3.0 THEORY OF MOV DESIGN BASIS OPERATION

3.1 MOV Functional Margin

An evaluation of the operation of a motor-operated gate valve begins with two simple questions. (1) Given a certain differential pressure across the disc, how much thrust (downward force) will be needed to lower the disc to close the valve? (2) How much thrust is the valve actuator capable of delivering? The following discussion takes a close look at these questions.

Unless otherwise noted, the discussion assumes a gate valve operating in the closing direction. We will address issues related specifically to the opening direction later in this textbook. The discussion also assumes a valve oriented with the stem vertical, so that downward means in the closing direction, and horizontal refers to the orientation of the valve inlet and outlet.

For clarity and simplicity, the discussion focuses on a rising-stem flexible-wedge gate valve equipped with a Limitorque ac-powered actuator. Actuators of other manufacture (Rotork, for example) differ from Limitorque actuators in their design details, but their operation is similar enough that principles applicable to a Limitorque actuator apply in a general way to all electric-motor-powered valve actuators.

A diagram showing a flexible-wedge gate valve equipped with a Limitorque actuator is shown in Figure 3-1. The main components are the electric motor, the gearbox, the yoke, and the valve. The figure includes illustration of the terminology used in the following discussion. This diagram illustrates all the important issues related to the function of motor-operated gate valves.

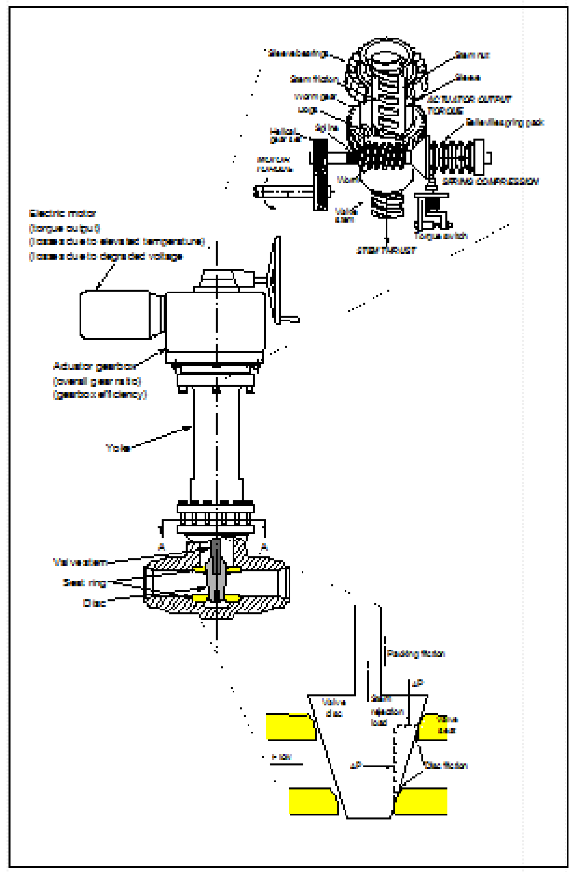


Figure 3-1 Limitorque Actuator with Flexible Wedge Gate Valve

Figure 3-1 shows details of valve components, illustrating the technical issues involved in evaluating valve operability.

The effort to answer the two simple questions cited previously must necessarily address a rather long list of specific questions about specific components and specific parameters. These can be broadly categorized into four groups, as follows:

Group 1: Pressure and friction loads on the valve stem, disc, and seat.

- What is the friction load (a portion of the stem thrust load) created by the passing of the valve stem through the packing seal? (packing friction load)
- What is the cross-sectional are of stem?
- What is the fluid pressure on the upstream side of the disc? (Upstream pressure times stem area equals stem rejection load.)
- What is the differential pressure (DP) across the disc?
- What is the disc surface area exposed to the differential pressure? (Disc area times differential pressure equals the force pushing the disc against the seat.) What definition of disc area is appropriate for the calculation? (The definition might be based on nominal valve diameter, inside seat diameter, mean seat diameter, or outside disc diameter.)
- What are the other pressure forces acting on the disc? (The wedge shape of a flexible-wedge disc creates an area elliptical in shape on the horizontal plane on the top of the disc; this area is acted on by the differential pressure, pushing downward on the disc.)
- What other forces are peculiar to the shape of the disc?
- What is the disc friction, that is, the coefficient of friction at the interface where the disc slides on the valve body seat? (This one turns out to be the most important question of all.)
- What are the applicable structural limits for the valve and gearbox?
- What is the possibility of damage to the disc, guides, and/or seat during closure against high flows? How would such damage affect the valve's thrust requirement?

