pressure.

The test valve with instrumentation is shown in *Figure 4, pg. 5*. The axial load is measured by a load cell installed between the operator and the valve stuffing box. The actuator cylinder had oil under the piston and air pressure over. Trapping the oil fixed the average disk position. If the disk were stationary, the friction created by the side load would have introduced an indeterminate component into the stem load cell measurement. These friction forces were nullified by oscillating the stem with a pulsator on the oil system. The pulsator was a small displacement intensifier with its check valves removed. The amplitude of the disk stroking was +/- 1/32 inch with a frequency of about 5 Hertz. The average force reading was considered the friction-free value of axial load. A correction was made for the measured bonnet pressure acting on the stem area.

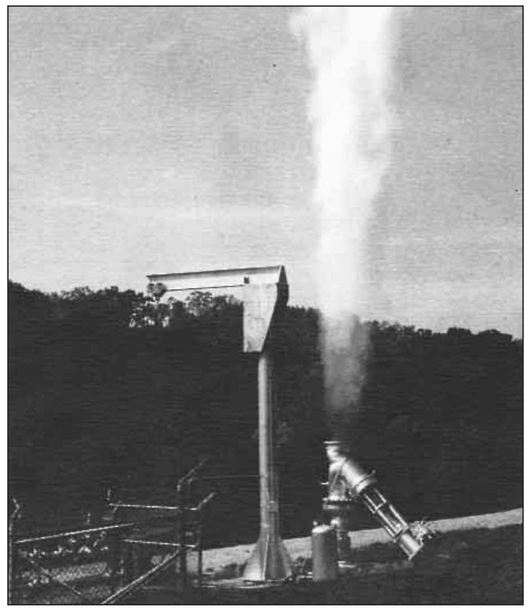

Figure 3: Seconds before shut-off, a Size 16 main steam isolation valve vents 1500 psi to atmosphere at Rockwell's high-pressure pipe storage facility near Pittsburgh. Test series demonstrated fail-safe closure in times as short as three seconds.

Side loads were measured by tiny load cells installed in a recess in the disk and in the piston as shown in *Figure 4*. They were carefully shimmed to protrude only mils beyond the recess. The strain gage wires were brought out through a connector in the bonnet. The pulsator was not used while the side force measurements were taken.the disk assembly was installed in various rotational positions as well as lifts to locate the maximum side force. Most of these fixed position tests were run with inlet pressures of 800 psi (55 bars) or more. In some of the side load tests the inlet pressure was reduced to avoid overloading the 1000 lb. (450 Kg) load cells.

Since all tests were run under critical flow conditions, the forces measured depended only on inlet pressure and were independent of downstream pressure. This independence has been checked over wide ranges of inlet pressure and valve lift. To render the force measurements independent of valve size, the disk outside circular area was selected for scaling. Thus the dimensionless coefficient for both axial and side forces is

$$G = \frac{1}{(\text{inlet press.})} \times \frac{\text{load}}{(\text{disk out. cir. area})}$$

This may be thought of as the fraction of the force required for a completely unbalanced disk just lifted off the seat.

