3.2 Design Basis Valve Stem Thrust

For rising-stem valves (gate and globe valves), the thrust needed to operate the valve is a summation of these types of loads:

a. Loads not dependent on the fluid conditions. These include loads such as disc and stem weight and packing drag. The disc and stem weight is often small compared to the packing drag, therefore, this load is often referred to as the packing load.

b. Loads caused by the valve internal pressure trying to expel the valve stem from the valve body in a piston-like effect. This is called the stem rejection load. The stem rejection load includes all loads dependent on system pressure.

Loads caused by the differential pressure acting on the valve disc surfaces. This is called the disc load or valve DP load. It includes all loads dependent on valve differential pressure.

For rising-stem valves:

$$F_{stem} = F_{pack} \pm F_{stem\ rej} + F_{disc}$$

Equation (3-1)

Where

F_{stem} = total stem load

F_{pack} = stem packing load

$F_{stem\ rej}$ = stem rejection load = $P_{up} A_{stem}$

$F_{disc} = f\,(P_{up} - P_{down})\,A_{DP}$

Note the ± sign for the stem rejection load. This appears because the stem rejection load is always out of the valve body, thus it resists closure and assists opening.

3.2.1 Gate Valve Stem Thrust

Figure 3-3 is a data plot showing stem thrust measurements taken during testing of the subject 6-inch gate valve. The thrust measurement is in the negative convention, a feature of the instrumentation used to obtain the data (negative thrust indicates compression in the stem, and positive thrust indicates tension in the stem).

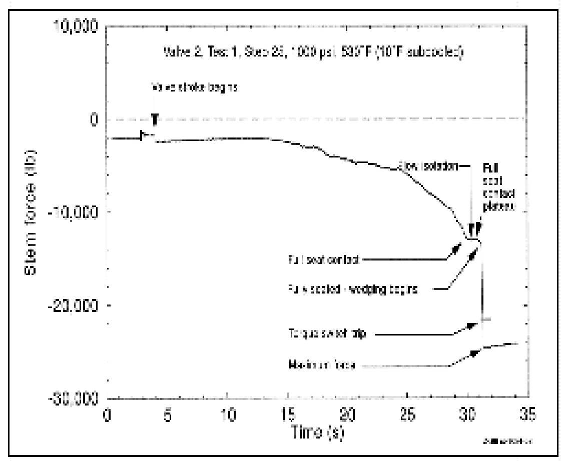


Figure 3-3 Stem Thrust Measurements

At the beginning of the closing stroke, the measured stem thrust is low, but as the disc approaches the seat and restricts the flow, the thrust increases. Flow isolation is indicated by the point in the trace where the thrust stabilizes momentarily. At this point, the disc is riding fully on the downstream valve body seat. The thrust increases abruptly a moment later, when the disc wedges tightly between the upstream and downstream valve body seats, and the thrust continues to increase until the torque switch trips. Motor inertia and other effects cause a small increase in the thrust after torque switch trip. In this test program, the valve actuator's torque switch was set higher than normal, to ensure that the valve would close all the way before the torque switch tripped, so that we could obtain all the data we needed from the test. With the exception of the high torque switch setting (in this data plot, torque switch trip occurs at a higher thrust than normal), this figure is typical of valve thrust data, and serves to illustrate how the magnitude of the stem thrust changes during the closing stroke. The crucial load occurs at or just after flow isolation, when the maximum differential pressure is applied to the maximum surface area of the disc.

The various loads that contribute to the total stem load are illustrated in Figure 3-4, a sketch showing the disc engaged against the downstream seats during a closing stroke, after flow isolation but before the disc is fully wedged in the seats. The angle of the disc in this sketch is exaggerated for clarity. The following formula mathematically represents the stem load for a valve operating in the closing direction:

$$F_{stem} = F_{pack} + F_{stem\ rej} - F_{elps} + F_{net\ stem}$$

Equation (3-2)

Where

F_{stem} = total stem load

F_{pack} = stem packing load

$F_{stem\ rej}$ = stem rejection load = $P_{up}\, A_{stem}$

P_{up} = upstream pressure

$A_{stem} = stem\ area = \pi \left(\frac{D_{stem}}{2}\right)^2$

D_{stem} = stem diameter

F_{elps} = elliptical pressure load
= $\Delta P\, A_{disc} \tan \alpha$

ΔP = differential pressure across disc
= $P_{up} - P_{down}$

P_{down} = downstream pressure

$A_{disc} = \pi \left(\frac{D_{mean}}{2}\right)^2 = disc\ area$

D_{mean} = mean diameter of the disc seat

α = seat angle from vertical

$\tan \alpha$ = 0.08749 for a seat angle of 5 degrees

$F_{net\ stem}$ = net stem load (discussed in detail later)

Using this formula, we can begin calculating an estimate of the stem load for our example valve, using valve dimensions, design basis conditions, and other parameters as input to the formula.

Minor loads

The most significant load represented in the previous equation is the net stem load, namely, that portion of the stem load caused by resistance to motion at the disc/seat interface. Our discussion proceeds with an evaluation of the other three, less significant loads.

In precise terms, the stem packing load applicable to the closing direction is the friction at the stem packing seal, minus the weight of the disc and stem. The packing load can be directly measured in an instrumented test with no flow or pressure, or it can be calculated from the results of a test with pressure but no flow (by subtracting the stem rejection load from the total load). Assuming no fluctuation in the actual packing friction during the no-pressure, no-flow test sequence, the measured packing load will be a little higher during opening than during closing, because of the weight of the disc and stem. Actual packing loads can vary considerably, depending on the type of packing and on the packing adjustment. Often, a default value of 1,000 lb_f per inch of stem diameter is used.

As stated earlier, the stem rejection load is the load caused by the pressure inside of the valve trying to expel the stem. It resists motion during valve closure but assists during opening. The stem rejection load resists motion during valve closure (but assists during opening). Stem cross-sectional area times upstream pressure equals the stem rejection load ($F_{stem\ rej}$ in Figure 3-4).