Group 2: Friction load at the valve stem and stem nut.

- What are the diameter, thread pitch, and thread lead for the valve stem?
- What is the coefficient of friction at the interface where the stem nut turns on the stem? How do variations in the stem friction affect the diagnosis of the valve's operability?
- What torque switch setting is appropriate for the design basis loads anticipated for the valve?

Group 3: Output of the electric motor.

- What is the rated torque of the electric motor that drives the actuator?
- At what motor speed is that torque delivered? (This issue is more important to dc motors than to ac motors.)
- How does operation at elevated temperature affect the output of the motor?
- How does operation at reduced voltage affect the output of the motor?

Group 4: Performance of the actuator gearbox.

- What is the overall gear ratio of the gearbox?
- What will the efficiency of the gearbox be at the expected operating conditions? (The actuator manufacturer publishes nominal values for the running efficiency, pullout efficiency, and stall efficiency; the actual efficiency might be different from the published nominal values.)

The normal operation of a motor-operated gate valve (in the closing direction) is as follows (refer to Figure 3-1 for terminology and illustration). The ac electric motor, typically turning at about 1700 rpm in an ac-powered operator (some ac motors turn at other speeds, for example, 3400 rpm), provides input to the gearbox, where a gear reduction increases the torque and reduces the speed. The gear reduction typically includes a set of helical gears and a worm/worm-gear set. The worm drives the worm gear, which directly drives the stem nut. The torque applied by the gearbox to the stem nut is the actuator torque, referred to frequently in this textbook (see upper inset, Figure 3). The stem nut turns on the threaded portion of the valve stem. Thrust bearings hold the stem nut in place, so that the rotation of the stem nut on the stem drives the stem downward. The downward movement of the stem drives the disc downward to close the valve by blocking the valve's orifice.

In a typical actuator, the worm gear and stem nut are equipped with dogs, such that at the beginning of the closing stroke, the worm gear makes about one-half turn on the stem nut before engagement; the stem nut does not begin to turn until the worm gear dog hits the stem nut dog. The same thing happens at the beginning of the opening stroke: the worm gear makes about a half turn on the stem nut in the opening direction, the worm gear dog engages the stem nut dog, and then the stem nut turns on the stem. This mechanism allows the motor to get up to speed on startup before it must operate against a load. Guides on both sides of the valve orifice interface with guides on the disc to stabilize the position of the disc as it travels. In some valves, the guides are hard surfaced, but in most they are not. As the leading edge of the wedge-shaped disc approaches the bottom of the valve orifice, the disc transitions from travelling on the guides to travelling on the downstream valve body seat. At the moment of flow isolation, the maximum applicable surface area of the disc is exposed to the maximum differential pressure across the disc. This point in the closing stroke is usually the point at which the crucial load occurs. At this point, the downstream disc seat is pressed against the downstream valve body seat by the differential pressure, but there is still a gap between the upstream valve body seat and the upstream disc seat. After flow isolation, the stem continues to drive the disc downward into the seats. The stem presses the wedge-shaped disc into the wedge-shaped seat assembly in the valve, producing (hopefully) a leak-tight fit.

By this point in the closing stroke, resistance to motion in the stem (the stem thrust load) has already caused the stem nut to be difficult to turn on the stem. This resistance to rotation at the stem nut has caused the worm (in the gearbox) to move itself in relation to the worm gear (move to the right in Figure 3-1, upper inset) and to partially compress the torque spring. When the disc seats, downward stem motion virtually ceases, stem nut rotation virtually ceases, and as the helical gear output shaft continues to turn, the worm climbs the now stationary worm gear, sliding further on the splined shaft (to the right in Figure 3-1, upper inset) until it trips the torque switch. The tripping of the torque switch actuates a relay, which in turn shuts off power to the electric motor. Motor momentum continues to drive the worm against the torque spring after the torque switch trips, producing an increase in stem thrust after torque switch trip.