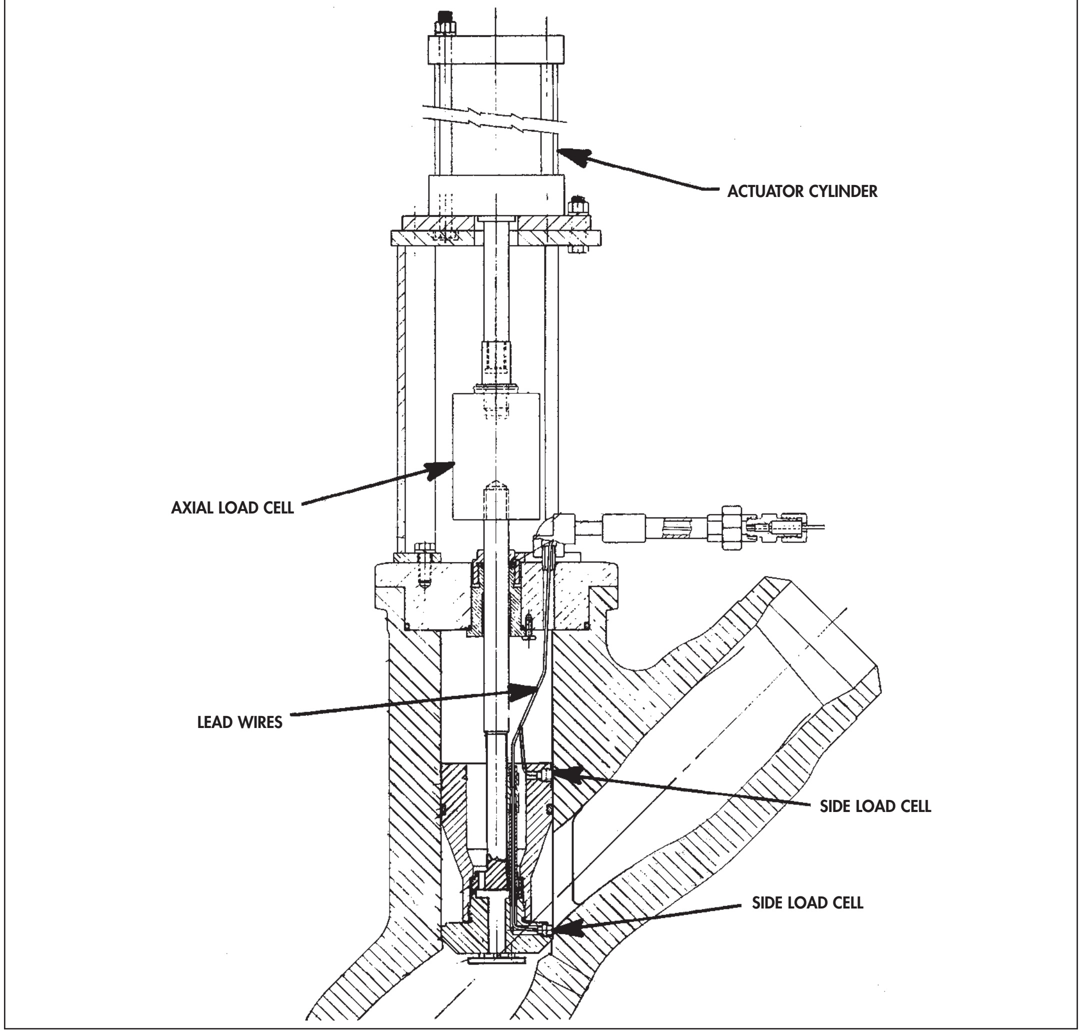

Figure 4: A size 6 ANSI Class 2500 Flite-Flow modified for testing.

Figures 5 to 8, pgs. 6-7, represent the axial and bearing force coefficients G_A and G_B for several geometries with flow from overseat and underseat. Originally only a line break causing flow from overseat was considered but if the break should occur between the steam generator and the valve, the valve must also close in some reactor systems to close off the steam header. Such a valve is referred to as a bidirectional MSIV.

Axial Forces

Note in *Figure 5* for flow from overseat that the disk without conical nose has a maximum pushup force (resisting closing) at low lift of about 4% of the completely unbalanced force (curve 1). This is increased to 9% when the nose cone is added (curve 2). It was theorized that this unanticipated force was due to the "new moon" portion of the disk upstream of the seat with respect to the flow path. (See arrows on sketch in *Figure 5*.) This is under locally higher pressure than the balance hole feels in the center of the disk. The effectiveness of the "new moon" area is increased by the conical disk. Exploration of the pressure distribution under the disk at these conditions confirmed this theory and suggested that radial holes from the balance hole directly to this area would increase the average pressure transmitted to the bonnet enough to balance the pushup forces. If the disk is not to be restrained from rotation, these radial holes must be spaced around the entire circumference. The model disk was modified and after several selections curve 3 of Figure 5 was obtained showing a small pulldown (assistance to closing) at all lifts. Thus the streamlining effect of the conical disk could be utilized without the need for increasing the size of the operator.