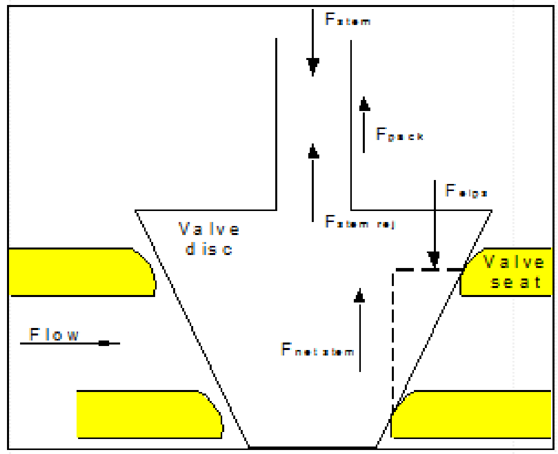


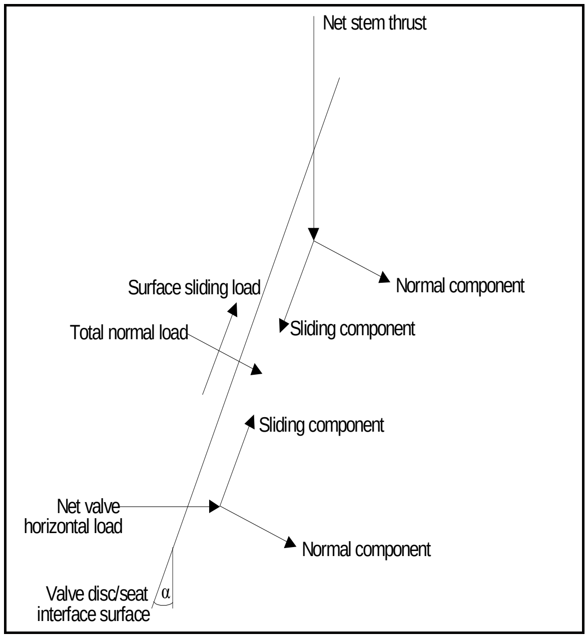
Figure 3-4 Various Stem Loads

Additional stem rejection forces can occur in wedge-type gate valves where the upstream and downstream pressures acting on the angled wedge surface cause additional vertical forces. Often, full differential pressure is used because the point of interest during the closing stroke is at or immediately after flow isolation, when the maximum upstream pressure is pushing against the maximum applicable disc area, and the disc is sliding on the downstream valve body seat, just before wedging. The elliptical pressure load on the top of the disc (F_{elps} in Figure 3-4) is the differential pressure times the area of the ellipse. The length of the ellipse is defined by the mean diameter of the seat, The elliptical pressure load assists during valve closure (but resists during opening).

(A note about differential pressures: occasionally results of tests using cold water or water with high subcooling show a downstream pressure less than 0 psi gauge, approaching 0 psi absolute. The occurrence of this negative gauge pressure immediately downstream of the disc at flow isolation is a phenomenon due to the momentum effects of the downstream fluid. In these uncommon instances, the differential pressure is greater than the upstream pressure.)

Net Stem Load

The discussion so far has addressed the three minor loads imposed on the valve stem during closure: the packing friction load, the stem rejection load, and the elliptical pressure load. The major stem load as the valve approaches and achieves flow isolation is the net stem load, that is, the stem load (a vertical load) created by the disc as it slides on the downstream valve body seat. If a valve is instrumented to measure stem thrust, one can calculate the net stem load as follows: total stem thrust minus packing load minus stem rejection load plus elliptical pressure load equals the net stem load.



1**Figure 3-5 Net Stem Load During Valve Closure**

Figure 3-5 shows the complex interaction of vertical forces, horizontal forces, and disc/seat angles involved in the net stem load during valve closure. Figure 3-5 shows only the net stem load, ignoring the minor loads discussed in the previous paragraphs. The term *net stem thrust* (indicated at the top of Figure 3-5) is equivalent to the net stem load. (In precise terms, *thrust* refers to the force imposed by the actuator on the top of the stem, pushing the stem downward; *load* refers to the resistance to motion at the bottom of the stem. Sometimes, however, the terms *thrust* and *load* are used interchangeably.)

If we were to look at the net stem load in very simple terms, ignoring the disc angle, we could treat the net stem load as the differential pressure times the disc area times the disc friction (coefficient of friction at the disc/seat interface). However, this simple view has a history of distorting the analysis of test results and producing inaccurate estimates of valve operating requirements. The following discussion is based on the more complex, more accurate view indicated in Figure 3-5. In this view, the friction at the interface between the disc and the valve body seats is represented as a true coefficient of friction, rather than a disc factor that includes other unknown or hidden variables along with the friction.

The following discussion describes mathematical formulas that represent the view shown in Figure 3-5. These formulas have two important applications.

- In an evaluation of valve operability. Where the net stem load is the unknown variable, standard or known variables serve as input to the formulas to produce an estimate of the net stem load.
- In an evaluation of test data. Where the net stem load can be determined from test measurements, the data can be analyzed in terms of the single unknown variable, namely, the disc friction.

The following discussion incorporates both of these applications (but not in a simple two-step sequence).

We now proceed with an evaluation of the net stem load. The mean seat diameter is used for this calculation. Disc area times differential pressure equals the horizontal force on the disc (see Figure 3-4). Note that because of the angle of the seat, this horizontal force is not exactly the same as the normal force (a force that is perpendicular to the seat), which holds the disc against the seat. The horizontal force is based on the area of the circular profile of the seat when viewed from the horizontal direction.

Figure 3-5 shows how we resolve the horizontal force (defined above) and the net stem thrust (described earlier) into normal components and sliding components. The normal component (a) of the horizontal force (b) combines with the relatively small normal component (c) of the net downward thrust of the stem (d) to create the total normal load (e). The total normal load times the disc friction produces a sliding load that represents the most significant load that the valve actuator must overcome. That load plus the sliding component (f) of the horizontal force equals the total sliding load (g), which is the total load that the sliding component (h) of the net stem thrust must overcome during valve closure.

The mathematical representation of the normal and sliding forces is:

$F_n = \Delta P\ A_{disc}\ cos\ \alpha + F_{net\ stem}\ sin\ \alpha$

Equation (3.3)

$F_s = \Delta P\ A_{disc}\ sin\ \alpha - F_{net\ stem}\ cos\ \alpha$

Equation (3.4)

Where

F_n = normal force

F_s = sliding force

ΔP = differential pressure across disc

A_{disc} = disc area(based on mean seat diameter)

α = seat angle from vertical

$cos\ \alpha$ = 0.99619 for a seat angle of 5º

$sin\ \alpha$ = 0.08716 for a seat angle of 5º

$F_{net\ stem}$ = net stem load (the resistance to motion imposed on stem by disc as it slides on the seat)

Since the sliding force F_s is equal to the normal force F_n times the coefficient of friction f, the coefficient of friction at the interface between the disc and the seat can be represented as F_s/F_n. Thus, substituting, rearranging, and simplifying yields the following representation of the net stem load:

$$F_{net\ stem} = net\ stem\ load = \frac{\Delta P\ A_{disc}\,(f \cos\alpha + \sin\alpha)}{\cos\alpha - f \sin\alpha}$$

Equation (3-5)

f = F_s / F_n

= disc friction (coefficient of friction)

This formula serves as the basis for making predictions of the net stem load and for evaluating the results of instrumented tests. Remember, if you know the differential pressure (and valve dimensions and geometry), and if you assume a conservative value for the disc friction, you can estimate the net stem load. If you measure the stem thrust and the differential pressure, you can calculate the net stem load using the approach described earlier in this chapter (net stem load equals total stem thrust minus packing load minus stem rejection load plus elliptical pressure load), and you can use the result to determine the actual disc friction.