The preceding description assumes a valve actuator controlled by a torque switch. This is a typical configuration. In this configuration, the torque switch serves to limit the amount of torque the valve actuator can produce. The high torque load that occurs when the disc wedges in the seats is what shuts the motor off, by means of the torque switch. Some valves, however, are controlled by limit switches that turn the motor off on valve position rather than on actuator torque. These valves might or might not include a torque switch that serves as a safety device to prevent structural overloads, for example, in the event that the limit switch fails to shut off the motor in time. Whether the torque switch serves as a control device or a safety device, the setting of the torque switch is equally important. An incorrectly set torque switch can either (a) cause the motor to shut off before the valve is all the way closed, or (b) cause the motor to stall after the valve is closed, by failing to shut off power to the motor.

Usually the actuators are equipped with a torque switch bypass switch controlled by valve position. The purpose is to bypass the torque switch at the beginning of the opening stroke to allow the actuator to get the disc off the seat without a torque limit. After the valve position reaches an adjustable set point, the bypass switch is disabled, and the torque switch becomes the controlling mechanism. (The bypass switch was incorrectly set in the Davis-Besse valves that failed to open in the 1985 event.)

Let's take a closer look at the parameters that affect a valve's ability to operate, beginning with the rated torque output of the electric motor. The typical analysis assumes that the actual output is at least as great as the rated output. Typical in-plant diagnostic tests do not measure actual motor torque output, but motor torque can be inferred from measurements of motor voltage and current, if the motor has been performance tested, or if the manufacturer's generic performance curves are available. The analysis multiplies the rated motor torque by a factor to account for operation at elevated temperature, then by another factor to account for operation at reduced voltage, if applicable. The outcome of this calculation is the estimated output torque of the electric motor.

The estimated output torque of the motor is the input torque to the gearbox. The gearbox includes a set of helical gears that usually (but not always) provides a speed reduction (and corresponding torque increase) to the drive train. Another, usually larger speed reduction (and torque increase) occurs where the worm drives the worm gear (refer to Figure 3-1, upper inset). The worm gear directly drives the stem nut. Torque applied to the stem nut by the actuator is the gearbox output torque. The gearbox output torque can be measured as the reaction torque in the valve stem, with appropriately installed instruments, such as strain gauges mounted on the stem or torque cells mounted where the valve yoke is fastened to either the gearbox or the valve. Gearbox output torque can be roughly estimated from measurements of torque spring compression or torque spring reaction force, once the applied torque is great enough to begin to compress the torque spring. Gearbox output torque can also be estimated from the gearbox input torque times the overall gear ratio (as produced by both the helical gear set and the worm/worm-gear mechanism) times the efficiency of the gearbox.

The rotation of the stem nut on the threaded stem is the mechanical process that converts the actuator output torque to thrust in the valve stem. The mathematical relationship between the actuator torque and the stem thrust is known as the stem factor (torque divided by thrust equals stem factor.) For a given valve stem and stem nut, the stem factor consists partly of mathematical constants determined by the stem diameter, pitch, and lead configurations (how many rotations of the stem nut produce how much linear travel in the stem). The single variable in this relationship is the coefficient of friction at the interface between the stem nut and the stem. In this textbook, for brevity, we refer to this stem/stem-nut coefficient of friction simply as stem friction. In any valve, the stem factor (and the stem friction) can be calculated from measurements of actuator torque and stem thrust, if the valve is instrumented for both measurements. (Measurement of stem torque can substitute for measurement of actuator torque, since the reaction torque in the stem, which is mechanically prevented from turning, is equal to the torque applied to the stem nut by the worm gear.) Variability in the stem friction is one of the difficulties the analyst must face in any effort to diagnose the operation of a gate valve.

As the actuator moves the stem downward, it encounters resistance created by several minor loads and one major load. The sum of those loads is the total stem load, which is equivalent to the total stem thrust that the actuator must apply as it closes the valve. The total stem thrust can be measured with appropriate instrumentation (load cells, strain gauges) attached at the yoke connection or attached to the valve stem or to the yoke. Stems equipped with strain gauges already calibrated for measurement of both thrust and torque are commercially available. In the absence of measurements, the total stem load can be estimated using appropriate values for the applicable parameters.