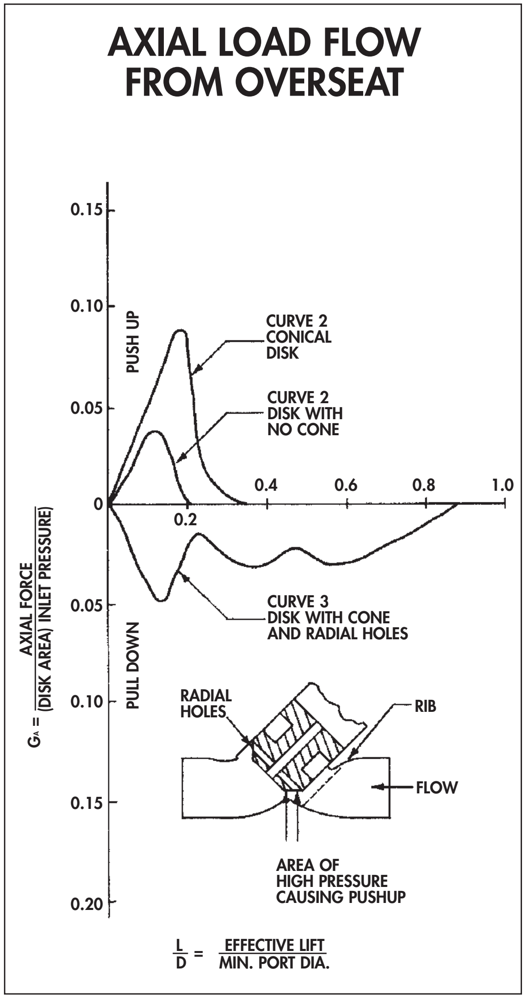

Figure 5: Axial load from overseat.

When the disk was tested with flow from under the seat (*Figure 6*), the disk shape was found to have negligible effect. With the open balance hole a large pulldown or closing force was discovered. This is more

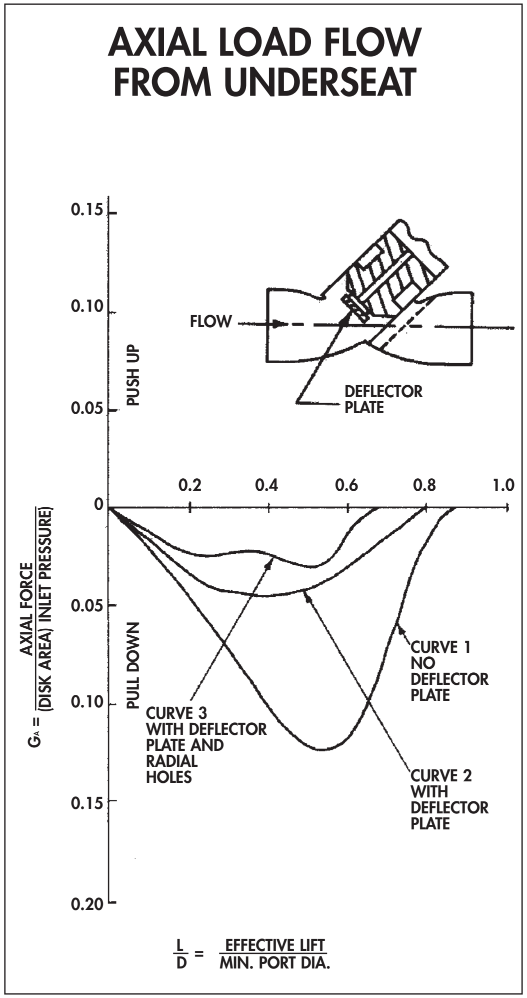

Figure 6: Axial load from underseat.

than 12% of the unbalanced force at a lift of half a port diameter (curve 1). Although this force assists in closing it can overtax the small stem or overpressure the dashpot which is resisting this force. It cannot compensate for high frictional forces because the latter are maximum at low lift where the pulldown force diminishes toward zero.

The pulldown force is attributed to total (static + velocity) pressure being transmitted up the balance hole while only a fraction of the velocity pressure is applied to the average disk surface. This results in a net downward force. A deflector plate was added to protect the balance hole from most of the velocity pressure. The result for the selected configuration is shown by curve 2 of *Figure 6* where the maximum pulldown force is reduced to a safe 4.5%. The deflector plate was found to have negligible effect for flow from over the seat. The radial holes were found to reduce the pulldown a bit more when combined with the deflector plate (curve 3 of *Figure 6*).

Bearing Forces

The bearing forces between the disk-piston assembly and the guide ribs were always found to be in the direction of flow. The bearing load on the piston was always less than 10% of the load on the disk, indicating that almost no significant couple was being applied by uneven axial pressure distribution on the disk.

The bearing force coefficients are shown in *Figures 7 and 8* on the same scale as *Figures 5 and 6*. Two disks were tested with bearing heights H equal to 8% and 12% of the disk diameter D_d. The body had 4 ribs at 90° and the disk was generously relieved above the bearing surface. Note that the bearing coefficient GB is proportional to the height of the bearing surface H.