Estimating the total stem load. The following equation shows the process for performing the calculation. As presented here, the equation includes the minor loads (packing load, stem rejection load, and elliptical pressure load) discussed earlier in this chapter, then evaluates the net stem load as described in the previous paragraphs. Thus, with inclusion of the additive bounding term, the formula for estimating the total stem load is:

$$F_{stem} = F_{pack} + F_{stem\ rej} - F_{elps} + \frac{\Delta P\, A_{disc}\,(f \cos\alpha + \sin\alpha) + 50\, A_{disc}}{\cos\alpha - f \sin\alpha}$$

Equation (3-6)

Where

f = 0.40 for fluid subcooling less than 70ºF

f = 0.50 for fluid subcooling 70ºF or greater

This is an estimate of the maximum stem thrust needed to close the valve at design basis conditions. The estimate is reasonably conservative without being excessively so.

Determining applicability of the INL correlation.

Use of the INL formula to make estimates like this one requires some reliable strategy for determining that the INL correlation, and the corresponding disc friction value of either 0.4 or 0.5, is applicable to the valve in question. We recommend that the valve be tested at a differential pressure of about 300 to 400 psid or greater. The valve should be equipped with instrumentation to monitor upstream pressure, differential pressure, and stem thrust, with these parameters recorded during the seating portion of the stroke, that is after flow isolation but before full wedging. If the results of the test thus recorded, and expressed in terms of normalized sliding load versus normalized normal load, fall within the bounds of the INL correlation as shown in, Figure 3-6, the valve can be considered typical. If the results fall outside those bounds, the valve cannot be considered typical, and it might be necessary to use some other method to estimate the disc friction for the valve.

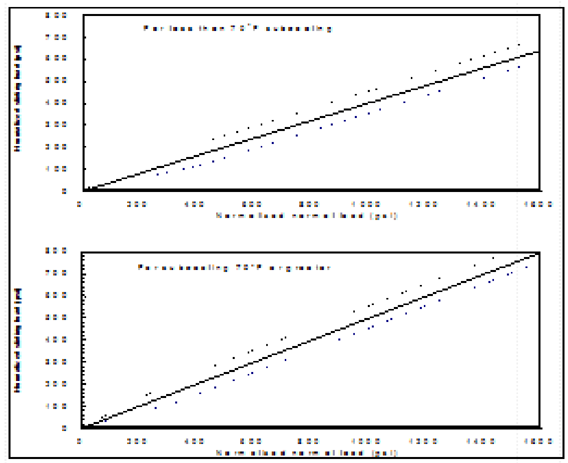


Figure 3-6 INL Secondary Equation Correlation

Note that use of the INL correlation in this manner does not constitute an extrapolation. With an extrapolation, the disc friction determined from a low-load test is used to estimate the stem thrust at a higher load. With the INL correlation, the low-load test is used to determine typicality. If typicality is successfully determined, then it is acceptable to use the INL formula (Equation 3-7) and the applicable disc friction term (either 0.40 or 0.50) to estimate the stem thrust at a higher load.

The INL correlation, as shown in Figure 3-6, includes a secondary equation that narrows the acceptable bounds for the data scatter at lower loads. At the lower loads, the upper and lower bounds are not defined as constant offsets relative to the normalized normal disc load, but instead are represented as a ± 30% variation in the friction factor. The threshold that separates the lower loads from the moderate to high loads for the 0.40 disc friction is a normalized normal force F_{nn} of 415 lb_f per $in.^2$ disc area, a value that corresponds with a differential pressure of about 400 psid in a gate valve with a seat angle of 5 degrees. For the 0.50 disc friction, the threshold is a normalized normal force F_{nn} of 330 lb_f per $in.^2$ disc area, corresponding with a differential pressure of about 320 psid. The mathematical representation of the correlation, showing both equations and including the upper and lower bounding terms to account for data scatter, is:

For DP > 415 psi (330 psi if the fluid subcooling is greater than 70°F)

$$F_{stem} = F_{pack} + F_{stem\ rej} - F_{elps} + \frac{\Delta P\, A_{disc}\,(f \cos\alpha + \sin\alpha) \pm 50\, A_{disc}}{\cos\alpha - f \sin\alpha}$$

Equation (3-7)

For DP < 415 psi (330 psi if the fluid subcooling is greater than 70°F)

$$F_{stem} = F_{pack} + F_{stem\ rej} - F_{elps} + \frac{\Delta P\, A_{disc}\,[f \cos\alpha\,(1.0 \pm 0.3) + \sin\alpha]}{\cos\alpha - f \sin\alpha\,(1.0 \pm 0.3)}$$

Equation (3-8)

The development of the portion of the INL correlation that addresses the lower loads is described in NUREG/CR-6100, *Gate Valve and Motor-Operator Research Findings*.

If low-load testing shows that the valve is not typical of the valves tested by INL and used to develop the correlation, the reason might be one of the following:

- New valves and newly refurbished valves tend to operate with unusually low disc friction values. Such valves generally operate at more typical disc friction values after they have been subjected to many strokes at cold water conditions or exposed to hot water or steam conditions. This process of achieving a stable disc friction via multiple strokes and/or exposure to hot water or steam is sometimes referred to in the literature as *preconditioning*.
- Some valves, particularly those manufactured by Borg-Warner, tend to operate with higher disc friction. This issue discussed in NRC Information Notice 89-61.

- The disc guide clearances in some valves are such that during closure against high differential pressure, the disc tips on the guides enough that an unusual pressure distribution around the disc occurs, and the disc seat does not engage smoothly against the valve body seat.

The performance of this last group of valves is called *nonlinear*, because the differential pressure times the area of the disc exposed to pressure does not produce a linear relationship when plotted against the stem thrust measurement. In extreme cases, damage to the guides and seats can occur, especially if the leading edge of the disc gouges the valve body seat as the valve approaches flow isolation.

Gate valves with nonlinear performance

Figure 3-7 shows a stem thrust trace from a test in which valve damage did not occur, but the performance of the valve was nonlinear, as indicated by the hook shape just before flow isolation. Compare this figure with Figure 3-3, a stem thrust trace displaying the classic, linear response. Valves with nonlinear performance cannot be evaluated using the INL correlation described earlier in this chapter.

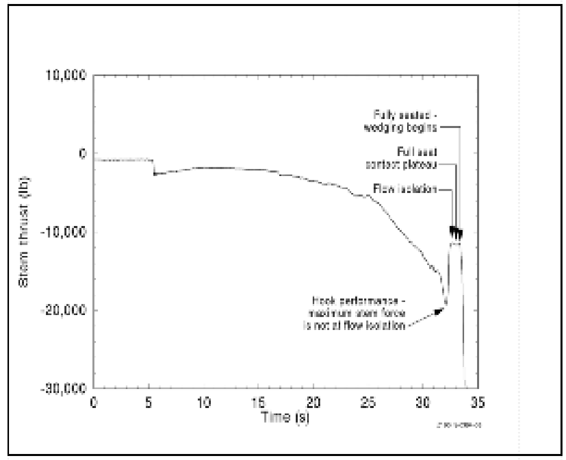


Figure 3-7 Gate Valve Stem Thrust Measurement

Figure 3-8 illustrates some of the flow and pressure dynamics involved during a valve closure in which the disc tips. These flow dynamics might contribute to the nonlinearity in the stem thrust measurements in a valve closure in the absence of valve damage.