The packing load (see Figure 3-1, lower inset) is one of the minor loads; this load is caused by friction where the stem passes through the packing seal at the top of the valve. This is typically an adjustable seal. The load here can vary depending on how tight the adjustment is. If the packing adjustment is too loose, the seal will leak. With appropriate instrumentation, the analyst can directly measure the packing load in tests with no flow or pressure in the valve, or he can calculate it from the results of tests with pressure but no flow. In the absence of measurements, a default value for the packing load can be calculated using a formula provided by the valve manufacturer.

The stem rejection load (another minor load) consists of the pressure inside the valve (upstream pressure) trying to expel the stem (Figure 3-1, lower inset). The analyst can calculate this load from measurements or estimates of upstream pressure (pressure times stem cross-sectional area equals stem rejection load). The stem rejection load resists stem motion during valve closure but assists during an opening stroke.

Another minor load is created by the pressure in the bonnet (the same as the upstream pressure) pushing down on the top of the wedge-shaped disc (after flow isolation but before full wedging). This load can be directly calculated as the differential pressure (upstream pressure minus downstream pressure) times the area of the ellipse representing the downstream seat orifice when viewed from above. This elliptical area is created by the angle (from vertical) of the downstream seat in a wedge-type valve In Figure 3-1 (lower inset), this load is represented as the differential pressure (DP) pushing down on the top of the disc. For brevity, we call this load the elliptical pressure load; namely, the load created by the differential pressure on the ellipse. This minor load on the disc resists stem motion during an opening stroke but assists during valve closure.

The major load that creates resistance to stem motion is the load that, for brevity, we call the net stem load; namely, the vertical load imposed on the stem by the disc's resistance to motion as it slides on the seat. We call it the net stem load because it represents the portion of the stem load that remains after the other portions of the stem load (the three minor loads) have been accounted for. Ignoring the angle of the downstream seat, we can define the net stem load as the horizontal force pushing the disc against the seat, times the coefficient of friction at the interface between the disc sealing surfaces and the seat sealing surfaces. For simplicity and brevity, we refer to this disc/seat coefficient of friction as the disc friction. Different metals have different friction characteristics. In most gate valves, the disc and seat sealing surfaces are hard surfaced with Stellite 6, a cobalt-based alloy. In this course, reference to disc friction almost always assumes wet Stellite 6 surfaces with little or no aging (very little or no corrosion). Exceptions are noted where they occur.

The horizontal force pushing the disc against the seat is mathematically represented as the disc area times the differential pressure (horizontal DP in Figure 3-1, lower inset). In this textbook, we define the disc area in terms of the mean seat diameter, that is, the distance from the middle of the seat on one side to the middle of the seat on the other side. Other possible definitions have been used in various valve analysis methods, including the valve nominal diameter, the orifice diameter (inside seat diameter), and the outside disc diameter. The definition we use assumes that the very slight curvature of the valve seating surface and the disc seating surface causes the actual point of contact to be the middle of the seat.

Earlier we phrased our two simple questions in terms of stem thrust: that is, how much thrust is needed (required thrust), and how much thrust is available (available thrust). If the available thrust is greater than the required thrust, the valve will close. The difference between the available thrust and the required thrust is the thrust margin. Various changes in the parameters illustrated in Figure 3-1 can result in changes in the margin. For example, an increase in the disc friction, an increase in the stem friction, a decrease in the torque switch setting, an increase in gearbox friction, or a decrease in the motor output torque can cause a decrease in the margin. If the margin decreases beyond zero, or in other words, if the required thrust is greater than the available thrust, the valve will not close.

In some contexts, it makes more sense to look at the margin in terms of actuator torque instead of stem thrust. Actuator torque is the amount of torque applied to the stem nut by the actuator as it turns the stem nut on the stem (Figure 3-1, upper inset). When we look at the torque margin instead of the thrust margin, we phrase the two simple questions a little differently: How much actuator torque is needed? (required torque) How much torque is available? (available torque) One advantage of this view is that a discussion of torque margin is applicable not only to rising stem valves (gate valves and globe valves), but also to quarter turn valves (butterfly valves, ball valves, and plug valves). The main advantage, however, is that the available torque is directly controlled by the torque switch, while the available thrust is not. The available thrust is only indirectly controlled by the torque switch. At the same torque switch setting, the thrust delivered to the valve at torque switch trip can vary with changes in the stem friction, which can vary depending, for example, on the load profile imposed on the valve during the closing stroke. In addition, the stem friction can vary from one valve to the next. (These and other issues related to stem friction are discussed in detail in a subsequent chapter.) The view that looks at torque margin instead of thrust margin can treat the torque switch setting as the parameter that directly controls the available actuator output.