These data conform to the theory that with flow from underseat the maximum bearing load corresponds to the upstream pressure applied to the projected disk bearing area (bearing height × disk diameter). The downstream effective pressure must then be zero. For flow from overseat the maximum bearing load is 40% of the upstream pressure applied to the same projected area.

For valves with 3 equally spaced ribs the projected area would be the height × 86% of the disk diameter, since the confined area is limited to 120° of arc. It is very important then to minimize disk bearing height below a generous relief. If no disk relief were used (cylindrical "pot" piston) these results would indicate a bearing load greater than the completely unbalanced axial load (like a gate valve). Although the bearing load is only 40% as great for flow from over the seat, it is significant. With three ribs the disk may be wedged between the widely spaced ribs on the downstream side, further increasing bearing loads and concentrating the bearing stress at the inner edge of the ribs. With four ribs at 90° this cannot happen since the bearing loads are then confined to the center plane of the valve.

Dynamic Tests

The above tests were run with the disk stationary except for the friction nullifying oscillation on axial load tests. Are there any additional forces encountered within the valve during a rapid closing?

To determine this, dynamic closings were tried. The actuator applied to the size 6

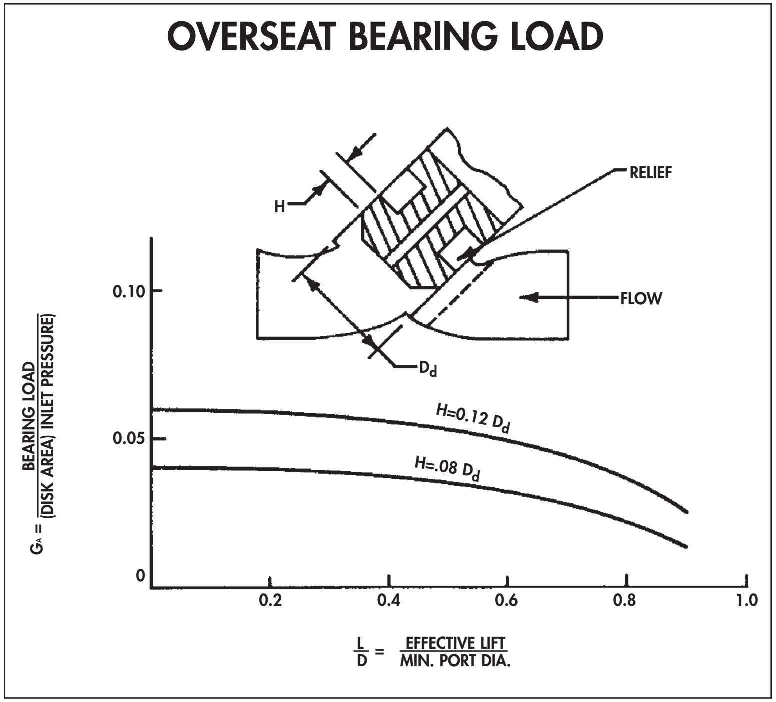

Figure 7: Overseat bearing load.

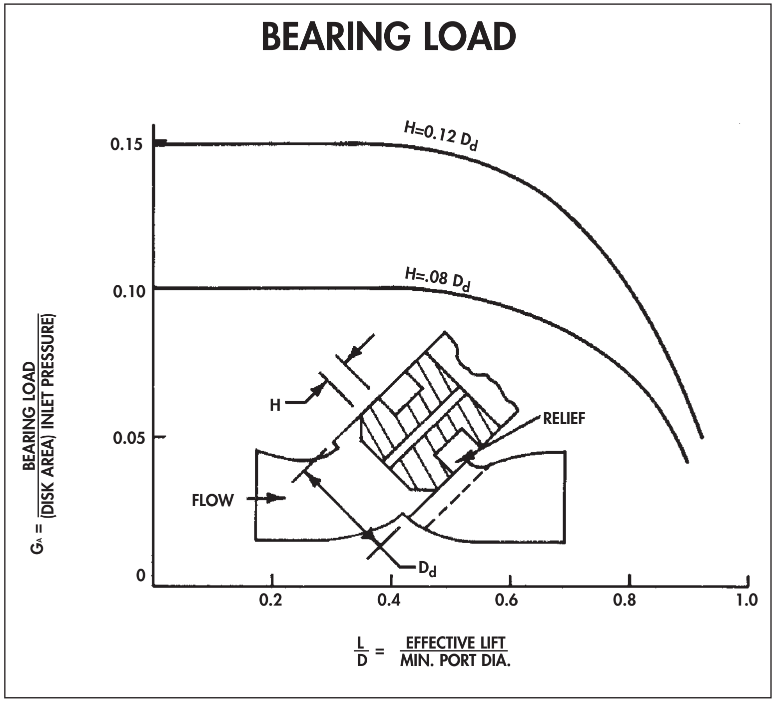

Figure 8: Underseat bearing load.