An instrumented, best-effort test at a differential pressure of at least 200 psid or half the design basis differential pressure (whichever is smaller) is probably sufficient to determine whether the valve’s performance is linear or nonlinear. Of course, the 300 to 400+ psid test used to determine typicality for applicability of the INL correlation (as described above) would serve this purpose. For valves with non-linear performance, it might be possible, under some circumstances, to use the results of the best-effort differential pressure test to make a prediction of the valve's response at full design-basis differential pressure. A brief description of that method is presented here. Additional discussion of this issue is presented in NUREG/CR-6100 *Gate Valve and Motor-Operator Research Findings*.

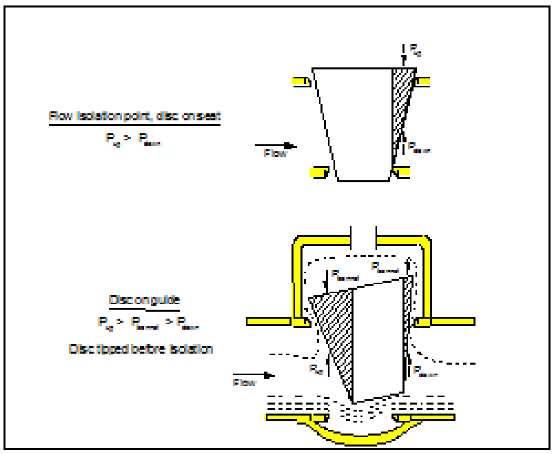


Figure 3-8 Valve Disk Tip

Use of this method for valves with non-linear performance assumes that valve damage is not a concern. (In some instances, non-linear performance is accompanied by damage to the guides and/or seats.) In instances where damage does not occur, the best effort differential pressure test described above (200 psid or half the design basis differential pressure) is probably sufficient to produce a baseline non-linear response that can be used in an extrapolation. The flow must be sufficient to produce the minimum differential pressure before flow isolation. In cases where the design basis conditions include a low degree of subcooling, it might be a good idea to approach or match that degree of subcooling in the best effort test, so that any effects due to the flashing of steam in the orifice are duplicated. Remember, too, that hot water tends to produce a lower disc friction than cold water, and hot water near boiling tends to produce a lower disc friction than hot water with a large degree of subcooling. The test must be conducted with adequate measurements of stem thrust, upstream pressure, and differential pressure to provide input to the following formula:

$$F_{stem} = C_{hook}\,\Delta P + P_{up}\,A_{stem} + F_{pack}$$

Equation (3-9)

Where

F_{stem} = stem thrust

C_{hook} = hooking factor

ΔP = differential pressure

P_{up} = upstream pressure

A_{stem} = stem area

F_{pack} = packing drag

The hooking factor is simply a numerical value that broadly represents the combined effects of the disc area, disc friction, and idiosyncrasies of the pressure distribution, disc/guide contact, and disc/seat contact for that particular valve. To use this formula, solve for the hooking factor, using the actual packing drag (measured in a no-load test), the maximum stem thrust measured at the bottom of the hook during the best effort test, and the differential pressure and upstream pressure measured at the point in time when the maximum stem thrust was measured. Then use the hooking factor thus derived, along with the design basis packing drag (not the same as the *measured* packing drag) and the design basis differential and upstream pressures, to calculate an estimate of the stem thrust required at design basis conditions. Note that this method has not been validated. Note also that the results of this kind of extrapolation are applicable only to the single valve that produced the baseline data.

Some valves are subject to damage to the guides or seats when exposed to very high design-basis differential pressures.

Figure 3-9 shows stem thrust traces from tests in which valve damage occurred (compare with Figure 3-7 and with Figure 3-3). Of the six valves we tested in our 1989 NRC/INL valve tests, two suffered significant guide and/or seat damage during closure against very high design basis loads.

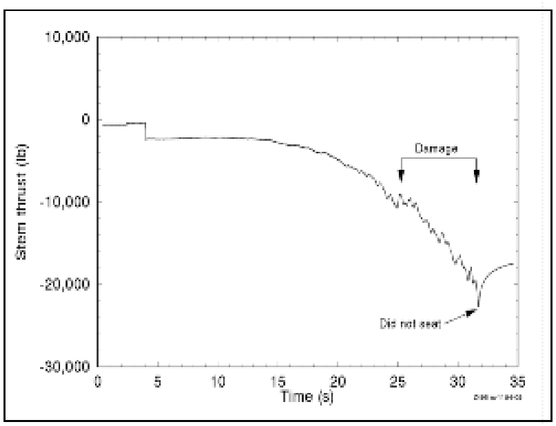


Figure 3-10 Valve Damage to Disc Guides

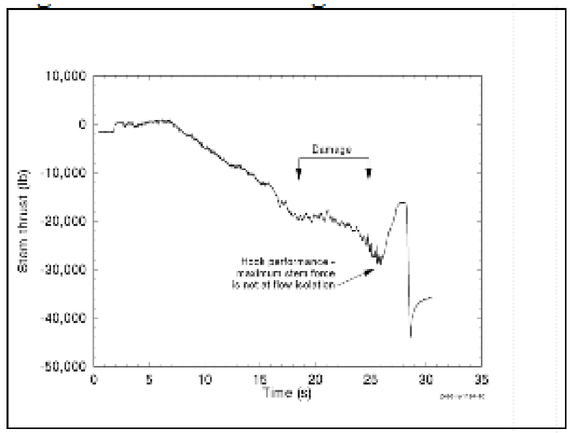


Figure 3-9 Stem Thrust Measurements During Damage

Figure 3-10 is a photograph showing typical damage.

Figure 3-11 illustrates how tipping of the disc during closure at high flow loads can cause damage to either the seats or guides, depending on which components bear the excessive load at the point of contact. We know of no reliable method for determining in advance which valves are likely to experience such damage. Similarly, we know of no reliable method for predicting the requirements of such valves.