Most of the discussion in this textbook focuses on torque margin rather than thrust margin. Exceptions are noted where they occur. The main difference between the two views, in terms of how the difference affects the evaluation of the valve's operability, is that the torque margin view treats the stem friction as one of the loads that contributes to the requirements calculation, while the thrust margin view treats the stem friction as one of the variable parameters that affects what is available.

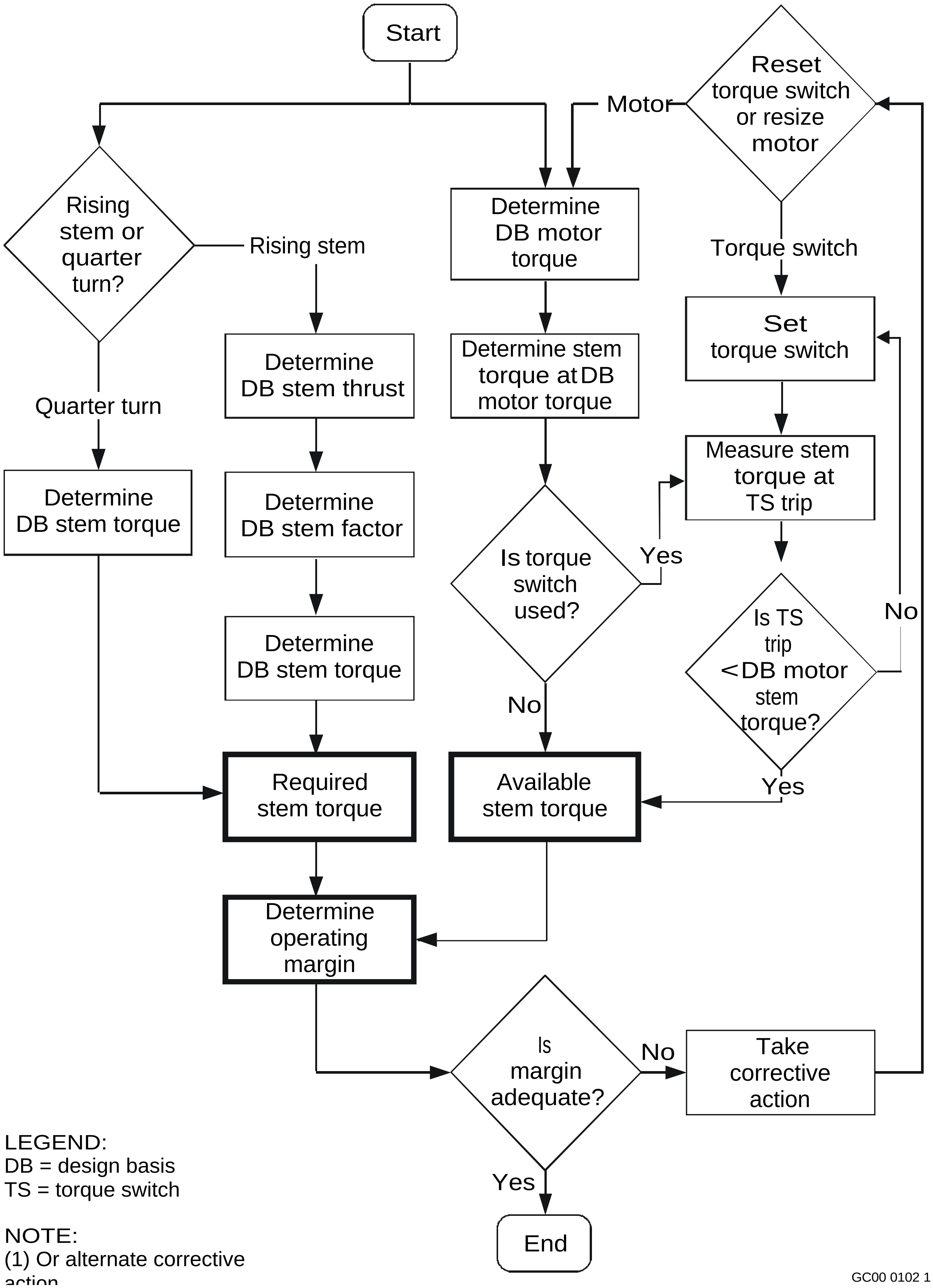
Ideally, the torque switch alone determines the available actuator torque. However, an unfavorable situation can occur if the motor output and gearbox friction parameters are such that the capability of the actuator is less than that anticipated by the torque switch setting. In this case, the motor will stall before the torque switch trips. Thus, it can be said that the torque switch setting usually determines the available actuator torque, but if conditions are such that the motor will stall before the torque switch trips, it is the capability of the electric motor, not the torque switch setting, that limits the available actuator torque.