prototype valve was operated by oil pressure with the speed controlled by a constant flow rate valve exhausting the underpiston oil. A size 8 upstream storage field valve was opened in less than ½ second allowing the stored pressure to reach the open MSIV. Closing times for the valve ranged from 0.8 to 3 seconds with initial upstream pressures remaining substantially constant up to 900 psi (62 bars). During these tests the bearing load transducers were removed and the balance hole (with deflector plate) was open. The stem force measured was essentially that calculated by equation (2) below, indicating that no unexpected forces were present.

The open balance hole has been an oversimplification. MSIV designs provide for closing this hole at main valve seating either with the stem end in unidirectional valves or with a check element in bidirectional valves. The purpose is to avoid leakage past the piston guide and through the balance hole of the seated valve since balance is not required with pressure from over the seat. If pressure is applied from underseat, the check valve must open to avoid lifting the disk or overloading the stem due to fully unbalanced load.

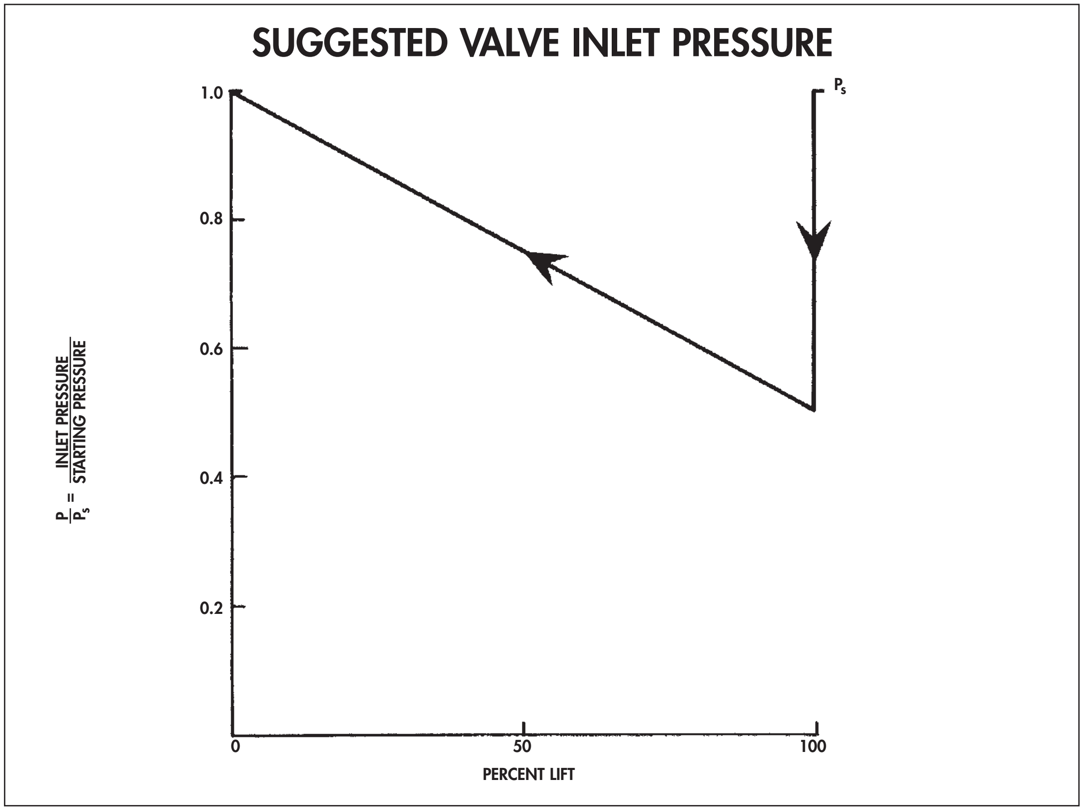

Figure 9: Suggested Valve Inlet Pressure.

The action of this valving in the balance hole is not confined to the seated position. Relative motion between stem and disk on dynamic closing causes the balance hole to be momentarily throttled or closed. This causes variations in the bonnet pressure and resulting spikes in the stem load. These load spikes on a representative oscillographic record were about 3% of the unbalanced load when a check element was added. The balance hole valving produced an increased average closing load on dynamic closing by holding some of the initial high pressure gas in the bonnet chamber for a longer duration.