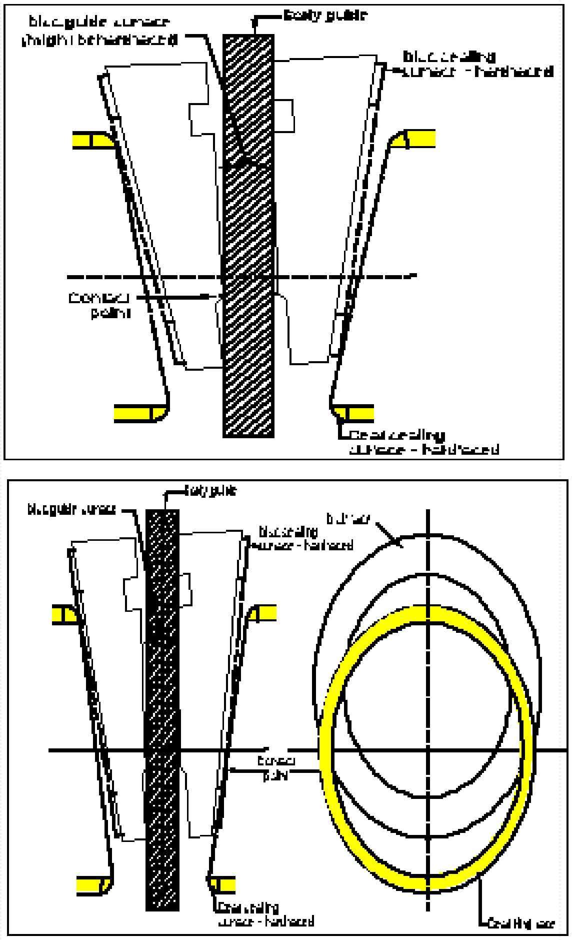


Figure 3-11 Point of Contact Damage

Opening Gate Valves

The design basis operating requirements for some valves include a requirement to be able to open against specified loads and at specified conditions. In general, however, any valve in a safety-related system needs to have the capability to open against any conceivable differential pressure that might occur in the system in which the valve is installed; such capability might prove essential in a scenario in which a valve is intentionally or inadvertently closed and then needs to be reopened. The ability of a valve to open can be as important in the mitigation of an accident as the ability to close. This was true, for example, in the loss of feedwater incident (near-accident) at the Davis Besse plant in 1985.

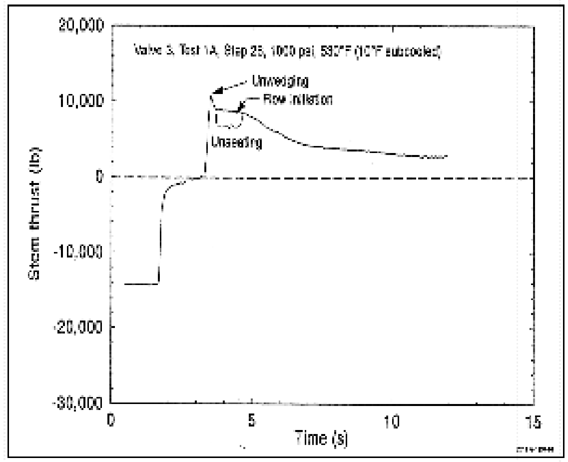


Figure 3-12 Typical Opening Stroke Stem Thrust Measurement

Figure 3-12 shows a typical stem force trace from a valve opening test. The highest load occurs while the disc is sliding on the seat, while the full disc area is exposed to the full differential pressure, before flow initiation. This opening response corresponds with the typical, linear closing response described earlier in this section. A spike occurs at unwedging, a plateau during unseating, and a decline in the load after flow initiation.

Figure 3-13 shows a stem force trace illustrating the occasional nonlinear opening response. In this case, the highest load occurs after flow initiation, creating the appearance of a "hump" in the trace. We performed an analysis that showed that this hump is not a manifestation of an unusually high load after flow initiation, but is instead a manifestation of an unusually low load before flow initiation. This analysis is documented in NUREG/CR-6100, *Gate Valve and Motor-Operator Research Findings*.

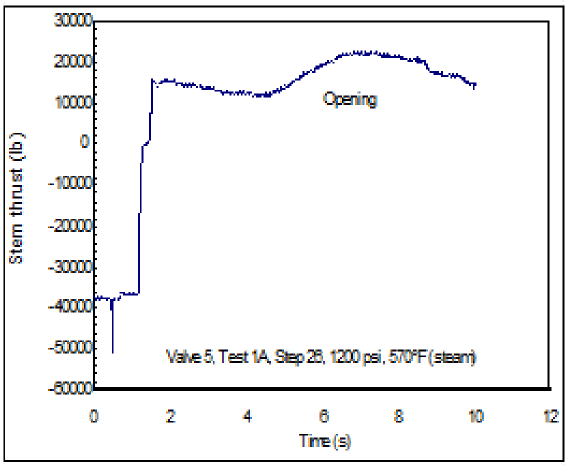


Figure 3-13 Atypical Opening Stroke Stem Thrust Measurement

The formula we use to estimate the opening load is:

$$F_{stem} = F_{pack} - F_{stem\ rej} + F_{elps} \quad \frac{\Delta P\, A_{disc}\,(f\cos\alpha - \sin\alpha) + 80\, A_{disc}}{\cos\alpha + f\sin\alpha}$$

Equation (3-10)

Where

F_{stem} = total stem thrust load

F_{pack} = stem packing load

$F_{stem\ rej}$ = stem rejection load = $P_{up}\, A_{stem}$

P_{up} = upstream pressure

A_{stem} = stem area

$= \pi \left(\frac{D_{stem}}{2}\right)^2$

D_{stem} = stem diameter

F_{elps} = elliptical pressure load

$= \Delta P\, A_{seat}\ \tan\alpha$

ΔP = differential pressure across disc

$= P_{up} - P_{down}$

P_{down} = downstream pressure

A_{disc} = disc area

$= \pi \left(\frac{D_{mean}}{2}\right)^2$

D_{mean} = mean diameter of the disc seat

α = seat angle from vertical

$\tan\alpha$ = 0.08749 for a seat angle of 5°

f = disc/seat coefficient of friction = 0.50

$\cos\alpha$ = 0.99619 for a seat angle of 5°

$\sin\alpha$ = 0.08716 for a seat angle of 5°

This formula is basically the same as the formula for estimating the closing load, except for sign changes. Further, the term 80 A_{disc}, an additive term that provides conservatism to the estimate, is a higher numerical value than the one used in the formula for closing. The term for disc friction in the formula for opening is 0.50 regardless of temperature or degree of subcooling.

A valve evaluation can use this formula, with a disc friction value of 0.50, to estimate a valve's opening thrust, if there is evidence that the valve's behavior is typical of the INL test valves that produced the correlation described below. As with the formula for closing, we recommend that a best effort test be performed with instrumentation sufficient to monitor the stem thrust, upstream pressure, and differential pressure, with a focus on collecting data near the beginning of the opening stroke, while the disc is riding on the downstream seat before flow initiation. If the results of the test, in terms of normalized sliding load versus normalized normal load, fall within the bounds of the INL correlation as shown in Figure 3-14, the valve can be considered typical, and the INL correlation can be considered applicable. If the results fall outside those bounds, the valve cannot be considered typical, and some other method must be used to estimate the disc friction for the valve. Figure 3-14 shows the slope and the bounds of the INL opening correlation, along with the test data that we used to develop the correlation.