In some valve applications there is no torque switch, or the torque switch has been bypassed, and limit switches control the valve's operation. In these valves, the available actuator torque is determined, ultimately, by the capability of the motor. In the event of too great a stem load, the electric motor will stall when its torque capability is exceeded. If the motor is not protected by a thermal overload switch, or if the overload switch has been bypassed, a stalled motor will burn out.

If an evaluation of valve operability determines that the margin is small or non-existent, the solution is not always so simple as to change the torque switch to a higher setting. Before making such a change, the analyst must ensure that the higher torque switch setting does not create a situation where the motor might stall before the torque switch trips. Even if the motor is protected by thermal overload switches, such an occurrence is not acceptable in a safety-related valve, because the delay while plant operators wait for the overload switch to reset might make an accident worse. If the analysis determines that a higher torque switch setting might cause the motor to stall, it might be necessary to upgrade the valve by installing a more powerful motor or actuator (changing the gear ratio in the actuator might suffice).

In addition, the analyst must ensure that the higher torque switch setting does not create a situation where the actuator exceeds the physical displacement limits of the torque spring or the overload limits of the valve or the gearbox. If the torque switch is set too high, the torque spring might reach its maximum compression before the torque switch trips, in which case the motor will stall (the torque switch cannot trip). Overload limits are a concern if the torque switch setting is such that, with sufficiently powerful motor and actuator, the torque or thrust output of the actuator exceeds the manufacturers' published allowable torque and thrust limits for the valve or the gearbox before the torque switch trips. The intent of these limits is to prevent damage to the valve body, disc and valve seating surfaces, stem, gears, shafts, bearings, or gearbox housing. One of the purposes of the torque switch is to prevent such damage.

In summary, the task of the analyst is to determine whether the actuator has sufficient operating margin to close the valve at design basis conditions. Figure 3-2 is a flow chart presenting a simplified visual overview of the considerations involved in this task. (These considerations are described in detail in subsequent chapters of this textbook.)


Start
Rising stem or quarter turn?
Rising stem
Quarter turn
Determine DB stem torque
Determine DB stem thrust
Determine DB stem factor
Determine DB stem torque
Required stem torque
Determine DB motor torque
Determine stem torque at DB motor torque
Is torque switch used?
Yes
No
Available stem torque
Reset torque switch or resize motor
Motor
Torque switch
Set torque switch
Measure stem torque at TS trip
Is TS trip < DB motor stem torque?
No
Yes
Determine operating margin
Is margin adequate?
No
Take corrective action
Yes
End
LEGEND:
DB = design basis
TS = torque switch
NOTE:
(1) Or alternate corrective action
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In performing this task, the analyst determines the required actuator torque for the valve's design basis upstream and differential pressures, considering:

- Packing load
- Stem rejection load
- Elliptical pressure load
- Net stem load, including the disc friction
- Stem factor, including stem friction.

The analyst also determines the available actuator torque at the design basis operating conditions for the actuator, considering:

- Torque switch setting
- Rated torque of the electric motor
- Motor torque losses due to operation at reduced (design basis) voltage and elevated (design basis) temperature, if applicable.
- Overall gear ratio in the actuator
- Gearbox efficiency.
- Structural limits of the spring, gearbox, and valve.

The analyst determines (a) that the torque switch setting is adequate to close the valve against the design basis differential pressure and (b) that the motor is powerful enough to trip the torque switch. In making this determination, the analyst uses conservative (high) estimates of disc friction and stem friction and a conservative (low) estimate of gearbox efficiency. The analyst also determines (a) that the torque switch setting does not exceed the physical displacement limits of the torque spring and (b) that the torque switch setting does not exceed the published allowable load ratings of the valve and gearbox. In making the thrust limit determination, the analyst uses a conservative (low) estimate of stem friction.