Nuclear Main Stream Isolation Valves

Actuator Requirements

The actuator requirements may be found on a static basis from the following equation:

$$F = PA\,(G_A + \mu\, G_B + M) \qquad (2)$$

where:

- P = instantaneous valve inlet pressure–psi
- A = disk outside circular area-in2
- G_A = axial force coefficient (negative if pulldown) from *Figures 4 & 5*
- G_B = bearing force coefficient as outlined above
- μ = acceptable coefficient of friction at 60°F dry steam (some tests indicate this could be as great as 0.6).
- M = allowance for piston ring friction, stem push out, packing friction, operator friction and dashpot resistance. Rockwell data show 0.025 as a typical value.

The only variable in the above equation not so far specified is the instantaneous valve inlet pressure during line break. Most specifications have defined this as remaining constant regardless of the blowdown through the break. One architect engineer has ventured a computer prediction of a decline to less than 10% of original boiler pressure in 5 seconds, if the valve remained fully open. The curve in *Figure 9* is offered as a reasonable simplification of a very complex situation, P_s is the starting pressure: boiler drum pressure for flow from overseat. Or steam header pressure for flow from underseat. Due to inertia effects within the fluid, the inlet pressure to the valve drops to approximately half of P_s at line break. This loss of static pressure is required to reverse and accelerate flow. The valve acts as a critical flow nozzle limiting blowdown flow. As the valve closes the initial pressure will be linearly restored over the time of valve closure. With this curve the valve designer can determine the inlet pressure at the valve at any position independent of time, and hence solve the force equation at a number of lifts, knowing only the pressure prior to the break. He can then determine whether his actuator is adequate at all lifts to insure closing on the basis of a steady state analysis.

In an actual rapid closing of an MSIV, some pressure is momentarily trapped in the bonnet cavity giving an additional assist to the disk in reaching the seat. To calculate this additional stored energy assistance, a dynamic computer simulation can be used. Rockwell has made use of such simulations in design optimization, but a discussion is beyond the scope of this article.

Summary

If this article acquaints cognizant engineers in the nuclear field with the complexities of MSIV design, it will have served its purpose. The complexities center around designing for specified pressure loss and adequate actuator force while minimizing cost. If the design fails to consider all of these complexities, an MSIV may fail to function at the one event for which it was purchased, a catastrophic line break.

There is no substitute for a truly simulated line break test. However, it must reproduce the maximum possible coefficient of friction, and the maximum possible sustained upstream pressure to furnish complete proof of adequacy. Accomplishing this with an MSIV size valve is a formidable task.

Acknowledgement

The author is much indebted to R. L. Schweitzer, Senior Research Engineer in the Flow Control Division, who has conducted all the testing covered by this article.

Nuclear Main Stream Isolation Valves

Edward Vogt Valve Company
1900 South Saunders Street
Raleigh, North Carolina 27603 USA

Toll-Free Telephone Service
(U.S. and Canada)
Day: 1-800-225-6989

After Hours Customer Service:
1-800-543-3927

US Sales Offices
Tel: 919-832-0525
FAX: 919-831-3369
FAX: 919-831-3376
e-mail: eddie.vogt@btrinc.com

Vogt Operations
Tel: 812-218-7700
FAX: 812-218-7777
e-mail: eddie.vogt@btrinc.com

Mexico, Central America & Caribbean
Tel: 919-831-3233
FAX: 919-831-3369
e-mail: scott.mayo@btrinc.com

South America
Tel: 281-564-5671
FAX: 281-564-6052
e-mail: ruben.rios@btrinc.com

Northern Europe, India & South Africa
Tel: 44-1484-602519
FAX: 44-1484-609785
e-mail: kevin.henderson@btrinc.com

Southern Europe & French speaking North Africa
Tel: 33-4-77-40-18-79
FAX: 33-4-77-89-17-95
email: gilles.varona@btrinc.com

China
Tel: 86-10-6505-5548
FAX: 86-10-6505-5547
e-mail: david.xu@btrinc.com

Japan, Korea, Middle East & Australasia
Tel: 44-1444-443541
FAX: 44-1444-443540
e-mail: steve.corcoran@btrinc.com

Southeast Asia
Tel: 65-332-0557
FAX: 65-337-6179
e-mail: mickey.goh@btrinc.com

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www.edwardvogt.com

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