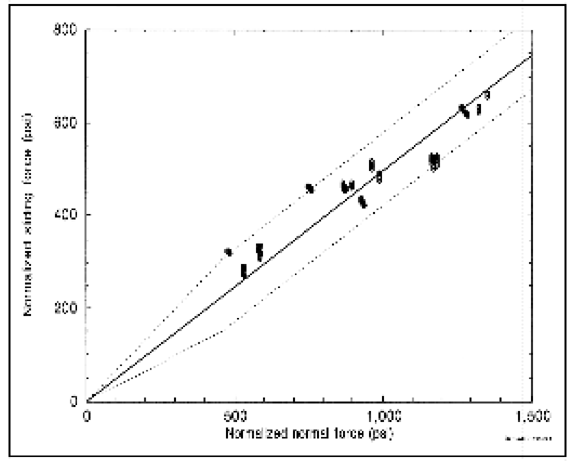


Figure 3-14 INL Correlation Bounds For Predicting Net Stem Thrust

The INL correlation, as shown in Figure 3-14, includes a secondary equation that narrows the acceptable bounds for the data scatter at lower loads. The threshold that separates the lower loads from the moderate to high loads is a normalized normal force F_{nn} of 450 psi, a value that corresponds with a differential pressure of about 425 psid in a valve with a seat angle of 5º. The mathematical representation of the correlation, showing both equations and including the upper and lower bounding terms to account for data scatter, is:

For $F_{nn} > 450$ psi

$$F_{stem} = F_{pack} - F_{stem\ rej} + F_{elps} + \frac{\Delta P\, A_{disc}\,(f\cos\alpha - \sin\alpha) \pm 80\, A_{disc}}{\cos\alpha + f\sin\alpha}$$

Equation (3-11)

For F_{nn} < 450 psi

$$F_{stem} = F_{pack} - F_{stem\ rej} + F_{elps} + \frac{\Delta P\, A_{disc}\,[(1.0 \pm 0.35) f \cos\alpha + \sin\alpha]}{\cos\alpha - (1.0 \pm 0.35) f \sin\alpha}$$

Equation (3-12)

The terms ± 80 A_{disc} and 1.0 ± 0.35 provide the correlation with the means to bound the scatter in the test results upon which the correlation is based.

In general, experience has shown that the thrust needed to unwedge a valve is lower in magnitude than the thrust that wedged the valve during the previous closing stroke. The point of interest for an opening stroke is after unwedging but before flow initiation, when the full differential pressure and the full disc area are effective. For most gate valves, the available thrust at this point is determined by the design basis stem friction and by the actuator capability at design basis motor conditions, not by the torque switch setting; most valve actuators are equipped with a bypass switch, controlled by stem position, that bypasses the torque switch during the early part of the opening stroke, until after flow initiation.

Parallel Disc Gate Valves

Of course, the main difference between flexible wedge gate valves and parallel disc gate valves is the shape of the disc. Figure 3-15 shows a cross section of a parallel disc gate valve. In the flex-wedge design discussed earlier in this textbook, the wedge-shaped disc seals by seating against a pair of matching valve body seats. The parallel disc design employs a different mechanism for sealing. For example, in the Anchor Darling parallel disc gate valve, sealing occurs when the disc assembly reaches the bottom of the valve, after the two parallel discs have lowered far enough to cover the two parallel faces of the valve body seats. At that point, further downward motion of the stem actuates a pair of interfacing wedges that pry the two discs apart (they are separate pieces), creating a tight fit by locking them in place against the valve body seats. Some parallel disc gate valves of other manufacture use other mechanisms for prying the discs apart toseal the seats, and some use no such mechanism at all, relying on the upstream pressure to hold the disc assembly against the downstream seat.

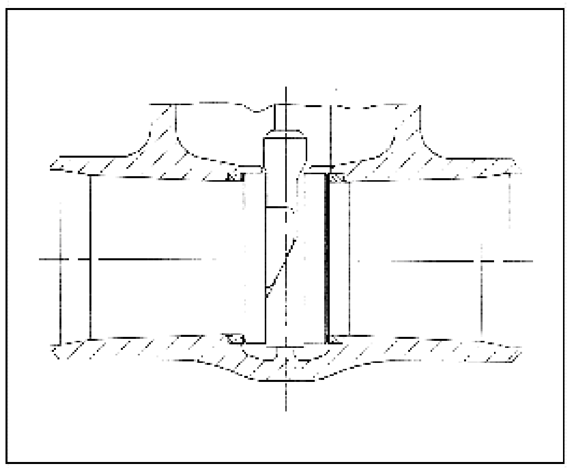


Figure 3-15 Anchor Darling Parallel Disc Gate Valve

The mathematical representation of the closing and opening loads experienced by parallel disc gate valves is similar to that of flex-wedge gate valves, but simpler. In the absence of a wedge shape, there is no elliptical pressure load on the top of the disc. More significant, with the parallel design the normal load on the disc is simply the differential pressure times the disc area, and the sliding load is simply the net stem thrust. Thus, the net stem load is simply the differential pressure times the disc area times the coefficient of friction at the disc/seat interface (disc friction).

Thus, the formula for estimating the required stem thrust for closing a parallel disc gate valve (or for analyzing the results of an instrumented test) is:

$F_{stem} = F_{pack} + F_{stem\ rej} + F_{net\ stem}$

Equation (3-13)

Where

F_{stem} = total stem thrust load

F_{pack} = stem packing load

$F_{stem\ rej}$ = stem rejection load = $P_{up}\ A_{stem}$

P_{up} = upstream pressure

A_{stem} = stem area

$= \pi \left(\frac{D_{stem}}{2}\right)^2$

D_{stem} = stem diameter

$F_{net\ stem}$ = net stem load

$= \Delta P\ A_{disc}\ f$

ΔP = differential pressure across disc

$= P_{up} - P_{down}$

P_{down} = downstream pressure

A_{disc} = disc area

$= \pi \left(\frac{D_{mean}}{2}\right)^2$

D_{mean} = mean diameter of the disc seat

f = disc friction (coefficient of friction at the interface between the disc and downstream valve body seat)

For the opening direction, the formula is:

$F_{stem} = F_{pack} - F_{stem\ rej} + F_{net\ stem}$

Equation (3-14)

The difference being that the stem rejection load assists during opening but resists during closing.

The main concern related to the use of these formulas in valve evaluations is the choice of an appropriate value for the disc friction. Before the late 1980s, the U.S. nuclear industry accepted the use of values as low as 0.20, though little or no testing had been performed to verify the adequacy of this value. It is likely that the use of this low a friction value in valve evaluations could produce inaccurate, unconservative estimates of the net stem load. The use of higher disc friction values, typically 0.40 but ranging from 0.3.to 0.50, is more typical today. The industry's shift to the higher friction values followed publication of the results of the NRC/INL flexible wedge gate valve testing and the results of parallel disc gate valve testing conducted by researchers in the Federal Republic of Germany (results described in U.S. NRC Information Notice 90-72). The higher friction values are more consistent with the results of the NRC/INL and German tests.

Typical Industry Methods and Alternative Methods

Since the mid-1980s, valve researchers, utilities, and utility organizations have developed several new analytical tools for evaluating the performance and the requirements of motor-operated valves in general and wedge-type gate valves in particular. Before that time, several tools were in use for evaluating or predicting the stem thrust load for a flex-wedge gate valve. Though these tools varied somewhat in certain details (the definition of the disc area, the default disc friction factor), they represented basically the same formula. We refer to this formula as the standard industry equation:

$$F_{stem} = F_{pack} \pm P_{up} A_{stem} + \Delta P \, A_{disc} \, \mu_d$$

Equation (3-15)

F_{stem} = total stem thrust load

F_{pack} = stem packing load

$P_{up} A_{stem}$ = stem rejection load = upstream pressure times stem cross-sectional area (added to the load during closing and subtracted from the load during opening)

ΔP = differential pressure

A_{disc} = disc area

μ_d = disc factor

As used in this formula and applied to wedge-type gate valves, the disc factor μ_d is not a true friction coefficient, but serves instead as a catch-all variable that covers the disc friction along with any other parameters not specifically covered by other components of the formula, including any idiosyncrasies associated with this formula's failure to account for the angle (from vertical) of the disc/seat interface.

Before the mid-1980s, use of a disc factor of 0.30 with the standard industry equation was common in U.S. nuclear industry evaluations of valve requirements. By the early 1990s, some evaluations used disc factors as high as 0.50. The definition of the disc area was not consistent. Use of a larger value for the disc area (one based, for example, on the outside disc area) produced a higher estimate of the stem thrust than use of a smaller value (based, for example, on the valve orifice diameter). Use of a smaller disc factor along with a larger disc area might produce a stem force estimate about the same in magnitude as use of a larger disc factor with a smaller disc area.

The standard industry equation is still used by the U.S. nuclear industry in many valve evaluations. We describe it here for completeness, so the reader can easily compare the standard industry equation with the INL formulas we recommend. Part of our research effort in the early 1990s, as we analyzed the results of our gate valve tests, consisted of an appraisal of the standard industry equation. We identified the following deficiencies:

- The disc factor of 0.30 was far too low; in some instances a disc factor of 0.50 was too low.
- It failed to consistently specify the mean seat diameter as the basis for determining the disc area
- It failed to account for the elliptical pressure load on the top of the disc
- It failed to isolate the disc friction, instead including it in the disc factor along with other unknown or unspecified variables.

At about the same time as we published the INL correlation, the EPRI's Nuclear Maintenance Assistance Center (NMAC) was developing an improved equation for gate valve evaluations. The NMAC equation is a proprietary methodology, very similar to the INL correlation.

Pressure Locking and Thermal Binding of Gate Valves

Pressure locking occurs when the high pressure of fluid trapped in the valve bonnet causes the valve to be difficult to open. Thermal binding occurs when thermal expansion/contraction effects squeeze the valve disc between the two seats, likewise causing the valve to be difficult to open.

In general, when a gate valve opens against an ordinary differential pressure load, the single major load the actuator must overcome is the resistance created by friction at the downstream disc/seat interface. Under differential pressure conditions, the upstream pressure tends to decrease or eliminate the load at the upstream disc/seat interface and apply the entire load to the downstream disc/seat interface. Typical formulas for estimating valve operating requirements are based on differential pressure across only one disc/seat interface.

Pressure locking occurs when the valve bonnet pressure is higher than both the upstream and downstream pressures. In most gate valves, including most flexible wedge gate valves, split wedge gate valves, and double-disc gate valves, the bonnet cavity communicates with the area between the disc faces. The effect is that the pressure of the fluid between the discs acts on both the upstream and the downstream discs, introducing resistance to motion at both disc/seat interfaces rather than just one. This adds to the total force necessary to unwedge/unseat the valve disc, compared with the ordinary differential pressure case. The various forces involved in a pressure lock (in a flexible-wedge gate valve) are indicated in Figure 3-16; notice in particular the pressure forces between the two disc halves. At its worst, pressure locking causes the valve to be locked in the closed position, such that the actuator does not have sufficient output capacity to open it (the motor stalls).

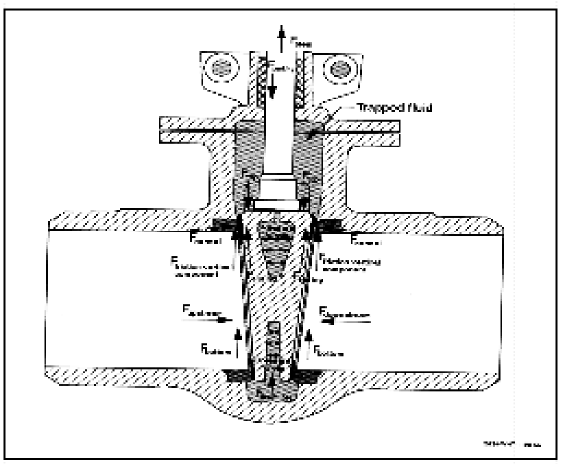


Figure 3-16 Pressure Locked Gate Valve

Pressure locking loads are much more difficult to predict than ordinary differential pressure loads (mentioned above and discussed earlier in this textbook), especially with flex-wedge gate designs. (The flex-wedge design is the most widely used of all gate valve disc designs.) With double-disc and split wedge gate valves, both discs respond equally and independently to the pressure of the fluid between the discs. However, the disc assembly in a flex-wedge valve is made from a single piece of metal, with the upstream and downstream halves of the disc connected in the center by a hub. As with the double-disc design, the two discs respond to the pressure of the fluid between them, but the area exposed to the pressure is smaller, because of the presence of the hub.

In addition, not all of the area exposed to the bonnet pressure (in the flex-wedge design) responds in a way that results in additional force at the disc/seat interface. When the disc assembly is exposed to pressure locking loads, part of the pressure load deforms the disc and presses it against the valve body seat, and part of the pressure load is reacted in the hub. The effect, theoretically, is that for valves of a given size, the more flexible disc design is more likely to be affected by pressure locking loads, with a greater stem force necessary to unwedge a pressure-locked disc assembly.

One other feature of the flex-wedge gate design contributes to the effects of pressure locking. The angle of the disc, usually about 5 degrees from vertical (in an upright valve), creates a horizontal area on the top of the disc that is acted on vertically by the difference between the bonnet pressure and the downstream pressure, and another acted on by difference between the bonnet pressure and the upstream pressure, as shown in Figure 3-17. These areas are typically modeled as elliptical areas defined by the downstream and upstream seat orifices in the valve when viewed from above (the axis parallel to the stem). The corresponding forces, indicated as F_{top} - F_{bottom} in Figure 3-16 and as F_{elps} in Figure 3-17, resist opening in much the same way that the stem rejection load assists opening.

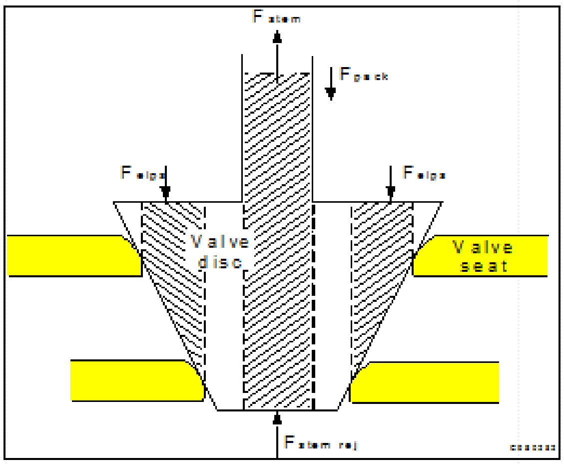


Figure 3-17 Locking Load Forces

When a valve opens against normal differential pressure, only the downstream F_{elps} force is active; the upstream F_{elps} force is zero, because the bonnet pressure is equal to the upstream pressure. However, in the pressure locked case, both the upstream and downstream F_{elps} loads act to resist opening, resulting in yet another increase in the opening load, as compared to the ordinary differential pressure opening situation. The magnitude of the load increase depends on the bonnet, upstream, and downstream pressures.

Taken together, the load increases described in the preceding paragraphs can cause the thrust needed to open a pressure locked valve to be higher than the value typically calculated by industry formulas for the design basis differential pressure conditions. Since most valve actuators are sized and set according to design basis conditions, the higher thrust demands due to pressure locking can exceed the capability of the actuator, causing the valve to fail to open (that is, the torque switch trips or the motor stalls before the actuator successfully unseats and opens the valve).

The bonnet pressure that causes pressure locking can be either hydraulically or thermally induced. Hydraulically induced pressure locking can result from various operational sequences involving low-pressure system interface with high-pressure systems, or from system depressurization during an accident. In a typical scenario, a valve closed at high pressure might experience pressure locking if an attempt is made to reopen the valve after both the upstream and downstream piping has been depressurized, and with the high pressure remaining in the bonnet. Such a scenario occurred, for example, in 1991 at the Fitzpatrick Nuclear Power Plant (discussed in the next paragraph). Thermally induced pressure locking can occur by thermal expansion of water trapped in the bonnet. A valve closed under cold conditions might experience pressure locking if the valve were later heated by a slug of hot fluid coming into contact with the closed disc, by convection or conduction from adjacent hot piping, or by steam from a line break inside the containment. An instance of thermally induced pressure locking occurred, for example, in 1995 at the Susquehanna Steam Electric Station (discussed in a subsequent paragraph).

The instance of pressure locking that occurred at the Fitzpatrick station was hydraulically induced (NRC Information Notice 92-26). The utility hydro-tested the piping between the inboard and outboard low-pressure coolant injection (LPCI) valves. The inboard LPCI valve is a 24-in. flexible-wedge motor-operated gate valve. After the hydro-test, the utility depressurized the piping between the valves and filled and vented the system to return it to service, unaware that high-pressure fluid remained trapped in the valve bonnet. About 10 hours later the utility commanded the inboard valve to open. The valve actuator was energized for about 30 seconds, and then the circuit breaker tripped. (The normal stroke time for this valve is 120 seconds.) The valve had failed to open. The root cause of the failure was pressure locking.

An instance of thermally induced pressure locking was reported at the Susquehanna Steam Electric Station in 1995 (NRC Information Notice 96-08). When performing a valve modification to eliminate the potential for pressure locking, the utility discovered internal damage to a high-pressure coolant injection (HPCI) valve in Unit 1. The utility determined that the damage had been caused by high pressure produced by heatup of fluid trapped in the valve bonnet. The heat source was the hot water in the feedwater system, which has a connection about three pipe diameters from the HPCI valve. No known attempt to open the valve was made while the valve was in a pressure locked condition, but the utility's analysis revealed that the actuator for the HPCI valve did not have sufficient thrust capability to open the valve against the pressure locking load that caused the valve damage.

Other examples of pressure locking are discussed in NUREG-1275.

The magnitudes of possible loads due to pressure locking depend on the valve design and on the pressures prevalent in the specific system where the valve is installed. Leaking valves tend to be less susceptible to pressure locking, because the leakage may prevent the bonnet from becoming or remaining pressurized. However, variations due to valve seating conditions, pressure conditions, and pressure changes cause valve leakage to be inconsistent, so that a valve that leaks under some conditions might not leak under other conditions. Typical modifications to gate valves to prevent pressure locking include venting the bonnet to the high-pressure side by drilling a hole through the disc, or by installing a small vent line between the bonnet and the upstream piping. The vent line might or might not be equipped with a check valve or a block valve.

Thermal binding is a term describing the effects of heatup and cooldown on differential expansion and contraction of the valve internals. Valves closed in the hot condition might experience thermal binding loads when the seat rings contract against the disc after cooldown, as shown in Figure 3-18. These binding loads may be minor, or they may be so large that the valve must be reheated in order to free the disc. In theory, a more flexible disc design is less likely to be affected by thermal binding loads. (As mentioned earlier, the opposite is true of pressure locking loads.)

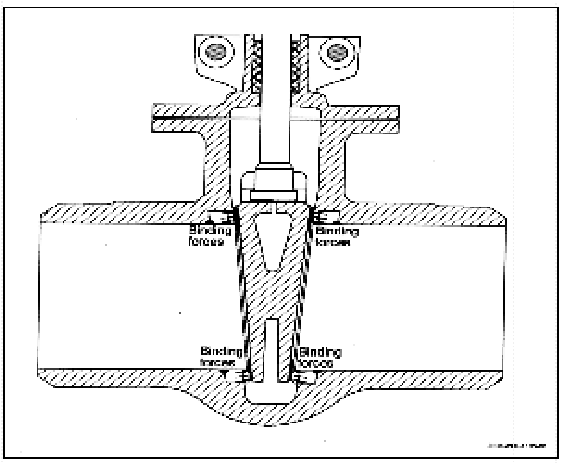


Figure 3-18 Thermal Binding Forces

Precisely estimating a pressure locking or thermal binding load is not a simple task. Theoretically, the pressure locking load in a parallel disc gate valve might be as high as twice the ordinary opening load against the same differential pressure. However, results from our NRC/INL laboratory tests showed pressure locking loads 2.05 to 2.40 times the ordinary opening pressure load. We do not know why the loads were higher than expected, but we wonder if the wedge-to-wedge mechanism in the Anchor Darling valve we tested failed to function normally, such that the disc assembly failed to collapse.

Theoretically, the pressure locking load in a flexible wedge gate valve should be twice the ordinary opening load against the same differential pressure, minus the differential pressure times twice the area of the cross-section of the hub, minus an unknown variable to account for the stiffness of the disc/hub assembly. Results from our NRC/INL laboratory tests of a flexible wedge gate valve showed pressure locking loads 1.1 to 1.5 times the ordinary opening pressure load. (These values are applicable only to the specific valve that produced the data, and are not applicable to any other valve.) The U.S. nuclear industry is preparing a formula for estimating pressure locking loads; this formula includes a calculation that accounts for disc/hub stiffness. The calculation is very complicated and is still undergoing validation (in 1999).

We know of no method for estimating thermal binding loads. An analysis of some NRC/INL test data showed that in instances where valves were subjected to thermal and depressurization cycles that might induce thermal binding conditions, the unseating loads ranged from about 40 to 120% of the previous seating load. In contrast, the ordinary unseating load, in the absence of any such thermal cycles, is typically about 20% of the previous seating load. The obvious risk is that a high unwedging load might cause the motor to stall, leaving the valve closed. The results of the NRC/INL pressure locking and thermal binding tests are documented in NUREG/CR-6611, *Results of Pressure Locking and Thermal Binding Tests of Gate Valves*.